PERFORMANCE AND EMISSION CHARACTERISTICS OF SPARK IGNITION ENGINES BASED ON ENGINE OPERATING PARAMETERS

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A thesis submitted to Graduate School in fulfillment for the requirements of the degree of Doctor of Philosophy in Energy Engineering of Egerton University

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DECLARATION AND RECOMMENDATION

Declaration

This thesis is my original work and has not been submitted for a degree in any other university.

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Recommendation

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DEDICATION

I dedicate this work to my wife Norah Langat, children; Maureen, Collins, Marion, Michelle, Malia and the memories of my late parents.

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ABSTRACT

Poor vehicle maintenance culture and high proportions of old vehicles are major contributing factors to high vehicle emission levels. The purpose of this study was to determine performance and emission characteristics of vehicles that were taken for inspection at the Vehicle Inspection Centre in Nairobi City based on engine operating parameters and to develop performance and emission prediction models. The specific objectives were; to determine vehicle exhaust emission levels, engine's performance and emission characteristics and to develop performance and emission models. The sample size comprised 384 petrol vehicles randomly selected. The key observations included vehicle usage, compression pressure, ignition angle, engine speed, spark plug gap, and vehicle category. The key variables examined were emission of CO, HC and CO_2, excess air factor (λ) and factors that influence emissions. Logistic regression model was fitted to determine the probability of tested vehicles failing emission tests based on the test variables. Field data were simulated using engine test bed and the effects of engine input variables on engine performance and emission were determined. Sub-model equations were generated from engine performance and emission curves and superimposed to develop engine performance and emission models. Validation, optimization and sensitivity analysis of the models were done. The mean vehicle usage ranged between 14328 km/yr and 19640 km/yr and the lowest compression pressure of 6.8 bar was recorded in the non-catalytic vehicles manufactured before 1986. Both categories of non-catalytic vehicles operated at a rich mixture. There was significant difference between the measured and standard values of exhaust emission gases. The models developed predicted well for engine performance and emission as expressed by low percentage error in most of the points. Optimization of Specific Fuel Consumption (SFC) model gave input variables of 2839 rpm, 16^{o} BTDC, 1.05 λ and spark plug gap of 0.8 mm. Excess air factor was found to be the most sensitive variable when adjusted by ±10%, it mostly affected engine performance and emissions. In conclusion, exhaust emission levels from vehicles measured in Nairobi City were 6.8% vol., 4.41% vol., 1.16% vol. and 0.46% vol. CO for non-catalytic vehicles manufactured before 1986, non-catalytic vehicles manufactured between 1986 and 2002, catalytic vehicles manufactured between 1986 and 2002, and catalytic vehicles manufactured after 2002 respectively. The mean values for HC were 1814 ppm, 1884 ppm, 333 ppm, and 253.4 ppm for non-catalytic vehicles manufactured before 1986, non-catalytic vehicles manufactured between 1986 and 2002, catalytic vehicles manufactured between 1986 and 2002, and catalytic vehicles manufactured after 2002 respectively. These values were significantly different from the limits given in KS 1515-2000. Excess air factor for non-catalytic vehicles manufactured before 1986 and those manufactured between 1986 and 2002 oper were 1.14 and 1.08 respectively, while for catalytic vehicles manufactured between 1986 and 2002, and catalytic vehicles manufactured after 2002 were within the required limit of 1±0.03. There were significant changes in engine performance and emissions when the input variables were changed from optimal values. The models developed predicted well engine performance and emission characteristics. It is recommended that emissions control mechanisms be put in place to reduce emission levels. There is also need to include more test parameters in the models so as to improve on the prediction levels.

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ABBREVIATIONS

AFR	Air Fuel Ratio
AIC	Akaike Information Criterion
API	American Petroleum Institute
BET	Basic Emission Test
BTDC	Before Top Dead Centre
CA	Crank Angle
CE – CERT	California college of Engineering – Centre for Environmental Research
CI	Compression Ignition
CNG	Compressed Natural Gas
CO	Carbon Monoxide
COM	Control Oriented Model
EC	European Commission
EGR	Exhaust Gas Recirculation
EPA	Environmental Protection Agency
EU	European Union
GDP	Gross Domestic Product
GDI	Gasoline Direct Injection
GHGs	Greenhouse Gasses
GPS	Global Positioning Satellites
GSSR	Global Sustainability Systems Research
HC	Hydrocarbon
HEP	High Emitter Profile
IC	Internal combustion
I/M	Inspection and maintenance
IVE	International Vehicle Emission
KEBS	Kenya Bureau of Standards
KNBS	Kenya National Bureau of Statistics
km	Kilometers
KS	Kenya Standards
KRA	Kenya Revenue Authority
LPG	Liquefied Petroleum Gas

MBT	Maximum Brake Torque
NO	Nitrogen Oxide
NOx	Oxides of Nitroge
PCV	Polyaromatric Hydrocarbon
PAH	Positive Crankcase Ventilation
PGE	Petroleum Groups Elements
PM	Particulate matter
ppm	Parts Per Million
RAM	Random Access Memory
RDBMS	Relation Database Management System
Rpm	Revolution per minute
SAE	Society of Automotive Engineers
SAS	Statistical Analyst System
SC	Schwarttz Criterion
SI	Spark Ignition
SFC	Specific fuel consumption
SST	Total Sum of Squares
SQPAL	Sequential Quadratic Programming Algorithm
TDC	Top Dead Centre
TWC	Three Way Catalytic Converter
UNECE	United Nation Economic Commission for Europe
UNEP	United Nations Environment Program
US	United State
VOCE	Vehicle Occupancy Characteristics Enumerator
WHO	World Health Organization
WRI	World Resource Institute

SYMBOLS

Symbol	Name	Unit
bP	Brake power	kW
D	Required levels of precision	
F	Force	N
fp	Friction power	kW
G	Spark plug gap	mm
G_L	Lower spark plug gap	mm
G_H	Upper spark plug gap	mm
h	Clearance volume	m^3
iP	Indicated power	kW
n	Sample size	
N	Engine speed	rpm
P	Pressure	N/m^2
ρ_a	Air density	Kg/m^3
P_c	Compression pressure	N/m^2
P'	Partial pressure	N/m^2
p_m	Mean effective pressure	N/m^2
m	Mass	kg
m_a	Air flow rate	kg/s
$m_{air\ stioch}$	Mass of stoichiometric air	kg
m_{cr}	Mixture mass in crevices	kg
m_f	Fuel flow rate	kg/s
Q_{HVi}	Lower heating value for species i	kJ/Kg
Q_{HVf}	Lower heating value for fuel	kJ/Kg
S	Engine speed	rpm
S_H	Upper engine speed	rpm
S_L	Lower engine speed	rpm
S_l	Laminar flame speed	
r_c	Compression ratio	
T	Engine torque	Nm
T_{man}	Intake air temperature	K
U	Vehicle usage	km/yr

V	Engine cylinder volume	m^3
v_a	Engine cylinder induced volume	m^3
V_{cr}	Volume of crevices	m^3
V_s	Engine cylinder swept volume	m^3
V_i	Ignition voltage	V
v_t	Throttle position	Degrees
v_t	Operating time	Minutes
W_i	Indicated work done per cycle	kJ
X_i	Value of the predictor variable in the i^{th} trial	
X_k	Explanatory Variables	
χ^2	*Chi*-square	
Y	Vehicles' test results	
γ'_i	Langragian multiplier	
y'	No. of moles of actual oxygen supplied to the engine	kmol
Y_i	Predicted exhaust gas content	
y^*	Measured exhaust gas content on the logarithmic scale	
$y_{ph,i}$	Output parameter of non-linear physical model	
Z^2	Normal variant associated with levels of significant	
a_i	i^{-th} regression coefficient	
β	Estimates regression parameters	
σ^2	Variance	
ε_i	Random error term	
n_R	Number of crank revolutions for each power stroke per cycle	
η_c	Combustion efficiency	%
η_f	Fuel conversion efficiency	%
η_m	Mechanical efficiency	%
θ	Ignition angle	Degrees
θ_H	Upper Ignition angle	Degrees
θ_L	Lower Ignition angle	Degrees
$\Delta\theta_c$	Total combustion duration in terms of crank angle	Degrees
μ^*	Parametric mass entrained within the flame region	kg
μ_i	Sub-model parameter	

λ_{L}	Lower λ value	
λ_{H}	Lower λ value	
τ^{*}	Total parametric mass of the gases	kg
ξ_0	Superimposed model parameter	

CHAPTER ONE

INTRODUCTION

Air pollution is one of the most serious environmental concerns in urban areas especially in view of its adverse effects on human health and the environment. Environmental impacts include damages to buildings and structures, vegetation and increasing green house effect (Kojima and Lovei, 2001; Gwilliam *et al*, 2004). In the developing countries around the world, an estimated 0.5 million to 1.0 million people die pre-maturely each year as a result of exposure to urban air pollution. Thousands of premature deaths and millions of respiratory illness cases are associated with air pollution in large cities. Exposure to lead contributes to behavioural problems and learning disabilities in urban children. The economic damage from air pollution was estimated between US$1 and US$4 billion per year in some parts of the world (World Bank 1997a; Kojima and Lovei, 2001; WHO, 2002). In Austria, France and Switzerland, for example, about 6% of all deaths (40,000) per year which is twice the annual deaths from traffic accidents are due to outdoor air pollution (Botter *et al*, 2002; Esteves and Barbosa, 2007). Vehicles are responsible for about half of this total and people living in cities die about 18 months earlier than they otherwise would have. Each year, outdoor air pollution causes over 25,000 new cases of chronic bronchitis, 800,000 episodes of asthma and bronchitis and 16 million lost person days of activity per year making health cost from traffic pollution to be about 1.7% of total GDP in these countries (Botter *et al*, 2002; Esteves and Barbosa, 2007).

Global emissions of all pollutants from on-road vehicles are projected to be substantially higher by 2030 than they are today as emissions from all pollutants are growing very rapidly and are projected to be three to six times higher unless strong control programs are implemented (Kojima and Lovei, 2001). The growth in vehicle emissions is of great concern to governments as they strive to protect public health and welfare. The harmful effects of conventional pollutants from motor vehicles such as hydrocarbon (HC) compounds, oxides of nitrogen (NOx), carbon monoxide (CO) and particulates matter (PM) on human health and environment are well documented, and scientific evidence continues to grow and becomes increasingly compelling. Greenhouse gas (GHG) emissions from motor vehicles present longer term problems and in most countries, over 90% of global warming potential of the direct-acting GHGs from the transportation sector come from carbon dioxide (Botter *et al*, 2002; Esteves and Barbosa, 2007).

The transport sector is expected to be responsible for about 75% of carbon compounds emission by the year 2020 and therefore reducing transport sector carbon compounds emissions will be crucial for stabilizing atmospheric concentrations of greenhouse gases (US EPA, 2002). It is incumbent upon government to reduce these harmful impacts of motor vehicle use because vehicles are long-lived and the world fleet continues to grow. The exhaust emission control programs to be introduced, if they are to be truly effective, must address the in-use fleet and the next generation fleet, and this largely depends on today's research. Providing information about highly emitting category of vehicles and the major contributing emission factors are very critical in developing sustainable vehicle emission programs (US EPA, 2002).

1.1 Background

Motor vehicles produce more air pollution than any other single human activity (WRI, 1997). Nearly 50 % of global CO, HCs and NO_x emissions from fossil fuel combustion come from petrol and diesel engines. In city centres and congested streets, traffic can be responsible for 80-90 % of these pollutants and thus their saturation is particularly severe in cities in developing countries (Whitelegg and Haq, 2003). Vehicle emissions mainly result from fuel combustion or evaporation. The most common types of transport fuels are gasoline (in leaded or unleaded form) for light duty vehicles and diesel for heavy duty vehicles such as buses and trucks. Also, commercial fuels used in light duty vehicles include alcohols (ethanol and methanol), gasoline alcohol mixture, compressed natural gas (CNG), and liquefied petroleum gases (LPG) for heavy-duty vehicles. Emissions from motor vehicles are from the exhaust, engine crank case and fuel system.

Carbon (iv) oxide (CO_2) and water vapour (H_2O), the main products of combustion are emitted in vehicle exhaust (Delaney *et al*, 2000). The other major pollutants emitted from gasoline-fuelled vehicles are CO, HCs, NO_x and lead (Pb) (for leaded fuel) (Whitlegg and Haq, 2003). The major operating variables that affect spark-ignition engine performance, efficiency, and emissions at any given load and speed are: compression pressure, spark timing, fuel/air or air/fuel ratio relative to the stoichiometric ratio, ignition voltage, specific emission and emission index (Heywood, 1998; Stone; 1999). The engine performance parameters of interest are power, torque, specific fuel consumption and combustion efficiency.

In recent years, emphasis on measurement of vehicle emissions has shifted from laboratory testing towards the analysis of 'real world' emissions. The 'real world' vehicle fleet is composed of new and ageing vehicles with widely varying maintenance and operational conditions (Wenzel *et al*, 2000). Since the ultimate goal of vehicles' emissions control devices and programmes is to improve ambient air quality, analyses of programmes and technology effectiveness should focus as much as possible on real world emissions reduction (Singer and Harley, 2000; Pokharel, *et a.l.,* 2000). A lot of research has been done on the development of efficient IC engines in order to reduce on emissions. However, the maintenance of these engines to ensure that they operate at the designed optimum conditions has been a challenge. The degree to which owners maintain their vehicles by providing tune-ups and servicing according to manufacturer's schedules can affect the likelihood of engine performance and emissions control system failure and therefore, tailpipe emissions (Wenzel, 1999). The high concentrations of vehicle exhaust emissions is more closely related to its maintenance than to engine designed parameters and therefore, relating engine performance parameters and emissions will help to explain this phenomena (Heywood, 1998; Wenzel, 1999).

Pollutant formation which is related to engine performance is rather difficult to predict theoretically or by numerical simulation. This is primarily because these phenomena are governed by the detailed spatial and temporal distribution of the mixture composition, temperature and pressure inside the combustion chamber (Yamamoto *et al*, 2002). Control-oriented models, therefore, often rely on the results of experiments which are summarized in appropriate maps. Once these maps are available, varying state variables (engine speed, manifold pressure, exhausts gas re-circulation rate, etc.) and accessible control variables (injection pressure and ignition timing) are used to derive the corresponding engine-output and emission values (Arsie *et al*, 1998; Atkinson *et al*, 1998; Yamamoto *et al*, 2002).

1.2 Statement of the Problem

Several factors account for the variability in emissions from different vehicles and the amount of environmental damage caused. Studies have shown that poor vehicles maintenance culture, high proportion of old vehicles, vehicles driving pattern and poor transport policies are major contributing factors to high vehicle emission levels (Kojima and Lovei, 2001; Mulaku and Kariuki, 2001; Zachariadis *et al.*, 2001; Whitelegg and Haq, 2003; Choo *et al*, 2007). These factors are normally considered when developing emission prediction models (Zachariadis *et al.*, 2001; Bin, 2003; Choo *et al*, 2007). The above factors are very common in the transport sector in Kenya. There is no policy in place to govern repair and service industry with the majority of vehicles serviced in the informal sector (*Jua Kali*). The quality of services in this sector is poor and this is likely to affect engine performance and emission levels (Langat *et al.*, 2004). Vehicle retirement is not a requirement at the moment in Kenya and most of the transport policies in operation were developed more than a decade ago (US EPA, 2002).

Previous studies in Nairobi showed particulate matter (PM) mean values of 239µg/m^3 and 396µg/m^3 for PM_{10} and $PM_{2.5}$ respectively (Van Vliet and Kinney, 2006). These concentrations are higher than the World Health Organization (WHO) limits of 150µg/m^3 and 65µg/m^3 for PM_{10} and $PM_{2.5}$ respectively (Maina, 2004; Van Vliet and Kinney, 2006). The high concentration of NOx during peak traffic hours is an indicator that vehicles are the major source of the pollutants. However, none of these studies linked pollutant levels with vehicle characteristics and engine operating conditions. Since high concentrations of a vehicle's exhaust emission is related to its maintenance rather than engine designed parameters, relating engine performance parameters and emissions will help to explain this phenomenon (Heywood, 1998; Wenzel, 1999). This research therefore, sought to determine vehicles' exhaust emission levels, establish the major contributing factors based on their category, usage and engine operating parameters, simulate operating parameters on an engine test bed, and develop performance and exhaust emission prediction models.

1.3 Research Objectives

1.3.1 Main Objective

The main objective of the study was to determine performance and emission characteristics of inspected vehicles in Nairobi City based on engine operating parameters and to develop performance and emission prediction models.

1.3.2 Specific Objectives

The specific objectives for the study were:

i. To determine the current exhaust emission levels of CO, HC and CO_2 in public service vehicles operating in Nairobi city, Kenya.

ii. To analyze engine performance and emission characteristics based on engine operating parameters.

iii. To develop performance and exhaust emission prediction models.

1.4 Research Questions

The following research questions were used in the study.

i. What are the current emission levels of public service vehicles operating in Nairobi City?

ii. To what extent do engine operating parameters affect performance and emission levels?

iii. To what extent do the developed models predict performance and exhaust emissions from engine operating parameters?

1.5 Research Justification

The growth in motor vehicle emissions is of great concern to governments as they strive to protect public health and welfare (UNEP, 2002). The harmful effects of conventional pollutants from motor vehicles on human health and ecosystems continue to grow and scientific evidence is required (Gwilliam *et al*, 2004). Greenhouse gas emissions from motor vehicles present longer-term problems, potentially with severe health, environmental and economic consequences. The transportation sector is responsible for about 26% of global carbon compounds emissions and this is projected to increase to 75% between 1997 and 2020 if no measures are put in place to control them (US EPA, 2002). Energy consumption is related to vehicular emissions and hence reducing energy consumption through well maintained vehicles will benefit both the owners and the economy. The development of performance and emission models will not only help to explain performance and emission characteristics of the vehicles, but can also be used to identify vehicles that require specific repairs or maintenance. This study, therefore, contributes towards understanding the relationship between vehicles' performance and emission of pollutant substances (CO, HC and CO_2) from vehicles and the various interactions that exist between the various factors that result in this problem.

1.6 Scope and Limitations

1.6.1 Scope

This study considered exhaust emissions levels of public service vehicles operating in Nairobi City. It considered specifically sources of exhaust emissions from various petrol vehicle categories, usage and engine operating parameters. The engine operating parameters collected from the field were used for simulation in an engine test bed to explain performance and emission characteristics of the vehicles considered in the study. The pollutants under consideration were carbon monoxide (CO), carbon dioxide (CO_2) and hydrocarbons (HC).

Pollutant emissions from industrial activities and other sources were not included in the study. Likewise, agricultural, construction and mining equipments (vehicles) were not considered as the policy, tools and institutional framework regulating their use are totally different from normal vehicles. The study also, did not include evaporative emissions.

1.6.2 Limitations

The results presented in this thesis were case dependent and must be tested and validated before being adopted for use in other petrol vehicles. Because in-use vehicle emissions are very variable, findings from this research are not directly comparable. This study developed spark ignition engine performance models through simulation and comparison of critical values found in the fleet of vehicles in the field. Theoretically the models cannot be used to forecast performance beyond the domain of variables used in developing the models. The effects of emission control technology deterioration on emission levels in old vehicles were not considered. Such limitations could introduce obvious uncertainties in the use of the model to make prediction for other fleets.

1.7 Outline of the Thesis

The introduction to the research is given in Chapter One, where background, problem statement, objectives, justification, scope and limitation of the study are discussed. Chapter Two reviews literature on vehicular emissions and emission factors. Previous related studies and methods used are reviewed and existing gaps are identified. Also reviewed are pollutants formation and abatement systems in SI engines and how they are related to engine performance. Different engine operating parameters that affect engine performance are reviewed. The equations governing engine performance are discussed. Internal combustion engine models with emphasis on performance and emission prediction and different model approaches are also reviewed. Chapter Three presents the methods used to determine engine operating parameters and sampling exhaust gases. Detailed procedures for simulating parameters that affect performance and emissions are given. Different data analysis techniques are also presented. The results are presented in Chapter Four, where the outcomes of exhaust emission levels, engine performance characteristics and the underlining factors are discussed. Individual performance and emission models and superimposed models are also discussed. The conclusions and recommendations are given in Chapter Five, while references and appendices are included at the end of the thesis.

CHAPTER TWO

LITERATURE REVIEW

2.1 Air Pollution

Air pollution is the presence in the atmosphere of man-made or natural substances in quantities likely to harm human, plant or animal life; to damage man-made materials and structures; to bring about changes in weather and climate, or to interfere with the enjoyment of life or property (Kojima and Lovei 2001; Gwilliam *et al,* 2004). The amounts of pollutants released to the atmosphere by fixed or mobile man-made sources is generally associated with the level of economic activity. Meteorological and topographical conditions affect dispersion and transportation of the pollutants, which can result in ambient concentrations that may harm people, structures and environment. In general, the effects on people are most intense in large urban centres with significant emission sources, unfavourable dispersion characteristics and high population densities. Although urban air quality in industrial countries has been controlled to some extent during the past two decades, in many developing countries it is worsening and becoming a major threat to the health and welfare of people and the environment (UNEP, 1999; WHO, 2002).

2.1.1 Motor Vehicle Emissions and Emission Factors

Vehicle emissions which occur near ground level and in densely populated areas, cause much greater human exposure to harmful pollutants in the immediate locality than do emissions from sources such as power plants that are situated at elevated levels and farther away from densely populated centres. In addition, vehicle exhaust particles being small and numerous, can be expected to have considerable health impacts. Pollution abatement in the transport sector is, therefore, becoming a more important factor in urban air quality management strategies (Kojima and Lovei, 2001; Gwilliam *et al*, 2004).

Real-world vehicle emissions are highly variable. Several factors account for the variability in emissions in different vehicles and the amount of environment damage caused (Wenzel and Ross, 1998; Wenzel *et al*, 2000,). However, due to relatively higher average temperatures, poor fuel quality, poor vehicle maintenance culture and high proportion of old vehicles, the level of emissions from mobile sources are unusually high (US EPA 1996; Kojima and Lovei, 2001; Mulaku and Kariuki, 2001; Whitelegg and Hag, 2003; Bin, 2003; Choo *et al*, 2007).

2.1.2 Inherent Variability in Vehicle Emissions

Emissions variability from one vehicle to the other spans several orders of magnitude, while the emissions of most vehicles will vary substantially with environmental and driving conditions. Emissions of some vehicles are unrepeatable: different emissions occur from one test to another, even when test conditions are carefully controlled (Heywood, 1998; Wenzel, 1999). Vehicle emission variability is a consequence of the way emissions are generated and how they are controlled. Exhaust emissions are formed in the engine as a result of unburned fuel, HC, and partially burned fuel, CO, and from undesirable side reactions, NOx.

Emissions control systems are designed to reduce pollutant formation in the engine-output to less harmful products in the catalytic converter. When functioning properly, modern vehicle-emission controls reduce tailpipe emissions levels to five percent or less of those observed from pre-control vehicles produced in the late 1960's (MECA, 1998; Bin, 2003; Choo *et al*, 2007). However, if the engine or the emissions control system fails to operate as designed, exhaust emissions may rise. There are numerous factors affecting the variability in emissions across different vehicles (Guzzella and Onder, 2004). Some of these factors are vehicle technology, age, mileage accumulation, models, maintenance and tampering. The factors are discussed in detail below.

(i) Vehicle technology

Emissions control techniques that have been incorporated into vehicles include the use of exhaust gas recirculation to reduce NOx formation in the engine, the addition of catalytic converters for exhaust gas treatment, the replacement of carburetors with throttle-body and port fuel injection, and computer control of air-fuel mixing and spark timing. In most cases, these and other vehicle-emission control improvements have been introduced to the entire new fleet over just a few model years. Real-world emissions are sensitive to vehicle technology independent of vehicle's age (Tounsi *et al*, 2003; Kim *et al*, 2003; Bin, 2003; Choo *et al*, 2007).

(ii) Vehicle age and mileage accumulation

As vehicles age and accumulate mileage, their emissions tend to increase. This is both a function of normal degradation of emissions controls of properly functioning vehicles, resulting in moderate emissions increases, and malfunction or outright failure of emissions controls on some vehicles, possibly resulting in very large increases in emissions, particularly CO and HC (Wenzel, 1999; Washburn *et al*, 2001; Bin, 2003). However, exhaust emission as a function of age and mileage accumulation can vary depending on vehicle maintenance culture. A more accurate way of determining the influence of the two variables in exhaust emission is by considering vehicle's annual usage. This is obtained by dividing total mileage accumulated by vehicle's age (US EPA, 2002, BAQ, 2002).

(iii) Vehicle models

Some vehicle models are simply designed and manufactured better than others. Some vehicle models and engine families are observed to have very low average emissions, while others exhibit very high rates of emission's control failure (Wenzel and Ross, 1998; Wenzel, 1999; Bureau of Automotive Repair, 2003). The design of a particular emission's control system affects both the initial effectiveness and the lifetime durability of the system, which in turn contributes to a model- specific emission's rate (Fomunung, 2000).

(iv) Maintenance and tampering

The degree to which owners maintain their vehicles by providing tune-ups and servicing according to manufacturer's schedules can affect the likelihood of engine or emissions control system failure and therefore tailpipe emissions. Outright tampering with vehicles, such as removing fuel tank inlet restrictors to permit fueling with leaded fuel that will degrade the catalytic converter or tuning engines to improve performance, can have a large impact on emissions (Wenzel *et al*, 2000, Bureau of Automotive Repair, 2002; Bin, 2003). Early inspection and maintenance (I/M) programs relied on visual inspection to discourage tampering. The advent of sophisticated on-board computers and sensors have greatly reduced the incentive to improve vehicle performance through tampering. In fact, tampering with the sophisticated electronics installed on today's vehicles will likely reduce performance as well as increase emissions. Requirements for extended manufacturer warranties have led to vehicle designs that are less sensitive to maintenance, at least within the warranty period. Nonetheless, there is evidence that maintenance can still affect real-world emissions from new vehicles, at least on some models (Wenzel 1999; Bin, 2003; Choo *et al*, 2007). Improper maintenance or repair can also lead to higher emissions (Bureau of Automotive Repair, 2002; Bureau of Automotive Repair, 2003).

The cumulative effects of hard driving or 'misuse" of a vehicle can also increase emissions. For example, prolonged high power driving, such as repeated towing of a trailer up mountain grades, leading to high engine temperatures which can cause premature damage to a catalytic converter, resulting in dramatic increase in emissions (Wenzel and Ross, 1998;

Wenzel, 1999; Washburn *et al*, 2001). There are many emissions control components that can malfunction or fail. Some of these malfunctions are interpreted by the vehicle in build systems; for instance, the onboard computer of a vehicle with a failed oxygen sensor may command a constant fuel enrichment, which can eventually lead to catalyst failure. Different component malfunctions result in different emissions consequences. In general, malfunctioning vehicles with high CO emissions tend also to have high HC emissions, while vehicles with high NOx emissions tend to have relatively low CO and HC emissions (Wenzel *et al,* 2000; Bureau of Automotive Repair, 2002).

2.1.3 Previous Studies

Different studies have been done in the field of motor vehicle emissions in the different regions of the world, especially to establish the level of air pollution from the operation of motor vehicles and the general urban air quality as a whole. Four of such studies which have relevance to this study due to similarities in regional climatic conditions, and socio-economic circumstances are: The impact of automobile emissions on the level of platinum and lead in Accra, Ghana conducted in 2001 by Kylander *et al*., (2003); The evaluation of evaporative emissions from gasoline powered motor vehicles under South African conditions, conducted in 2003 by Van der Westhuisena *et al*., (2004); The vehicle activity study in Nairobi, Kenya, conducted in March 2001 by the US EPA, CE-CERT, and GSSR; Small pilot air quality sample comparing urban $PM_{2.5}$ and black carbon levels in Nairobi, Kenya (US EPA 2002; Van Vliet and Kinney, 2006).

(i) Accra study

This study sought to estimate the release of Petroleum Groups Elements (PGE), Platinum (Pt) and Lead (Pb) from catalytic converters during vehicle operation in Accra. It focused on Pb and Pt levels in collected road dust and soils, and also included an inventory of a number of catalytic converters in Accra. To get a better idea of PGE emissions in Accra, data on vehicle fleet including manufacturers' home location or known years of production of a particular vehicle and the possible age range of the vehicle fleet was gathered. Sampling of sites for road dust collection was done in five locations in and around Accra. The sites were selected based on their traffic intensity and ease of sampling.

Nearly all vehicles in Ghana come from Europe through private sales or commercial operations. Platinum and lead concentrations in road dust increased with increasing traffic density. The highest traffic density in the study was found at Kotoka International Airport area with approximately 5,000 vehicles a day and an average Pt value of 55.0 μ g/g (Kylander *et al*., 2003). Although this study was limited to Pt and Pb, as a pioneering study in the field of pollutant emissions in Ghana, it was helpful to the analysis of this study since residence of both Accra and Nairobi cities share similar socio-economic and geographic conditions as well as similar fleet composition.

(ii) South African study

The main objective of this study was to quantify the amount of evaporative emissions released by gasoline powered motor vehicles subjected to a variety of conditions typical of South Africa (Van der Westhuisena *et al*., 2004). This stems from the fact that most previous research studies on evaporative emissions of in-service vehicles have been performed in US cities (Lyons *et al*, 2000; Delaney *et al*., 2000; Van der Westhuisena *et al*., 2004), where legislation is one of the strictest in the world. South Africa also has no legislation controlling vehicular emissions at the moment. This coupled with other factors such as South Africa's fleet composition difference in terms of emission control devices compared with other developed countries, the high vapour pressure fuel in South Africa compared with international levels (Van der Westhuisena *et al*., 1998), and also the general low turnover rate of new vehicles compared with international standards, imply that the level of HC evaporative emissions from South African vehicles is very high, hence the need for the study.

The study focused on total evaporative emissions and running losses. Because ambient temperatures are generally very high during the summer period in South Africa and fuel temperature inside the fuel tank may rise above 45ºC during driving, this therefore, called for a specific test (road tests) required to simulate fuel loss in an average South African gasoline powered motor vehicle, operated under various conditions. The test therefore required higher temperatures for longer periods of time than is prescribed in the standardized test procedures. This was achieved before the normal laboratory emission test designed by US EPA (US EPA, 1994) was conducted. The distances covered for the road test were Cape Town to Graaff-Reinette, Durbanville to Stellenbosch, Durbanville to Saldanha Bay, and urban driving conditions, of distances ranging from 30 km to 120 km.

Road tests under both urban and highway conditions in South Africa indicated that fuel temperatures can far exceed the maximum fuel temperature of 30 ºC specified for the prescribed evaporative emissions test. The mixed urban and highway driving tests indicated that temperatures can reach as high as 47 ºC, while the open road testing indicated temperatures reaching 52 ºC. Evaporative diurnal emissions of vehicles without evaporative emission control systems increased with increasing temperature. South African vehicles without evaporation control emitted ten times the amount allowed by the US EPA. The vehicle fitted with evaporation control was well within the US EPA limits. An extended-time diurnal test at higher temperatures was conducted. The results indicated that the amount of unburned HC emitted during the average life span of a South African gasoline powered vehicle without evaporative emission control system and driven volatility on vehicle evaporative emissions under conditions typical of South Africa, was about 291 litres (Kylander *et al.*, 2003).

To reduce fuel consumption and improve air quality, gasoline powered motor vehicles should be equipped with evaporative control equipment. Regulations to reduce the allowable limit of unburned HCs emitted from the entire vehicle should also be implemented. Fuel circulation test showed that a very small amount of fuel is emitted, which proves that evaporative emissions depend mainly on temperature of the fuel (Kylander *et al.*, 2003). The major limitation of this study lay in the fact that its scope was narrowed to just evaporative emissions (diurnal and running losses), which makes it difficult to draw experiences from, in the estimation of emission levels in the Nairobi city. South Africa's geographical and topographic conditions are also highly different from those pertaining in the Nairobi city.

(iii) Nairobi study

The aim of this study was to collect important vehicle related data to support development of an accurate estimate of on-road vehicular emissions for Nairobi. The study considered emissions from on-road vehicles to vary considerably depending upon three factors namely; vehicle type; driving behaviour; and local geographic and climatic conditions (US EPA; 2002). Based on these factors, data on on-road driving patterns, vehicles distribution, vehicle start-up patterns and vehicle counts were collected using Global Positioning System (GPS), digital videos, Vehicle Occupancy Characteristics Enumerator (VOCE), and parking lot surveillance, to help define vehicle types and driving behaviour in Nairobi. The collected data was then formatted and put into the international vehicle emission (IVE) model for estimating the criteria, toxic, and global warming pollutants from on-road vehicles (US EPA, 2002).

This study came up with information on the overall fleet activity distribution of Nairobi city, and vehicle technologies used in the IVE model, under classifications such as vehicle type, engine sizes, type of fuel consumed, vehicle make and model, registration year, model year, odometer reading, availability of catalyst, air/fuel control system's capability, frequency of maintenance, weight, age, exhaust control capability, and evaporative control capability. The study was, however, not exhaustive on how these findings could be used to estimate the total level of pollutant emissions from motor vehicle sources so as to ascertain the true level of vehicle emissions contribution to the air quality of the city of Nairobi. Nonetheless, as a pioneering study, its findings were important for analyzing the operating characteristics of the vehicles (US EPA, 2002).

A pilot air quality study in and around Nairobi and Ruiru by a research team from Columbia University's Mailman School of Public Health confirmed that there were very high levels of particulate air pollution, specially levels of $PM_{2.5}$ and black carbon in metropolitan Nairobi, especially on and near the roadways. The findings clearly indicated that transportation is the main contributing factor to poor air quality in Nairobi (Van Vliet and Kinney, 2006). Although the study was very small in scope and findings may not be considered statistically significant, the results indicated that there is critical need for consistent and large-scale air quality research. This research, therefore, sought to employ most of the factors taken into consideration in the two researches in Nairobi, in coming up with the findings towards estimating the level of motor vehicle emissions in Nairobi city.

2.1.4 Pollutant Formation in Internal Combustion Engines

Pollutant formation in internal combustion engines is rather difficult to predict theoretically or by numerical simulation. This is primarily because these phenomena are governed by the detailed spatial and temporal distribution of the mixture composition, temperature and pressure inside the combustion chamber (Yamamoto *et al,* 2002). Control-oriented models, therefore, often rely on the results of experiments which are summarized in appropriate maps. Once these maps are available, slowly varying state variables (engine speed, manifold pressure, EGR rate, etc.) and accessible control variables (injection pressure and ignition timing) are used to derive the corresponding engine-out pollution values (Arsie *et al*, 1998; Atkinson, 1998; Yamamoto *et al*, 2002).

All these approaches rely on the fundamental assumption that the pollution formation process depends, in a deterministic way, on the control inputs and on the thermodynamic boundary conditions mentioned above. Of course, this determinism is not entirely true in real situations, such that only average estimations of engine-out pollutant, concentration levels are possible. Moreover, aging and other, not easily modeled effects, cause large deviations in pollutant formation such that, in general, prediction errors are quite large. Control-oriented engine-out pollution models are therefore, often used as a means to predict the relative impact of a specific controller structure or algorithm on pollutant emission. In this section, the most important pollutant formation mechanisms are discussed using qualitative arguments. SI engines are analyzed starting with a discussion of the air/fuel ratio needed for a Stoichiometric combustion. As an example, the last part of this section shows a quantitative COM of the NO_X formation in homogeneous-charge SI engines (Guzzella and Onder, 2004).

2.1.5 Stoichiometric Combustion

While the actual reactions are much more complicated and involve a large number of intermediate species, the combustion of a hydrocarbon $H_{a'}C_{b'}$ occurs according to the following overall chemical reaction:

$$1 \text{ mol } H_{a'}C_{b'} + c' \text{ mol air} \rightarrow d' \text{ mol } H_2O + e' \text{ mol } CO_2 + f' \text{ mol } N_2 \qquad (2.1)$$

Assuming the ambient air to consist of 79 % N_2 and 21 % O_2 by volume, the following relations can be derived for a stoichiometric combustion

$$d' = \frac{a'}{2}, \quad e' = b', \quad f = \frac{137\left(1+\frac{y}{4}\right)}{12+y} kg \qquad (2.2)$$

where; $y = a/b$ is the fuel's hydrogen-to-carbon ratio by mass.

Accordingly, the stoichiometric mass of air needed to burn 1 kg of a hydrocarbon fuel $H_{a'}$ $C_{b'}$ is given by the expression

$$m_{air,stoich} = \frac{137(1+y/4)}{12+y} kg \qquad (2.3)$$

Figure 2.1 shows the form of equation (2.3) and the specific values for three important fuels. The limit values are 11.4 kg of air for pure carbon and 34.3 kg of air for pure hydrogen.

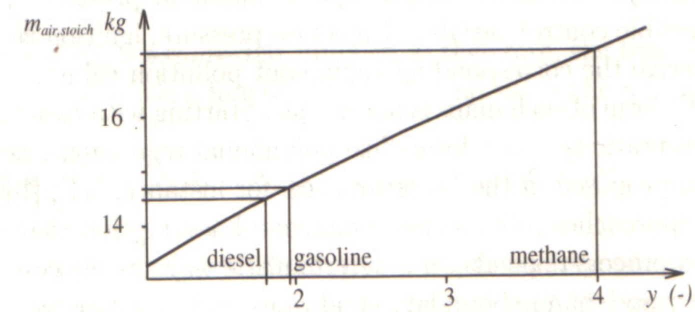

Fig. 2.1: Stoichiometric mass of air required to burn 1 kg of hydrocarbon fuel $H_{a'}$ $C_{b'}$ (Source: Guzzella and Onder, 2004)

The complete combustion assumed in equation (2.1) is not attained even in perfect thermodynamic equilibrium condition. In practice, traces (a few hundred to several tens of thousands of ppm) of pollutant species are always emitted. The main pollutants are:

(i) Products of an incomplete oxidation: carbon monoxide, CO, (with a concentration of typically one order of magnitude larger than those of any other pollutant in the exhaust gases) and hydrocarbon (either unburnt fuel or intermediate species) which, depending on the fuel, can occur as a simple or complex molecular species including cancer-causing components;

(ii) Products of an unwanted oxidation: mostly nitric oxide, NO, but also some NO_2.

In diffusion-flame combustion systems (stratified charge GDI engines), a substantial number of particulates (sized from a few tens of nanometers to several hundred microns in size) are generated as well (Heywood, 1998; Guzzella and Onder, 2004).

2.1.6 Pollutant Formation in SI Engines

Figure 2.2 shows typical pollutant concentrations in the exhaust gas of a port-injected SI engine as a function of the normalized air/fuel ratio. Qualitative explanations for the shape of the curves shown in this figure are given below.

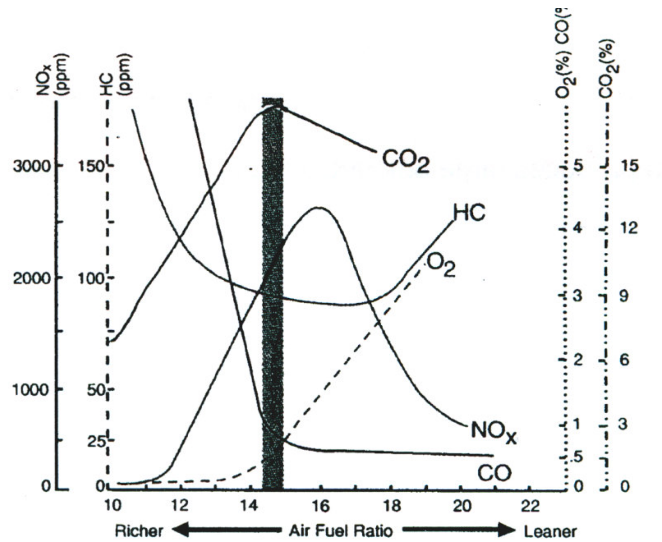

Fig. 2.2: Typical engine-out pollutant emission of a port-injected SI engine (Source: Guzzella and Onder, 2004)

(i) Hydrocarbon and carbon monoxide

In rich conditions, the lack of oxygen produces very high HC and CO concentrations because there is no oxidation taking place. When the engine is operated under extremely lean conditions, misfires start to occur which, of course, cause increased HC emission. The misfires also reduce in-cylinder and exhaust temperatures, which lead to smaller post-combustion oxidation. CO does not increase due to misfires (when the mixture fails to ignite, no CO can be formed). The reason that at intermediate air/fuel ratios (14.7:1) the HC and CO concentrations do not reach the (almost) zero equilibrium levels can be explained as follows:

(i) Due to the cylinder's charge quick cooling during the expansion stroke, the necessary time required to reach chemical equilibrium is not available (high CO and HC concentrations are "frozen" shortly after their formation during combustion).

(ii) During the compression stroke, the mixture is forced into several crevices present in the cylinder, and part of the mixture is absorbed into the oil film covering the cylinder walls (Fig. 2.3). As soon as the pressure decreases during the expansion stroke (and after combustion has terminated), these HC molecules are released into the cylinder. Since cylinder charge temperatures have already fallen substantially, a non-negligible portion of these HC molecules escape post-flame oxidation.

(iii) When the flame front reaches the proximity of the cylinder walls (0.1 mm distance), large heat losses to the wall quench the flame before all the mixture has been oxidized.

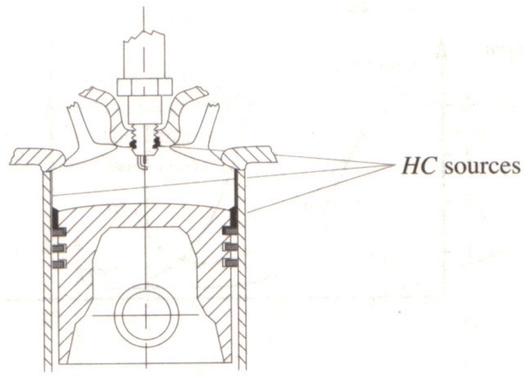

Fig. 2.3 Crevices and other sources of engine HC emissions (Source: Guzzella and Onder, 2004)

Since a substantial part of the HC and CO is always oxidized in the post-flame phase inside the cylinder, and even during blow-down in the exhaust manifold, all effects that influence the conditions for this phenomenon (oxygen, temperature and time), also affect the HC and CO emission engine-out. The pollutant formation in direct-injection SI engines in homogeneous mode is essentially the same as in port-injection engines. In stratified mode, however, substantial differences appear, for instance, the late injection may eliminate the crevice losses.

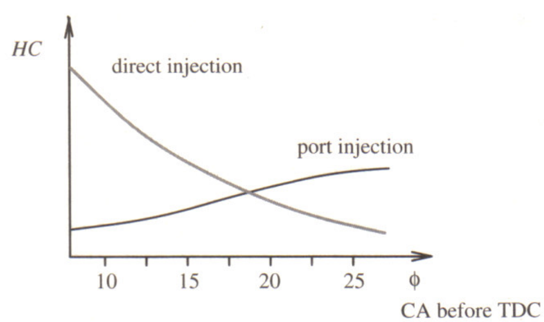

Fig. 2.4: Hydrocarbon emission of a port-injection SI engine (source; Guzzella and Onder, 2004)

Figure 2.4 shows that this does not necessarily lead to reduced pollutant levels. In this figure, HC emissions are plotted as a function of ignition timing. While the port-injected engine has increasing HC levels due to later ignition (the thermodynamic efficiency decreases, hence the burnt gases are hotter, which improves post-combustion oxidation), the direct-injection engine displays the inverse behaviuor (Ferguson, 1986). This shows that in direct-injection engines the ignition point may not be shifted without an appropriate change in injection parameters (Sun *et al*, 1999). Note that in modern engines "blow-by" gases, i.e., that part of the mixture that enters the crank shaft compartment due to leakages of the piston rings, are collected and fed back to the combustion process.

(ii) Carbon dioxide

The target fixed at Kyoto Protocol was a 5.2 % reduction of emissions in all sectors of the economy compared to 1990 levels by 2008-2012 by all member countries. Relative carbon dioxide emissions from transport have risen rapidly in recent years in developed world, from 21 % of the total in 1990 to 28 % in 2004, but currently there are no standards for CO_2 emission limits for pollution from vehicles. EU transport emissions of CO_2 currently account for about 3.5 % of global CO_2 emissions. Any action taken to reduce CO_2 emissions will have to involve curbing transport emissions. Passenger cars account for about half the transport-related CO_2 emissions in the European Union. This figure is higher in developing countries because of high proportion of old and poorly maintained vehicles (EU, 2007).

Carbon dioxide is the most ubiquitous of all the greenhouse gases and developed nations have started addressing the production of this greenhouse gas although currently it is considered as non poisonous gas and has no limits. For example, the purpose of Directive 1999/94/EC of the European Parliament and the Council of 13th December 1999 relating to the availability of consumer information on fuel economy and CO_2 emissions in respect of the marketing of new passenger cars is to ensure that information relating to the fuel economy and CO_2 emissions of new passenger cars offered for sale or lease in the Community is made available to consumers in order to enable consumers to make an informed choice (EU, 2007).

The CO_2 emission limits generated by vehicles are nowadays subject to a voluntary agreement. The ultimate EU target with voluntary agreements is to reach an average CO_2 emission (as measured according to Commission Directive 93/116/EC) of 120 g/km for all new passenger cars by 2012. However, as it becomes increasingly clear that the agreement will not deliver (having achieved only 160 g/km in 2005, from 186 g/km in 1995), lawmakers have started considering regulations. In late 2005, the European Parliament passed a resolution in support of mandatory CO_2 emission standards to replace current voluntary commitments by the auto manufacturers and labeling (EU, 2007).

In late 2006, in response to a new report, by the European Federation for Transport and Environment documenting lack of progress on the voluntary targets, the European Commission announced that it was working on a proposal for legally-binding limits of CO_2 emissions from cars. According to the mentioned European Federation for Transport and Environment study, the Fiat model is the best performer in Europe. On 7th February 2007 the European Commission published its key draft proposal of 2007 EC legislation, to limit average CO_2 emissions from the European fleet of cars to 120 g CO_2/km. Some people interpreted this as meaning that all manufacturers would have to average 120 g for their fleet, but this was not the case. There has been need for a longer-term target that doubles fuel efficiency of new cars over the next decade to 80 g/km by 2020. It reported new-car emissions from European producers slipped to 160 g/km on average in 2006 (reduced only 0.2 percent), still way off a voluntary goal of 140 g/km by 2008 (EU, 2007).

2.1.7 Strategies Involved in Reducing Vehicular Emissions

Since 1980, the permitted levels for the concentrations of carbon monoxide CO, hydrocarbons HC, and NO_X (NO, NO_2) have been reduced significantly (Guzzella and Onder, 2004). A continuation of this trend is to be expected as legislations are imposed in Europe, USA, Japan and other countries. In contrast to the European regulations, which limit the total hydrocarbon emissions, the US regulations consider only the non-methane hydrocarbons. This choice reflects the fact that the methane emissions by passenger cars are very low compared to the agricultural emissions. However, the US standards regulate formaldehyde (HCHO) levels, which are not explicitly limited in the European Union. Common to all of these emission limits is that they may not be exceeded during a predefined driving cycle. This includes a warm-up phase, transients and idle periods with defined boundary conditions such as ambient temperatures (Guzzella and Onder, 2004).

Today's stringent emission regulations and those in the future will be difficult to meet. Only homogeneous charge SI engines operated with a stoichiometric air/fuel ratio can easily meet these limits by using three-way catalytic converters. All other engines (Diesel, direct-injection SI engines) must be combined with more complex exhaust gas after treatment systems such as lean NO_X traps, selective catalytic reduction converters and particulate filters. In this section, three-way catalytic converters and selective catalytic reduction converters are analyzed from a control engineering point of view (Tounsi *et al*, 2003; Kim *et al*; 2003).

The most common pollution abatement system for SI port-injection engines is the three-way catalytic converter (TWC). It derives its name from its ability to simultaneously reduce NO_X and oxidize CO and HC. State-of-the-art systems are capable of removing more than 98% of the pollutants. This can only be achieved by operating the engine within very narrow air/fuel ratio limits. A catalyst is a substance that promotes chemical reactions without being dissipated (Guzzella and Onder, 2004). An ideal TWC promotes the following three primary chemical reactions;

$$NO + CO \rightarrow N_2 + CO_2 \tag{2.4}$$

$$CO + \frac{1}{2}O_2 \rightarrow CO_2 \tag{2.5}$$

$$H_{a'}C_{b'} + \left(b' + \frac{a'}{4}\right)O_2 \rightarrow b'CO_2 + \frac{a'}{2}H_2O \tag{2.6}$$

In reality, the system of reactions is far more complex (Jobson *et al*, 1996). Several dozen reactions are involved and other species, apart from those mentioned are produced. For example, the concentration of N_2O is often increased by the reactions taking place inside the TWC.

Generally, the reactants are first absorbed on the catalytic surface. Catalytic substances, such as Platinum *(*Pt), Rhodium (Rh*)* or Palladium (Pd*),* weaken the bonds of the absorbed species and, thus, allow for the formation of the desired products, according to the law of minimizing the (chemical) potential energy. The products are then deabsorbed and thus released to the gas phase. Reactions (2.4) to (2.6) clearly show that optimal conversion rates can only be achieved if exactly as much oxygen is available as is used for the oxidant of CO and HC. It would be favorable if (2.4) were dominant. In the presence of excess oxygen, (A/F ratios λ>1) the reaction (2.5) always occurs in parallel to the other reactions and, therefore, under lean conditions usually inhibits the reduction of NO. With a shortage of oxygen at λ< 1, all NO is converted, but the removal of HC and CO is incomplete. Therefore, the TWC can be operated only within the very narrow band of air/fuel ratios around λ= 1 as explained by Guzzella and Onder, 2004.

Since air/fuel ratio sensors show a significant cross-sensitivity to hydrogen, this effect has to be taken into account for a correct interpretation of the downstream air/fuel ratio sensor signal. In fact, for rich mixture, the air/fuel can be employed as a hydrogen sensor (Auckenthaler *et al*, 2004). Especially during transients, when the wall-wetting dynamics and other disturbances render a complete compensation of the varying air mass flow difficult, the air-to-fuel ratio cannot always be kept within the narrow band that is necessary for the TWC to work efficiently.

Therefore, the catalyst must be able to cope with temporary excursions to the lean or to the rich side. This is achieved by incorporating a reservoir for oxygen, which lies beneath the catalytically active surface and which consists of special materials such as cerium. Excess oxygen can then temporarily be stored in these materials as well as on the catalytic surface. The oxygen storage within cerium can be described chemically by the following reaction (Guzzella and Onder, 2004).

$$Ce_2\,O_3 + O^* \leftrightarrow 2\,Ce\,O_2 \tag{2.7}$$

The absorbed oxygen O^* combines with $Ce_2\,O_3$ to form a new grid structure. The ratio of the catalytically active surfaces of the noble metal (Pt, Pd, Rh) and the cerium considerably influence the water-gas shift reaction

$$CO + H_2O \rightarrow CO_2 + H_2 \tag{2.8}$$

This results in a changing H_2/CO ratio during the lifetime of the TWC, since the catalytically active surface decreases with aging. Therefore, new TWCs promote the water-gas shift reaction significantly more than aged ones, as has been described above. The TWC consists of a carrier (usually ceramic or metallic substrate) and coating material. First, the washcoat (aluminum oxide) is applied to enlarge the surface of the TWC. Then the additional oxygen storage material (cerium oxide) is brought on the washcoat, and, finally, stabilizers and noble metals are placed on top. The better the distribution of the various components, the better the activity of the TWC.

2.2 Operating Variables that affect SI Engine Performance and Emission

The major operating variables that affect spark-ignition engine performance, efficiency, and emissions at any given load and speed are: compression pressure, spark timing, fuel/air or air/fuel ratio relative to the stoichiometric ratio, ignition voltage, fuel quality and specific emission and emission index (Heywood, 1998; Heywood and Bandvadeker, 2004). The engine performance parameters of interest are power, torque, specific fuel consumption and combustion efficiency.

2.2.1 Compression Pressure

The ability of the engine cylinder to hold pressures during compression is very important as pressure data for the gas in the cylinder can be used to calculate work transfer from the gas to the piston. The cylinder pressure and the corresponding cylinder volumes through the engine cycle can be used to express the work done (Heywood, 1998; Stone, 1999);

$$W_i = \oint p\,dv \tag{2.9}$$

Where W_i = indicated work done per cycle (kJ)

p = pressure developed in the cylinder (N/m^2)

dv = volume of the charge induced per cycle (m^3)

The power in the cylinder is related to indicated work per cycle by;

$$i.P = \frac{W_i N}{n_R} \tag{2.10}$$

Where $i.P$ = indicated power developed per cycle

N = speed of the engine in rev/min

n_R = number of crank revolutions for each power stroke per cycle

Also, the power delivered by the engine is given by;

$$b.P = 2\pi NT \tag{2.11}$$

but, $T = Fb$

Where T = torque developed (Nm)

F = force exerted by expanding gases on piston head (N)

b = crank radius (m)

But, a more useful relative engine performance measure is by use of mean effective pressure given by;

$$p_m = \frac{Pn_R}{v_d N} \tag{2.12}$$

The maximum mean effective pressure of good engine design is well established and is essentially constant over a wide range of engine sizes. For most of the SI naturally aspirated engines, the mean effective pressure (which correspond to compression pressure) ranges from 10-13 bars. For design and operational calculations, the engine displacement required to provide a given torque or power at specified speed can be established by assuming appropriate values for mean effective pressure for particular applications (Heywood, 1998).

Part of the gross indicated work per cycle is used to expel exhaust gases and inducts fresh charge. An additional proportion (fP) is used to overcome the friction of the bearings, pistons and other mechanical components of the engine and to drive engine accessories. Thus the net power output (brake power) from the engine is less than the indicated power as it is affected by changes in the engine operational parameters. The relationship between the indicated power and the brake power is given by (Stone, 1999);

$$i.P = bP + fP \tag{2.13}$$

Where; bP = brake power

fP = friction power

Hence mechanical efficiency of an engine is given by;

$$\eta_m = 1 - \frac{fP}{ip} \tag{2.14}$$

Also, in an engine test, specific fuel consumption (SFC) is an important parameter in determining engine efficiency and is related to power developed by the engine;

$$SFC = \frac{\dot{m}_f}{i.P} \tag{2.15}$$

where; $\dot{m}_f$ = fuel flow rate

2.2.2 Spark Timing

If combustion starts too early in the cycle, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke is too large, yet again, if combustion starts too late, the peak cylinder pressure is reduced and the expansion stroke work transfer from the gas to the piston decreases. There exists a particular spark timing which gives maximum engine torque at fixed speed, and mixture composition and flow rate. It is referred to as maximum brake torque—timing (MBT) (Heywood, 1998). This timing also gives maximum brake power and minimum brake specific fuel consumption. MBT timing depends on speed; as speed increases the spark must be advanced to maintain optimum timing because the duration of the combustion process in crank angle degrees increases. Optimum spark timing also depends on load. As load and intake manifold pressure are decreased, the spark timing must be further advanced to maintain optimum engine performance.

Thus accurate determination of MBT timing is difficult, but is important because NO and HC emissions vary significantly with spark timing. In practice, to permit a more precise definition of spark timing, the spark is often retarded to give a 1 or 2 percent reduction in torque from the maximum value. Spark timing affects peak cylinder pressure and therefore peak unburned and burned gas temperatures. Retarding spark timing from the optimum reduces these variables, increases exhaust temperature and both engine efficiency and heat loss to the combustion chamber walls are decreased. Retarded timing is sometimes used to reduce hydrocarbon emissions by increasing the fraction oxidized during expansion and exhaust due to the higher burned gas temperatures that result. Retarded timing may be used at engine idle to bring the ignition point closer to TDC where conditions for avoiding misfire are more favourable (Stone, 1999, Abam *et al*, 2007).

2.2.3 Ignition Voltage

The voltage required by a spark plug is the maximum high theoretical voltage necessary for spark discharge. The high voltage causes electric field between the electrodes, so that the spark gap is ionized and thus becomes conducive. The high voltage generated by the ignition system can exceed 30,000 volts. The voltage reserve is the difference between this available ignition voltage and the maximum requirements at the spark plug. The maximum ignition voltage increases as a function of time due to the large electrode gap that accompany the aging process. Ignition miss occurs when this process advances to the point where the requirements exceed the available voltage (Stone, 1999). Hence electrode gap influences ignition voltage requirements. The smaller the gap, the lower the ignition voltage requirements. A narrow electrode gap will reduce the voltage required to produce an arc, but the shorter spark gap can transfer only minimal energy to the mixture and ignition miss can occur. Higher voltages are required to support an arc across a larger gap. This type of gap is effective in transforming energy to the mixture. The electrode gap is usually between 0.7mm to 1.1 mm. The precise optimized electrode gaps for individual engines are specified by engine manufacturers and are given in the repair manuals (Heywood, 1998).

2.2.4 Air/Fuel Ratio

Mixture composition effects are usually discussed in terms of the air/fuel ratio (or fuel/air ratio) because in engine tests, the air and fuel flow rates to the engine can be measured directly and because the fuel metering system is designed to provide the appropriate fuel flow for the actual air flow at each speed and load. However, the relative proportions of fuel and air can be stated more generally in terms of the fuel/air ratio or the relative air/fuel ratio. The combustion characteristics of fuel-air mixtures and the properties of combustion products, which govern engine performance, efficiency, and emissions, correlate best for a wide range of fuels relative to the stoichiometric mixture proportions. Where appropriate, therefore, lambda is normally used as the defining parameter (Heywood, 1998; Antoni *et al*, 2002).

For lean mixtures, the theoretical fuel conversion efficiency increases linearly as excess air factor lambda (λ) increases to a value of 1.03 (Antoni *et al*, 2002). Combustion of mixtures leaner than stoichiometric produces products at lower temperature, and with less dissociation of the triatomic molecules CO_2 and H_2O. Thus the fraction of the chemical energy of the fuel which is released as sensible energy near TDC is greater and hence a greater fraction of the fuel's energy is transferred as work to the piston during expansion, and the fraction of the fuel's available energy rejected to the exhaust system decreases. Engine fuel consumption and efficiency well lean of stoichiometric depend strongly on the engine combustion chamber design and the combustion characteristics of the chamber. Thus the equivalence ratio for optimum fuel consumption at a given load depends on the details of chamber design and mixture preparation quality. It also varies for a given chamber over the part-throttle load and speed range. For lighter loads and lower speeds, it is closer to stoichiometric since the residual gas fraction is higher and combustion quality is poorer with greater dilution and at lower speeds (Antoni *et al*, 2002; Guzzella and Onder, 2004).

The equivalence ratio requirements of a spark-ignition engine over the full load and speed range can now be explained from the point of view of performance and efficiency. However, since emissions depend on λ also, emission control requirements may dictate a different engine calibration. The mixture requirements in the induction system are usually discussed in relation to steady and transient engine operation. Steady operation includes operation at a given speed and load over several engine cycles with a warmed-up engine. Transient operation includes engine starting, engine warm-up to steady-state temperatures, and changing rapidly from one engine load and speed to another (Stone, 1999).

Lambda is an important parameter controlling spark-ignition engine emissions. The critical factors affecting emissions, that are governed by the λ, are the oxygen concentration and the temperature of the burned gases. Excess oxygen is available in the burned gases lean of stoichiometric. The maximum burned gas temperatures occur slightly rich of stoichiometric at the start of the expansion stroke, and at the stoichiometric composition at the end of expansion and during the exhaust process. HC emissions decrease as the stoichiometric point is approached, that is, increasing oxygen concentration and increasing expansion and exhaust stroke temperatures result in increasing HC burn up (Antoni *et al*, 2002). For moderately lean mixtures, HC emission levels vary little with equivalence ratio. Decreasing fuel concentration and increasing oxygen concentration essentially offset the effect of decreasing bulk gas temperatures. As the lean operating limit of the engine is approached, combustion quality deteriorates significantly and HC emissions start to rise again due to the occurrence of occasional partial-burning cycles. For still leaner mixtures, HC emissions rise more rapidly due to the increasing frequency of partial-burning cycles, and even the occurrence of completely misfiring cycles. The equivalence ratio at which partial-burning and misfiring cycles just start to appear depends on details of the engine combustion and fuel preparation systems, as well as the load and speed point. For rich mixtures, CO levels are high because complete oxidation of the fuel carbon to CO_2 is not possible due to insufficient oxygen. For lean mixtures, CO levels are approximately constant at a low level of about 0.5 percent or less (Guzzella and Onder, 2004).

2.2.5 Compression Ratio

Only limited studies have examined the effect of compression ratio on spark-ignition engine performance and efficiency over a wide range of compression ratios (Antoni *et al*, 2002). However it has been found that exhaust temperature decreases as compression ratio and efficiency increase until the compression ratio corresponding to maximum efficiency is reached. It has also been found that heat losses to the combustion chamber walls, as a fraction of the fuel's chemical energy, also decrease as both the compression ratio and efficiency increase. Also, increasing the compression ratio increases exhaust hydrocarbon emissions. Several trends that could contribute to this are; increased importance of crevice volumes at high r_c; lower gas temperatures during the latter part of the expansion stroke, thus producing less HC oxidation in the cylinder; decreasing residual gas fraction, thus increasing the fraction of in-cylinder HC exhausted; and lower exhaust temperatures, hence less oxidation in the exhaust system (Heywood, 1998; Stone, 1999). However, this parameter was kept constant for this study.

2.2.6 Combustion Inefficiency

Internal combustion engine exhaust contains combustible species namely CO, H_2, unburned HC and particulates as well as complete combustion products such as CO_2 and H_2O. When their concentrations are known, combustion efficiency η_c can be calculated. The chemical energy carried out of the engine in the combustibles represents the combustion inefficiency given by

$$1-\eta_C = \frac{\sum_i x_i Q_{HVi}}{\left[\frac{\dot{m}_f}{\dot{m}_a + \dot{m}_f}\right] Q_{HVf}} \tag{2.16}$$

Where the x_i are the mass fractions of CO, H_2, HC and particulate respectively, the Q_{HVi} are the lower heating values of these species, and the subscripts *f* and *a* denote fuel and air respectively. The heating values for CO and H_2 can be taken to be 10.1 MJ/kg and 120 MJ/kg respectively (Heywood, 1998; Stone, 1999). The composition of the unburned HC is not usually known, however, the heating value (typically 42 to 44 MJ/kg) is used (Heywood, 1988; Stone, 1999). The particulates present only in diesels are soot with some absorbed hydrocarbons, usually the mass fraction is low enough for their contribution to be small, and heating values for solid carbon of 32.8 MJ/kg can be used (Heywood, 1998; Stone, 1999).

2.3 Internal Combustion Engine Models

There are numerous ways of describing reality through models as explained by Ramstedt (2004), Silverlind, (2001). Some are more complex than others and different approaches may differ in both structure and accuracy. Choosing the model depends on particular situation and especially the field of application. A physical equation theoretically describing the system is the most common method since it creates a general model working for many operating areas. Its drawback is that reality might be difficult to describe correctly in the theory (Arsie *et al*, 1998). Another common approach used in model development is entirely based on measurement. The measured data is stored as a table of two or more dimensions in a so called black box and a functional relationship, mainly regression, is developed depending on input signals. This approach often provides accurate results since it is based directly on empirical formulation (Kumar and Antony 2008, Saidur *et al,* 2008). Regression analysis is a statistical methodology that utilizes the relationship between two or more quantitative variables so that one variable can be predicted from the other or others (Neter and Kutner, 1996). There are many different kinds of regression models like linear regression models, exponential or non-linear regression models and logistic regression models. The application of each of the models depend on the relationships of the variables.

2.3.1 Logistic Regression Models

Logistic regression models explain the probability of an event occurring given certain input variables. The models have been used in vehicle emission analysis to explain the probability of vehicle emission characteristics when certain emission input variables are given. The Radian High Emitter Profile (HEP) model (Bureau of Automotive Repair, 1998) is a logistic regression model with input variables that include vehicle type, model year, catalytic converters, odometer readings and type of fuel system. Many of the variables have been identified in the literature as being correlated with high emitting vehicles (Wenzel and Ross, 1998; Choo *et al*, 2007). For example, vehicle characteristics such as vehicle age (model year, odometer readings (mileage), fuel type and fuel system have been associated with higher emission or higher failure rates (Wayne and Horie, 1983; Kahn, 1996; Washburnn *et al*, 2001; Bin, 2003). Other technology based relationships that have been explored in logistic regression modelling include those between the failure rates and repairs of specific emissions control system components such as catalyst, oxygen sensors or exhaust gas recirculation (EGR) and high emission (Lawson *et al*, 1996; Heirigs *et al*, 1996; Wenzel and Ross, 1998; Choo *et al*, 2007). However, the models developed were dependent on vehicle characteristics and emission test variables and they can only be used on vehicles with the same characteristics as the vehicles used in model development (Choo *et al*, 2007). This study, therefore, adopted the approach to develop a logistic model to determine which of the engine's input variables like vehicle usage, compression pressure, ignition angle, engine speed and spark plug gap contributed to vehicle's passing or failing exhaust emission tests based on KS 1515 standards. The identified variables were further sujected to non-linear regression models to explain the effects of the engine operating parameters on engine performance and emission characteristics.

2.3.2 Non-linear Physical Regression Models

A general non-linear physical model (subscript ph) can be developed to express the engine output parameters y' (i.e T, P, CO and HC) as a function of engine operating variables v and of vector parameter p'

$$y'_{ph,i} = f(p'_i, v_i) \tag{2.17}$$

where subscript i refers to given engine operating conditions like engine torque and engine speed, mixture ratio, ignition angle and manifold pressure (Arsie *et al*, 1996b, Arsie *et al*, 1998). The parameters p' can represent physical quantities used in the model as kinetic rates, number of zones to account for temperature stratification in burned gases, and various coefficients. In many cases, their values cannot be considered constant over the entire operational range of the engine. Therefore, they are in turn expressed as a function of engine operating variables *v,* by means of n' and further parameters β, in order that the model be used in a predictive way over the whole operating range;

$$y_{ph,i} = f(p'_i(\beta, v_i), v_i, \beta) \tag{2.18}$$

A direct approach of the equation would require the following steps

(i) specification of the parameters p' of the physical model

(ii) specification of the functional structure of the relationships $p'(\beta, v_i)$

(iii) determination of the optimal values of the n' parameters β by comparison of predicted and observed values over the set of M' experimental conditions, solving a non-linear regression problem

$$minS_\beta(\beta) = \sum_{i=1}^{m'} \left[y_{ph,i}\left(p'_i(\beta, v_i), v_i, \beta\right) - y^* \right] \tag{2.19}$$

Where, y^* = measured exhaust gas content on the logarithmic scale

(iv) estimation of the statistical significance of the solution.

It has been noticed that step (iii) involves repeated model evaluation over the entire set of experimental data. For real cases, many thousands of model evaluation would be required with very high computational costs. Moreover, the entire process from (ii) to (iv) should be repeated each time a different functional structure $p'(\beta, v)$ has to be assumed, since most of the information used to arrive at the solution of equation (2.19) cannot be utilize and therefore lost (Arsie *et al*, 1998). This statement confirms the limitations of the physical models and the next sets of non-linear black box regression models were considered.

2.3.3 Multiple Non-linear Black Box Models

The development of this class of models does not require an accurate description of the physical phenomena responsible for engine performance and emission characteristics (Arsie *et al*, 1998; Saidur *et al*, 2008). It allows reducing substantially the computational time and reaching precision level required for prediction (Kumar and Antony, 2008). Multiple non-linear regressions have been applied for model structure definition paying particular attention to statistical significance of model parameters. The influence exerted by each engine variable over the experimental data set has been accurately analyzed by using the p – level (Arsie *et al*, 1998). An example of non-linear regression model has been used to describe mechanical efficiency of an engine in the following equation;

$$\begin{aligned}\eta_m = a_0 + a_1[NT \times 10^{-3}] + a_2[T \times 10^{-2}]^2 + a_3[(N \times 10^{-1})^2(p_{man} \times 10^{-4})^2]^{0.2365} \\ + a_4\left(\frac{10^3}{N}\right) + a_5[(N \times 10^{-3})(T \times 10^{-2})^2]^{-6.89.10^{-3}} + a_6(T_{man}) + a_7(p_{man})^{0.3304} \\ + a_8(\dot{m}_{air}) \qquad (2.20)\end{aligned}$$

where a_i = i-th regression coefficient

N = engine speed (rpm)

T = brake torque (Nm)

p_{man} = manifold pressure N/m^2

T_{man} = intake air temperature (K)

$\dot{m}_{air}$ = intake air flow (kg/s)

The model parameters a_i have been evaluated by means of a multiple linear regression technique, while the fraction exponents have been computed by means of non-linear regression techniques. The model prediction levels have been tested by comparison between the predicted and measured mechanical efficiency over engine's operating conditions.

Non-linear regression analysis has also been used in the original Taylor's equation used for determining the overall combustion duration in an IC engine (Kumar and Antony 2008);

$$\Delta\theta_c = 40 + 5\left(\frac{N}{600} - 1\right) + 166\left(\frac{12.5}{y'} + 1.1\right)^2 \qquad (2.21)$$

where $\Delta\theta_c$ = total combustion duration in terms of crank angle

N = engine speed (rpm)

y' = no. of moles of actual oxygen supplied to the engine

An error between the original equation and experimental values was computed and an error curve fitted in the form of a logistic equation. The error in the original equation was minimised by incorporating the a logistic equation resulting in the modified Taylor's equation;

$$\Delta\theta_c = 40 + 5\left(\frac{N}{600} - 1\right) + 166\left(\frac{12.5}{y'} + 1.1\right)^2 - \frac{37.71}{1 + 241\exp(-1.2r)} \qquad (2.22)$$

where r is compression ratio

Also the usage of 2-zone model (burned and unburned regions) together with couple analysis of flame front location and cylinder pressure data gives the following burning equation (Kumar and Antony 2008);

$$\frac{d}{dt}(mb) = \rho_u AFS_l \, {\mu^*}/{\tau^*} \qquad (2.23)$$

where mb = mass of burned gases (kg)

ρ_u = density of unburned gases (kg/m^3)

AF = air fuel ratio

S_l = laminar flame speed (m/s)

μ^* = parametric mass entrained within the flame region that has yet to burn (kg)

τ^* = total parametric mass of the gases (kg)

Equation (2.23) when effectively integrated over the relevant portion of the total combustion process, assuming turbulent characteristic velocity as proportional to mean piston speed gives the equation for the flame development angle as:

$$\Delta\theta_d = C(Vp\gamma)^{1/3}\left(h/_{S_l}\right)^{2/3} \quad (2.24)$$

where $\Delta\theta_d$ = flame development angle/ignition delay

V = cylinder volume

p = cylinder pressure

γ = kinematic viscosity

h = clearance height at the point of ignition

Equation (2.24) is applicable to SI engines in general but does not take into account the parameters related to engine geometry especially the location of the spark plug, swirl generation intake process and the effect of compression ratio. The influence of the mentioned parameters have been indirectly taken into account by fitting a polynomial equation of the 3rd order and by minimizing error by using non-linear regression analysis. The evaluation of the C in the equation has been done by using the model (Kumar and Antony, 2008).

Non-linear regression models can also be converted into linear regression. For example, the exhaust gas content concentration (ppm) as a function of engine speed (rpm), throttle position (%) and operating time (minutes) was developed (Saidur *et al*, 2008). The function relationship between exhaust gas response and independent variables is expressed in the form;

$$E' = C'N^a v_t^b t_o^c \quad (2.25)$$

where; E' = exhaust gas concentrations

C' = Regression coefficient

N = engine speed

v_t = throttle position

t_o = operating time (mins)

a, b, c = numerical values

The parameter or regression coefficient can be determined by the least square method by linearizing equation (2.25) in the logarithmic form as;

$$lnE' = lnC' + alnN + blnv_t + clnt_o \quad (2.26)$$

From equation (2.26), the following linear mathematical model was developed;

$$\eta' = \beta_o x_o + \beta_1 x_1 + \beta_2 x_2 + \beta_3 x_3 \quad (2.27)$$

where η' is true response surface roughness on a logarithmic scale with x_o = 1, x_1, x_2, and x_3 are the logarithm transformations of the engine speed, throttle position and the operation time and $\beta_o, \beta_1,\ \beta_2,\ \beta_3$ are parameters to be estimated.

The predicted error can be expressed as;

$$Y_i = y' - \varepsilon = b'_o x_o + b'_1 x_1 + b'_2 x_2 + b'_3 x_3 \qquad (2.28)$$

where Y_i is predicted exhaust gas content and y' is measured exhaust gas content on the logarithmic scale, ε is experimental random error and b' is the estimated value of β parameter. The b' values were estimated by least square method. Saidur *et al*, 2008 developed estimation response models for CO, CO_2 and HC in linear form with four coefficients;

$$Y_{CO} = 0.5535 + 0.2453x_1 + 0.3010x_2 + 0.00786x_3 \qquad (2.29a)$$

$$Y_{\mathrm{CO2}} = 2.1792 + 0.1428x_1 + 0.05762x_2 - 0.00336x_3 \qquad (2.29b)$$

$$Y_{\mathrm{HC}} = 4.5473 + 0.2630x_1 - 0.04396x_2 - 0.000674x_3 \qquad (2.29c)$$

where x_i is engine input variables and their limits must be defined. Combining independent variables equations (2.29a, …, 2.29c) can be written in the form $Y_i = lnE'$ that refers to equation (2.25), so that the estimated response of emission E' in the logarithmic function can be used. The new relation in exponential form was used to construct emission concentration of CO and CO_2 in percentage volume and HC in parts per million as a function of engine speed, throttle position and operation time.

The expressions in equations (2.17 to 2.19) explain how non-linear physical models can be constructed. However, with the limitations associated, with physical models earlier cited, this approach was not used in the development of models in the study. Equations (2.20 to 2.29) reviewed the development of various IC engines regression models. They explained how physical parameters that are not included in functional relationships can indirectly be accounted for by parameter coefficients of the models. This approach was used in the study to develop performance and emission prediction models based on experimental data.

2.3.4 Internal Combustion Engine Model Optimization

Optimization takes place in many engineering models. It aims at estimating the parameters of numerical models from experimental data for example calibration of engines.

The optimization problems consist in minimizing a function that is complex and expensive to estimate and for which derivatives are often not available with nonlinear constraints, and sometimes with several objectives among which it is necessary to find the best compromise (Berghen, 2004; Bonnans *at al*, 2003; Pianese and Rizzo, 1996). Optimization of IC engines aims at finding the best control parameters of an engine over multiple objectives such as jointly minimizing polluting agent emission and fuel consumption. It is a multi-objective nonlinear constrained optimization problem where analytical gradients are not available. Most of these numerical gradients are computed using already developed software like Sequential Quadratic Programming Algorithm (SQPAL). In developing this software, a general constrained optimization problem of the following nature is considered;

$$\underbrace{min}_{x\in\Omega} \; f(x) \text{ subject to } c_E(x) = 0,\, c_I(x) \leq 0, \tag{2.30}$$

Where a real-value function $f: \Omega \rightarrow R$ is defined in an open set Ω in IR^n, c_E and c_I are vectors of equality and inequality constraint functions, respectively. Feasibility set is further defined as $X = \{x \in \Omega$: $c_E(x) = 0, c_I(x) \leq 0$ and assume that f, c_E and c_I are differential functions. Moreover, c_E is subjective for all x in the open set Ω.

Presently, numerical methods to solve (2.30) can be grouped into two;

(i) The class of penalty methods, which includes the augmented Langrangian approaches and their Interior Point (IP) approaches

(ii) The class of direct Newtonian methods, which is mainly formed of the Sequential Quadratic Programming (SQP) approach.

Often, actual algorithms combine elements of the two classes, but their main features make them belonging to one of them. The choice of class of algorithms strongly depends on the features of the optimization problem to solve. The key issue is to balance the time spent in the simulator and in the optimization procedure. For this study, the first class was used applying equation (2.30).

2.3.5 Model Sensitivity Analysis

According to Saltelli *et al*, 2000, sensitivity analysis is the study about the relations between input and output of a model. The analysis focuses mainly on determining which of the input variable's variances influence the model the most, and which of the input variables has to be known more accurately to reduce the output variance. The simplest and most common procedure for assessing the effects of parameter variations on a model's result is to vary selected input parameters once at a time and compute the output and record the corresponding changes in the results (responses). The model parameters responsible for the largest relative changes in the responses are then classified as the most important. For complex models, the large amount of computing time needed by such recalculations severely restricts the scope of this sensitivity analysis procedure. In practice, this means only a few parameters that are seen to be important are analyzed (Sobol, 1993).

2.4 Concluding Remarks

Air pollution in general and vehicular emissions in particular were reviewed in this chapter. Specific references to factors that contribute to high vehicular emissions like vehicle age, mileage accumulation, vehicle models, vehicle technology, and maintenance culture have been reviewed. Previous studies in Nairobi and other places like South Africa and Accra city (Ghana) were reviewed and gaps in knowledge identified and incorporated in the study. Pollution formation in SI engines was also considered to help in understanding findings of the study and in drawing conclusions. Strategies used in reducing vehicular emissions were also reviewed with specific reference to pollution abatement systems and the factors that affect their efficiencies.

Different engine operating parameters that affect engine performance were reviewed. The equations governing engine performance were discussed as well. This helped in understanding the performance of the tested vehicles from the field data collected. A number of IC engine model developments were reviewed. This included physical and empirical approaches. The governing equations were important as the basis for developing, validating, optimizing and sensitivity analysis of models developed in the study.

To address the existing gaps in determining vehicular emissions levels, performance characteristics and prediction model development, an innovative approach is required. Due to high vehicles variability with different operating and maintenance characteristics, estimating vehicle emission levels using KS 1515:2000 categories is difficult. The KS 1515:2000 standards have some limitations as they are used for both type approval and regular inspection of vehicles. There is also lack of stages and legal framework for improvement to international accepted standards as stipulated by United Nation Economic Commission for Europe (UN ECE) regulations on vehicle emissions (Syafruddin *et al*, 2002).

CHAPTER THREE

MATERIALS AND METHODS

3.1 Study Area

Nairobi is the largest metropolis in Kenya, it covers an area of 684 km^2 and in 2007 it was estimated to have a population of over 2.9 million people with a density of 4230 per km^2 (KNBS, 2008). Nairobi is the capital city of Kenya as well as industrial centre of the country. Commercial activities are concentrated within the city centre whereas most of the industrial activities are located to the southeast (Mulaku and Kariuki, 2001, Maina 2004).

Nairobi is reported to be the fastest growing city in the world after Mexico City (Mexico) and Maputo (Mozambique). It is home to many companies and organizations and is established as hub for business and culture. The Globalization and World Cities Study Group and Network defined it as a prominent social centre. It does not have a regular air quality management system yet, and any measurements of air pollution have been uncoordinated (Maina, 2004). Indeed out of 20 mainly developing country cities sampled for a UN study on air quality, the air quality in Nairobi city has been rapidly deteriorating and the situation can only get worse with the increasing population and industrial activities (Mulaku and Kariuki, 2001).

The main source of air pollution in the city is vehicle emission. Statistics from Kenya Revenue Authority (KRA, 2009) show that the vehicles imported into the country annually rose from 33,917 vehicles in 2003 to 121,831 vehicles at the end of 2008, bringing the total number of vehicles in the country to about 1.7 million. Most of these cars have been used for a number of years in their countries of origin before being imported into Kenya. They are known to emit far higher air pollutants than new and properly maintained vehicles. The levels of motor vehicle related air pollution is expected to be high in Nairobi City than any other urban centers due to the fact that most of the imported vehicles are mainly used in Nairobi. This research was conducted at the Motor Vehicle Inspection Centre, Likoni Road, Nairobi to determine performance and emission characteristics of the inspected vehicles. The study was done in conjunction with the Ministry of Transport, Republic of Kenya, and the Department of Industrial and Energy Engineering, Egerton University.

3.2 Study Population and Study Design

The study population comprised of petrol vehicles that were taken for inspection at Motor Vehicle Inspection Centre, Likoni Road, Nairobi. A cross sectional design was used in this study to determine exhaust emission levels and associated factors among the inspected vehicles. The key variables examined were demographic characteristics of the vehicles, exhaust emission levels and emission factors. Data was collected between the month of April and December 2007

3.3 Sample Selection

The sample size for this study comprised 384 petrol vehicles. They were randomly selected by use of random numbers and tested from the population of petrol vehicles that came for inspection. The sample size (n=384) was obtained using 95% confidence interval and a significant level of 5%. The following formula was used (Cochran, 1977):

$$n = \frac{Z^2 PQ}{D^2} \tag{3.1}$$

where, n = sample size

Z^2 = normal variant associated with levels of significance

$P^{'}$ = is the estimated emission prevalence = 50% (since no data on vehicles emissions was available)

$Q = 1 - P^{'}$

D= is the required level of precision

$$n = \frac{(1.96)^2\, 0.5 \times 0.5}{(0.05)^2} = 384 \text{ vehicles}$$

3.4 Determination of Exhaust Emission Levels

The role of the Motor Vehicle Inspection Unit among others is to carry out annual inspection of public service and commercial vehicles prior to licensing or registration, determine their roadworthiness and verify the engine and chassis number of the vehicle to be inspected. Routine procedure at the vehicle inspection centre is that once the vehicles are booked for inspection at the Kenya Revenue Authority, they will be taken through a number of checks and tests. Those that fail to meet the requirements must correct the defects while those which meet the requirements will be issued with a certificate of inspection. It was at this point that the researcher tested the selected vehicles which passed inspection tests to determine their exhaust emission levels and associated factors. Vehicle category, vehicle usage and engine operating parameters affect vehicle exhaust emission levels. Vehicle usage was calculated by dividing mileage by age while age was calculated from the date of manufacture as indicated in the log book and mileage accumulation was read directly from the odometer. For vehicles whose odometer stopped working, mileage accumulation was calculated from a regression model develop by US EPA (US EPA, 2002).

Accumulated use (km)

$$= 489\times(\text{Yrs before Kenya}) + 19023\times(\text{Yrs in Kenya}) - 458.3\times(\text{Yrs in Kenya})^2 \qquad (3.2)$$

Data on vehicles' exhaust emission was collected using an AGS-200 Exhaust Gas Analyzer model for petrol engines as shown in Fig 3.1.

AGS200 gas analyzer

Fig.3.1 Exhaust gas analyzer

The parameters measured from the tail pipe were CO, CO_2, HC, and λ for different categories of vehicles and were displayed on the computer screen. Before the measurements were taken, the AGS 200 Exhaust Gas Analyzer was warmed up.

3.4.1 Exhaust Gases and Engine Operating Parameters Test Procedures

Exhaust emission tests were determined using AGS-200 Exhaust Gas Analyzer with the engine warmed and enrichment devices not operating. The engine was required to remain idling and was not subjected to any significant electrical loading. The exhaust system was required to be free from any leakage. For exhaust gases, the test criterion was based on KS1515-2000 specifications, where the tested vehicles were expected to meet individual gas limits. The limits for CO were given as 4.5%, 3.5%, 0.5%, and 0.2% for non-catalytic vehicles before 1986, non-catalytic vehicles between 1986 and 2002, catalytic vehicles between 1986 and 2002, and catalytic vehicles after 2002 respectively. HC limits were 1200 ppm for non-catalytic vehicles before 1986, non-catalytic vehicles between 1986 and 2002, while 250 ppm and 200 ppm for catalytic vehicles between 1986 and 2002 and catalytic vehicles after 2002 respectively. The limit for air excess factor lambda (λ) was only considered for catalytic vehicles between 1986 and 2002, and catalytic vehicles after 2002 which was taken as 1.00 ± 0.03. However, the overall test results of pass or fail was based on CO limits only.

(i) Non-Catalyst Test

A temperature and engine speed probe was connected to the engine to obtain the temperature and engine speed readings as shown in Fig. 3.2. Exhaust Gas Analyzer probe was also fitted in the exhaust pipe to determine the proportions of carbon monoxide (CO), hydrocarbon (HC), carbon dioxide (CO_2) and excess air factor λ in the exhaust gas over a period of 5 seconds at idle speed. If the vehicle met the CO requirements at its normal idling speed but failed the HC, the HC levels were checked at high idle speed of 2000 rpm.

Speed and Temperature measurements

Fig.3.2 MGT300-rpm counter/Temperature probe

(ii) Catalyst Test

Carbon monoxide (CO), hydrocarbon (HC) and lambda were measured at fast idle speed and CO checked again at idle speed. The first fast idle speed test was done by raising the engine speed to the vehicle specific fast idle speed mostly between 2500-3000 rpm and maintained for 30 seconds. CO, HC and λ values were recorded in the last 5 seconds as Basic Emission Test (BET) results. If the vehicle failed the 1st idle speed additional engine pre-conditioning was done by running the engine between 2000-3000 rpm for 3 minutes or until all emissions were within limits. After engine pre-conditioning, a second fast idle speed was done by repeating the procedure of first fast idle test. This was followed by catalyst stabilization, which required the vehicle specific fast idle speed be maintained for 30 seconds. Finally, the engine was allowed to idle for 30 seconds and during the last five seconds, the CO readings were recorded.

(iii) Compression Pressure

All the spark plugs were removed from the engine cylinder head and the throttle valve was left wide open to ensure that maximum amount of air enters into the cylinders. Then the compressor adaptor was screwed into the spark plug hole of cylinder number 1 as shown in Fig. 3.3. To protect the coil from high voltage, the primary lead from the negative terminal of the coil was disconnected. For the electronic system, the positive lead to the control unit was disconnected. The throttle was held wide open as the starter motor was operated to crank the engine through the four compression strokes. The needle moved around to indicate the maximum compression in the cylinder. The same procedure was repeated for the rest of the cylinders.

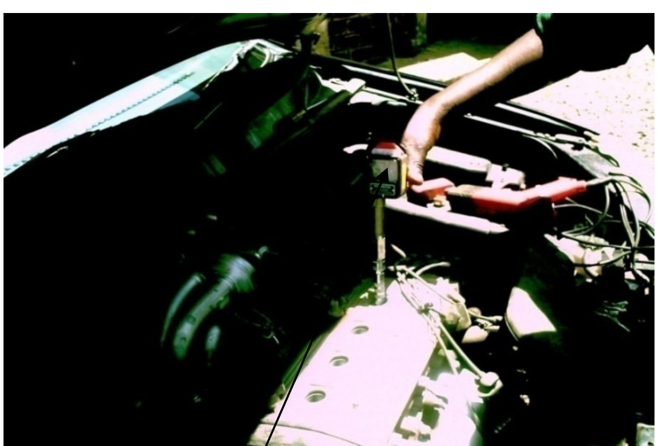

Fig.3.3 Engine compression pressure measurement

(iv) Ignition Angle

Ignition angle was determined by use of a stroboscope as shown in Fig. 3.4. The stroboscope lead was connected to number 1 spark plug cable when the engine was running at idle speed. Each time number 1 plug fired, the stroboscope flashed. This happened so quickly that when the light was pointed at the crankshaft pulley it appeared stationary. A value in degrees corresponding to a mark in the pulley was recorded.

Fig.3.4 Engine ignition angle measurement

(v) Spark Plug Gap

Spark plug gap was measured using a thickness gauge (feeler gauge). All spark plugs were removed from the engine and their gap checked and recorded for every vehicle.

3.4.2 Simulation of Field Data on an Engine Test Bed

The experimental apparatus comprised a hydraulic dynamometer with the auxiliary instrument allowing a complete measurement of the main engine parameters, such as torque, fuel consumption, air consumption and speed as shown in Fig 3.5. A gas analyzer was used for measuring the concentrations of carbon monoxide, carbon dioxide and unburned hydrocarbon in the combustion products. The analyzer also provided air-fuel ratio based on the concentration of some specific gases in the exhaust system.

A 4-cylinder four stroke engine of 2500 cm^3 displacement, 110kW output power at 5400 rpm and maximum torque of 220 Nm at 3800 rpm was used for the experiment. Several experiments were conducted in order to set reference parameters for necessary comparison and modeling. The investigation started by running the engine within a speed range of 1000 rpm to 5000 rpm and corresponding values of specific fuel consumption (SFC), engine torque and output power were recorded. Engine timing was adjusted from 5^o to 35^o BTDC. For each adjustment, maximum torque and corresponding engine rpm were recorded. Also, recorded were specific fuel consumption (SFC) and output brake power (bP). In order to obtain consistency of the data, air fuel mixture was set at a lambda value of 0.95.

The engine test rig had provision for adjustment of the air fuel ratio, which were measured using an exhaust gas analyzer. The engine ignition timing was fixed at 10^o BTDC and 2500 rpm. For this condition, corresponding SFC and torque were recorded while brake power and fuel conversion efficiency (η_f) was calculated. Spark plug of NGK type, recommended gap of 0.8 mm was used in the testing. The gap was adjusted between 0.6 mm to 1.1 mm and corresponding values of SFC and bP were recorded.

In the measurement of exhaust emissions, the engine speed was fixed at 2500 rpm. For this engine condition, three independent variables namely ignition timing, excess air factor (λ) and spark plug gap, were adjusted to determine their effects on carbon monoxide (CO), carbon dioxide (CO_2) and hydrocarbon (HC) emissions.

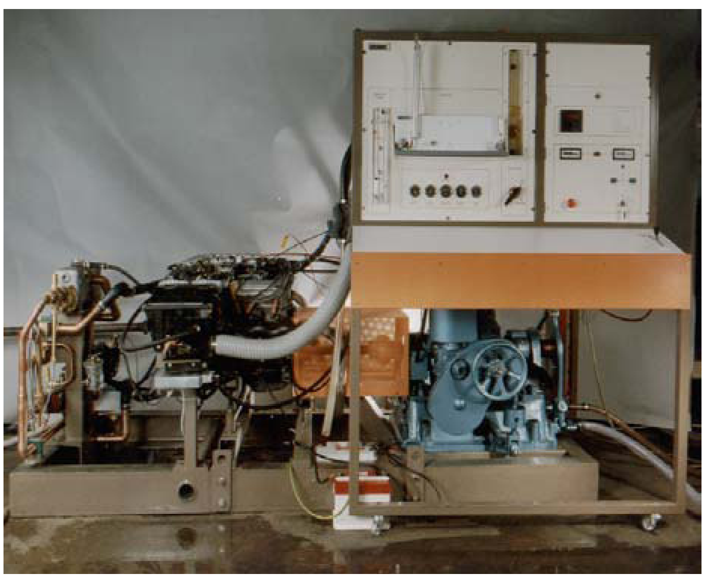

Fig. 3.5: Engine test bed used for simulation

3.5 Data Analysis

Data was coded and then entered into Microsoft Excel and Statistical Analysis System (SAS) Version 9.0 for analysis. Data cleaning was done and frequencies were ran. Cross tabulation was done to look for differences and relationships among variables. Descriptive analysis was carried out on vehicle characteristics and associated factors using t-test. *Chi*-square test was used to determine exhaust emission levels at 5% level of significance. A Logistic regression model was fitted on tested results and factors associated with it namely; vehicle usage, compression pressure, ignition angle, engine speed, and spark plug gap. The fitted logistic regression model was in the form:

$$\ln(Y) = \ln\left(\frac{\pi}{1-\pi}\right) = \beta_0 + \beta_1 U + \beta_2 P_c + \ldots + \beta_6 \lambda \tag{3.3}$$

where π =Prob($Y=y/U = x_1, P_c = x_2, \ldots, \lambda = x_6$) = $\left(\dfrac{e^{\beta_0+\beta_1 x_1+\ldots+\beta_6 x_6}}{1+e^{\beta_0+\beta_1 x_1+\ldots+\beta_6 x_6}}\right)$

Y=Test results

U = Vehicle usage

P_c= Compression pressure

θ = Ignition angle

S = Engine speed

G = Spark plug gap

λ = Lambda

β_i = Parameter coefficient

The fitted logistic model's effectiveness was assessed by overall model evaluation, statistical tests on the regression and the individual estimation parameters. The statistical test for the logistic regression coefficients was implemented using the Wald Chi-square. Standard engine performance equations were used to derive engine performance characteristics from the parameters, while non-linear regression models were used to predict engine performance and emissions based on engine operating parameters.

3.6 Engine Performance and Emission Prediction Modeling

The procedure for developing engine performance and emission models was through individual sub-models superimposed to obtain final models. From experimental data, graphs were drawn to determine their nature. From the drawn graphs, a general form of the model was assumed. In this case, most graphs were parabolic and some exponential in nature. The principle of least squares was employed as shown in Appendix I. The *Fit [ ]* function in *Mathematica* ® software was used to compute parameters of the sub-models developed. The syntax for a parabolic fit for sub-model of SFC versus crank angle for example was;

Fit $[\{\{\theta_1, F'_1\}, \{\theta_2, F'_2\}, \ldots\{\theta_n, F'_n\}\}, \{1,\theta,\theta^2\}, \theta]$

The same procedure was done to develop other parabolic sub-models. For exponential sub-model, the syntax was *Fit* $[\{\{\lambda_1, F'_1\}, \{\lambda_2, F'_2\}, \ldots, \{\lambda_n, F'_n\}\}, \{1, exp[\lambda]\}, \lambda]$

The same *Fit [ ]* function used in the sub-models was also applied respectively as shown in appendix I to get the superimposed model parameters. The syntax for the superimposed model for SFC prediction given speed (S), crank angle (θ), lambda (λ) and spark plug gap (G) was;

$Fit[\{\{S_1, \theta_1, \lambda_1, G_1, F'_1\}, \{S_2, \theta_2, \lambda_2, G_2, F'_2\}, \ldots, \{S_n, \theta_n, \lambda_n, G_n, F'_n\}\}, \{1, (a_oS^2+a_1S+a_2), (b_o\theta^2+b_1\theta+b_2), (c_o\lambda^2+c_1\lambda+c_2), (d_oG^2+d_1G+d_2)\}, \{S, \theta, \lambda, G\}]$

The same procedure was done for the development of maximum torque, brake power, carbon monoxide, carbon dioxide, and hydrocarbon prediction models.

3.7 Models Validation

Model validity refers to the stability and reasonableness of the regression coefficients, the plausibility and usability of the regression function, and the ability to generalize inferences drawn from the regression function. Validation is a useful and necessary part of the model-building process (Neter and Kutner, 1996). Separate data was used in checking the model for plausibility signs and magnitudes of estimated coefficients, agreement with earlier empirical results and theory, and model diagnostic checks such as distribution of error terms, and normality of error terms.

How well the model accounted for the variation in the data was given by the percentage error, which is the proportion of difference between the predicted and observed values in experiment given by:

$$\%\ error = \frac{predicted\ value - observed\ value}{predicted\ value} \times 100 \qquad (3.4)$$

3.8. Optimization of Performance and Emission Models

In this study, the SFC, the model was optimized against maximum torque, brake power, CO, CO_2, and HC models. SFC is an input parameter which affects all other engine output parameters while torque is an engine design parameter where other parameters like power are derived from. However, the function of four variables in this model was rather difficult to visualize and therefore, the quadratic functions developed were solved using *mathematica* ® software. To achieve this, the Langragian method of undetermined multipliers was used. This was done by taking;

$$F' = \psi_1(S, \lambda, \theta, G) + \gamma'_1\psi_2(T) + \gamma'_2\psi_3(bP) + \gamma'_3\psi_4(CO) + \gamma'_4\psi_5(CO_2) + \gamma'_5\psi_6(HC) \quad (3.5)$$

where $\gamma'_1, \gamma'_2, \gamma'_3, \gamma'_4$ and γ'_5 are the Langragian undetermined multiplier.

3.9 Sensitivity Analysis of Models

The sensitivity analysis was done by determining which of the input variables influenced the model output variances the most. Using the optimal SFC, HC and CO values of S, θ, λ and G from the Langragian function, the optimal values were changed by ±10% and the corresponding effects on SFC, maximum torque, power, CO, CO_2 and HC were analyzed.

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Vehicles Test Results

A total of 384 petrol vehicles were tested against the KS1515-2000 specification standards on vehicles exhaust emissions. For all the tests, emission concentrations were measured directly from exhaust gases which included hydrocarbon (HC), carbon monoxide (CO) and carbon dioxide (CO_2). Vehicles passed or failed the tailpipe inspection test based on the established standards of CO and HC. From the study, 267 failed the test criteria. There was significant difference between the vehicles which failed and those which passed emission tests in the category of non catalytic before 1986, non-catalytic between 1986 and 2002 and catalytic between 1986 and 2002 ($\chi^2 = 25.4848$, $p = 0.0001$; $\chi^2 = 37.435$, $p = 0.0001$; $\chi^2 = 10.952$, $p = 0.0001$ respectively). For vehicles manufactured after 2002, there was no significant difference between the vehicles which passed and those which failed emission tests ($\chi^2 = 0.2725$, $p = 0.6015$) as shown in Table 4.1

Table 4. 1: Vehicles test results for different categories

Test results	Non-catalytic Before 1986	Non-catalytic 1986-2002	Catalytic 1986-2002	Catalytic After 2002	Total
Fail	31	139	81	15	266
Pass	2	54	44	18	118
Total	33	193	125	33	384
χ^2	25.4848	37.435	10.9520	0.2725	57.0417
p	0.0001	0.0001	0.0001	0.6015	0.0001

The engine performance parameters including vehicle usage, compression pressure, ignition angle, engine speed, spark plug gap, and excess air factor (λ) were considered as shown in Table 4.2. The mean vehicle usage was 19640 km/yr, 16559 km/yr, 14328 km/yr and 13904 km/yr for non-catalytic vehicles before 1986, between 1986 and 2002, catalytic vehicles between 1986 and 2002 and after 2002 respectively. The vehicle usage for non-catalytic vehicles before 1986 was statistically significant from the rest of the vehicles. There was no significant difference between compression pressure of vehicles manufactured between 1986 and 2002 but differed significantly with those of vehicles manufactured before 1986 and after 2002. The lowest compression pressure of 6.8 bar was recorded in the non-catalytic vehicles before 1986, while vehicles manufactured after 2002 recorded highest average compression pressure of 11.5 bar. For ignition angles, only vehicles manufactured before 1986 were advanced at a mean value of 341^o BTDC while non-catalytic between 1986 and 2002, catalytic between 1986 and 2002 and catalytic after 2002 had average ignition angle at 348^o, 346^o and 350^o BTDC respectively. There was no significant difference between the ignition timing of the vehicles manufactured between 1986 and 2002 and differed significantly with those of vehicles manufactured before 1986 and those manufactured after 2002. The non-catalytic vehicles had slightly higher idle speed with a mean value of 968 rpm and 902 rpm for before 1986 category and between 1986 and 2002 category respectively, while catalytic vehicles between 1986 and catalytic after 2002 had average idle speed at 872 rpm and 806 rpm respectively. All the idle speeds significantly differed from each other. The spark plug gap for non-catalytic vehicles was 0.9 mm and 1.0 mm for catalytic vehicles. Both categories of non-catalytic vehicles operated at rich mixture of 0.85 lambda and 0.91 lambda for before 1986 category and between 1986 and 2002 category, while the two categories of catalytic vehicles operated at required mixture with a mean lambda value of 0.99. The excess air factor of catalytic vehicles was statistically significant from those of non-catalytic vehicles.

Table 4.2: Mean values of vehicle operating parameters

Engine operating parameters (mean values)	Vehicle category			
	< 1986 Non-catalytic	1986 -2002 Non-catalytic	1986 -2002 Catalytic	>2002 Catalytic
Vehicle usage (km/yr)	19640^a	16559^b	14328^b	13904^b
Compression pressure (bar)	6.8 ^{a}	9.6^b	9.8^b	11.5^c
Ignition angle (deg)	341^a	348^b	346^b	350^c
Engine speed (rpm)	968^a	902^b	872^c	806^d
Spark plug gap (mm)	0.9^a	0.9^a	1.0^b	1.0^b
Lambda	0.85^a	0.91^b	0.97^c	0.99^c

Means followed by same letters in the same row are not significant ($\alpha = 0.05$)

The results of vehicle usage conforms to the results of the study done by US EPA in 2002 where it found that the average vehicle usage in Nairobi was 17837 km/yr (US EPA, 2002). This showed that vehicles in Nairobi remain in use for many years as they accumulate a lot of mileage. The high vehicle usage tends to increase exhaust emissions especially if there are no proper vehicle maintenance programs (Wenzel, 1999). Low cylinder pressure as reported in non-catalytic vehicles before 1986 leads to low cylinder temperature which delays ignition commencement. This results in longer mixing times giving rise to larger pre-mixed combustion phases and conversely less diffusion combustion (Laguitton *et al*, 2006). Also, the pressure generated in the cylinder creates torque at the cylinder crankshaft and any compromise in cylinder pressure causes a corresponding loss in torque. This leads to low engine power, high fuel and oil consumption, off-gassing from positive crankcase ventilation (PCV) port and soaked PCV breather (Heywood, 1998; Stone, 1999).

Advance ignition angles as seen in non-catalytic vehicles before 1986 affects cylinder pressure as more time is allowed for pressure to build up and maximum cylinder pressure is most likely to be reached while the piston is still moving up. This will increase frictional power, hence, affecting engine power output as stated by Stone (1999), and Abam *et al*, (2007). The engine idle speed is meant to produce just enough engine power to take care of all frictional losses while engine is running unloaded. Higher engine speed than idle speed as seen in both categories of non-catalytic vehicles means the engines were producing unutilized power which has a lot of implications on fuel consumption and exhaust emissions (Heywood, 1998). The spark plug gap for non-catalytic vehicles was large implying that higher voltage is required to support an arc across a larger gap. However, this category of vehicles has limited ignition voltage as it utilizes convectional ignition system (Heywood, 1998). Both categories of non-catalytic vehicles operated on rich mixture which is an indication of high fuel consumption and exhaust emissions as also observed by Guzzella and Onder, 2004.

4.1.1 Vehicle Emission Levels

Exhaust emission levels were analyzed by use of t-test. The t-test tries to determine whether the difference between the mean values of the measured exhaust emission levels and the standard values as given in KS1515-2000 specifications were significant. The emission levels considered here were; carbon monoxide (CO), hydrocarbon (HC), carbon dioxide (CO_2) and the excess air factor (λ) for all categories of vehicles.

The difference in the means between the measured CO values and the standard CO values in all categories of vehicles were statistically significant with the mean values of CO measured being higher than that of standard values as shown in Table 4.3. The same observation was made between measured CO_2 and HC and standard values for all categories of vehicles. There was no significant difference in measured and standard lambda for both categories of catalytic vehicles, while those of non-catalytic vehicles differed significantly.

Table 4.3: Exhaust emission levels t-test analysis

Vehicle Category	Exhaust gases	$\bar{x}-\mu$	Std error	t
Non-catalytic before 1986	CO	2.32	0.36	6.54*
	CO_2	7.29	0.48	5.21*
	HC	614	44.2	10.95*
	Lambda	0.14	0.03	5.65*
Non-catalytic 1986 - 2002	CO	0.91	0.19	4.78*
	CO_2	5.45	0.21	26.37*
	HC	484	28	22.00*
	Lambda	0.08	0.01	9.10*
Catalytic	CO	1.16	0.12	9.56*
	CO_2	3.78	0.13	27.01*
	HC	83.0	27.0	9.56*
	Lambda	0.03	0.01	1.31^{NS}
Catalytic after 2002	CO	0.26	0.09	3.00*
	CO_2	2.64	0.10	6.91*
	HC	53.4	11.0	4.78*
	Lambda	0.01	0.001	0.65^{NS}

* $0.0001 \leq p \leq 0.05$, NS $P > 0.05$

Fig. 4.1 shows that age is a major contributing factor to emission levels. Vehicles manufactured before 1986 had emission levels of 51.6% higher than the recommended levels. The HC levels reduced from a mean value of 964 ppm to 616 ppm for the same category of vehicles manufactured between 1986 and 2002. Also, CO_2 levels increased from a mean value of 8.21% volume to 10.07% volume for non-catalytic vehicles manufactured before 1986 and those manufactured between 1986 and 2002 respectively. The same trend was observed for catalytic vehicles where CO, and HC for vehicles manufactured between 1986 and 2002 were higher than those manufactured after 2002. Vehicles manufactured before 1986, though high emitters, accounted for only 8.6% of the tested vehicles and 12.4% of the vehicles which failed the emission tests. The trend of decreasing emission with decrease in age was an indication that older vehicles produce higher emissions, and the more likely they are to be failed vehicles. The findings of the study were consistent with the findings of Wenzel and Ross (1998), Washburn *et al* (2001), and Bin (2003), which found out that as vehicles' mileage and age increases, the likelihood of being identified as failed vehicles increases. Age being a function of normal degradation of emissions controls of properly functioning vehicles can result in increase in emissions, and malfunction or outright failure of emission controls on some vehicles, possibly resulting in very large increases in emissions, particularly CO and HC (Wenzel, 1999). While the findings agree with the notion that emission control failures are typically associated with older vehicles, it is important to realize that older vehicles are not held to the same emission standards as newer vehicles. Instead the findings may suggest that a more complex relationship exists between emissions, technology, and age.

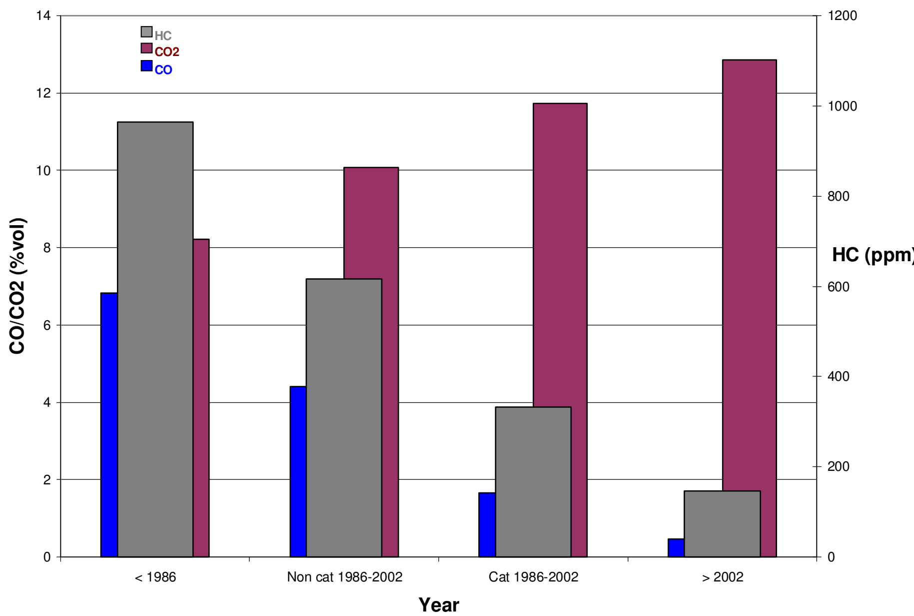

Fig. 4.1: Vehicle age versus emission levels

Vehicles manufactured before 1986 were likely to have accumulated high vehicle usage which could easily affect engine performance and efficiency due to repeated engine strain resulting in higher values of CO. Most of the non-catalytic vehicles manufactured between 1986 and 2002 use a carburetor as the fuel metering device. Repeated adjustment and repairs especially by unskilled mechanics make the system to be less accurate and may lead to wrong fuel metering (Langat *et al*, 2004). The observation was consistent with the 1996 American Petroleum Institute (API) study on vehicle emissions. A clear distinction was made between carbureted and fuel injected vehicles where carbureted vehicles tended to be older with higher mileage than fuel injected vehicles (Heirigs *et al*, 1996). The higher CO concentrations above the limits among the catalytic vehicles after 2002 could be because most of the vehicles were registered in the country before 2005 when leaded fuels were still in use and therefore the catalytic converters could be damaged.

Studies have shown that CO can be bound by blood haemoglobin in a stronger way than oxygen to form carboxyhaemoglobin (COHb) which inhibits the supply of oxygen to body tissues (Botter *et al*, 2002). Exposure to CO can affect the cardiovascular system, central nervous system, the foetus and other organs that are oxygen deficient. The impact of CO on the cardiovascular system can be easily observed even in low concentration. Heart and lung disease patients are the most sensitive group to CO exposure. It has been found that 16 % COHb concentration is hazardous to health (Botter *et al*, 2002). The effect of CO exposure in foetus is based on the principle that exposure to high concentrations of CO can decrease oxygen supply in pregnant women with the consequences of premature birth or baby born with abnormal weight. According to World Health Organization (WHO, 2002), exposing the most sensitive group of people (heart and lung disease patients) to CO with more than 2.5% concentration of COHb in the blood is equivalent to an exposure of CO with concentration of $35\mu g/m^3$ for one hour or $20\mu g/m^3$ for eight hours. Therefore to avoid concentration of COHb amounting to 2.5 % – 3.0 %, WHO suggests that exposure to CO may not exceed 25 ppm (29 $\mu g/m^3$) for one hour (WHO, 2002). This information is very important to ensure that vehicle exhaust emissions are within the limits to ensure that their effect on human life is minimized.

The presence of HC in exhaust gases indicated incomplete combustion. However, since hydrogen atom has higher affinity to oxygen than carbon atom, some of the HC may be oxidized to hydrogen and water hence reducing the overall HC levels in the exhaust gases. Pollutants such as polyaromatric hydrocarbon (PAH), ethylene and benzene have carcinogenic and mutagenic properties. Initially carbon dioxide was not considered a pollutant, therefore, it does not have specification limits like other exhaust gases. However, because of its impact on global warming, it is being currently targeted for reduction by improving engine efficiency. Carbon dioxide is an inevitable byproduct that is produced when the carbon from fuel is fully oxidized during the combustion process. The higher the carbon dioxide reading, the more efficient is the engine operation. Therefore, A/F imbalances, misfires, or engine mechanical problems may cause CO_2 to decrease. Near perfect combustion will result in carbon dioxide levels which approach the theoretical maximum of 15.5%. Carbon dioxide, like oxygen, is affected by air fuel ratio, spark timing, fuel quality and mechanical conditions of the engine as described by Stone, (1999), Guzzella and Onder, (2004).

Lambda which is a ratio of actual air/fuel ratio and stoichiometric air/fuel ratio is not a pollutant but it has a lot of impact on exhaust emission levels. CO and HC are relatively low and CO_2 is high near the theoretical value ($\lambda = 1 \pm 0.03$) while a rich air/fuel mixture results in high concentrations of CO and HC and low concentrations of CO_2. The importance of carrying out exhaust gas analysis in all the four gases (CO_2, CO, HC and lambda) are excellent troubleshooting tools. It allows one to narrow down the potential cause of driveability and emission concerns while focusing the troubleshooting tests on the area(s) most likely to be causing the concern, and hence save diagnostic time and minimize errors. In addition, it also gives the ability to measure the effectiveness of repairs by comparing exhaust readings before and after repairs (Bureau of Automotive Repair, 2003).

4.1.2 Probability of the Engine Parameters Affecting Test Results

Equation 3.3 was used to determine the probability of a vehicle failing the test criteria using parameters x_1, x_2, x_3, x_4, x_5 and x_6 as independent variables. The hypothesis tested was that the likelihood that a vehicle fails the test was related to; Y=Test results of vehicle usage, compression pressure, ignition angle, engine speed, spark plug gap and lambda. The independent variable was the test results while vehicle usage, compression pressure, ignition angle, engine speed, spark plug gap and lambda were the predictors/explanatory variables. The test results were coded as 1=Fail and 2=Pass, the vehicle categories were coded as 1=Non-catalytic before 1986, 2=Non-catalytic between 1986 and 2002, 3=Catalytic between 1986 and 2002 and 4=Catalytic after 2002. The model parameter estimates are presented in Table 4.4. Some variables like fuel type, body type and transmission type were excluded in the model as there was no theoretical justification to include them (Washburn *et al*, 2001). The estimated parameters were used to test the probability of the vehicle failing emission tests and were related to vehicle usage, compression pressure, ignition angle, engine speed and spark plug gap. From the table, the probability values suggest that the coefficients β_0, β_1, $\beta_2, \beta_3, \beta_4$ and β_5 apart from β_6 were not statistically significant at 5% significance level for vehicles manufactured before 1986, while $\beta_1, \beta_2, \beta_3$, β_5 and β_6 were statistically significant at 5% for non-catalytic vehicles manufactured between 1986 and 2002. Also β_1 and β_3 were significant at 5% for catalytic vehicles manufactured between 1986 and 2002, while β_1 and β_2 were significant for catalytic vehicles after 2002.

Table 4.4: Parameter estimate for logistic regression model

Vehicle Category	Parameter	Estimate	Std Error	P-value
Non-catalytic	β_0	464.6	223.1	0.9550
before 1986	β_1	-0.00112	0.00057	0.8292
	β_2	-15.7825	4.6535	0.7238
	β_3	-0.8071	4.2222	0.9734
	β_4	-0.0706	0.0074	0.9243
	β_5	34.6242	13.1	0.9462
	β_6	- 16.5329	4.5352	0.0013
Non-catalytic	β_0	-57.5845	41.6393	0.1667
between 1986	β_1	0.000037	0.000021	0.0777
and 2002	β_2	-1.7190	0.0328	0.0001
	β_3	0.00956	0.00049	0.0535
	β_4	0.1411	0.0111	0.2057
	β_5	18.8360	4.2141	0.0001
	β_6	-4.2119	1.3919	0.0025
Catalytic	β_0	523.0	267.5	0.1547
between 1986	β_1	0.000207	0.000077	0.0073
and 2002	β_2	-1.3152	0.4773	0.3733
	β_3	0.0372	0.0022	0.0928
	β_4	1.4733	0.09831	0.1340
	β_5	-28.1704	6.0589	0.2797
	β_6	2.9545	0.634	0.6511
Catalytic	β_0	17.9671	2.073	0.9307
after 2002	β_1	-0.00018	0.00010	0.0733
	β_2	-10.1167	4.4870	0.0242
	β_3	0.2216	0.0542	0.6827
	β_4	0.0553	0.0078	0.1818
	β_5	-20.4581	2.6836	0.1068
	β_6	-27.0092	0.5935	0.4411

According to the fitted model, the log of odds of a vehicle failing the test is positively related to vehicle usage, ignition angle and spark plug gap and negatively related to compression pressure and lambda for non-catalytic vehicles manufactured between 1986 and 2002, while positively related to vehicle usage and ignition angle for catalytic vehicles between 1986 and 2002. Also, the odds of vehicles failing was negatively related to vehicle usage and compression pressure for catalytic vehicles after 2002. In other words, the bigger the values for the positive variables, the higher the chances of the vehicle failing the test, and the smaller the values for the negative variables the higher the chances of the vehicle failing the test. Overall, vehicle usage and compression pressure influenced test results more than the other parameters. Lambda also influenced the test results for non-catalytic vehicles. This is because vehicle usage is a function of normal degradation of emission controls of properly functioning vehicles, resulting in moderate emission increase (Wenzel, 1999). Compression pressure and fuel metering can be affected by lack of proper inspection and maintenance. Long service intervals affect a vehicle's lubricant properties resulting in increase in wear which affects compression pressure and fuel metering (Heywood, 1998; Bin 2003).

4.1.3 Evaluations of the Fitted Logistic Models

The model effectiveness was assessed by overall model evaluation, statistical tests on the regression and the individual estimated parameters. The overall model was performed by examining the null model (intercept only model) and the fitted logistic regression model. The null model provides a baseline because it contains no predictors. A logistic model is said to be a better fit if its diagnostics are smaller than those of the intercept-only model. Consequently, the fitted logistic model has a better fit than the null model. This is proved by the Akaike Information Criterion (AIC), Likelihood Ratio and Schwarttz Criterion (SC) tests, all of which yielded similar conclusions. In all the cases, fitted logistic model minimized the AIC and SC, while maximizing the likelihood ratio relative to the null model. The tests are presented in the Table 4.5.

Table 4. 5: Overall model evaluation

Vehicle category	Criterion	Intercept only	Intercept and Covariates
Non-catalytic before 1986	AIC	17.090	12.017
	SC	18.586	20.996
	-2 Log L	15.090	0.017
Non-catalytic 1986-2002	AIC	230.804	121.547
	SC	234.067	141.124
	-2 Log L	228.804	109.547
Catalytic 1986-2002	AIC	164.169	55.529
	SC	166.997	72.499
	-2 Log L	162.169	43.529
Catalytic After 2002	AIC	47.475	38.613
	SC	48.971	47.598
	-2 Log L	45.475	26.613

The statistical test for the regression coefficients was implemented using the Wald *Chi*-square statistic for the three criteria. The results are presented in Table 4.6;

Table 4. 6: Wald *Chi*-square table

Vehicle category	Test	Chi-Square	df	P-value
Non-catalytic before 1986	Likelihood Ratio	15.0723	5	0.0101
	Score	11.3850	5	0.0443
	Wald	0.30250	5	0.9976
Non-catalytic 1986-2002	Likelihood Ratio	119.2569	5	0.0001
	Score	93.9806	5	0.0001
	Wald	45.1628	5	0.0001
Catalytic 1986-2002	Likelihood Ratio	118.6399	5	0.0001
	Score	83.0424	5	0.0001
	Wald	18.9953	5	0.0019
Catalytic after 2002	Likelihood Ratio	18.8611	5	0.0020
	Score	14.5865	5	0.0123
	Wald	8.5413	5	0.1288

The null hypothesis of no significant regression was strongly rejected for both categories of vehicles manufactured between 1986 and 2002 by the three tests at 5% significance level. For non-catalytic vehicles before 1986 and catalytic vehicles after 2002, the regression model was also considered to be significant because Likehood Ration and Score statistics were significance despite the fact that the Wald statistic failed to reject the hypothesis of no significance in regression. In overall, the combination of independent variables (vehicle usage, compression pressure, ignition angle, idle speed, and spark plug gap) significantly contributed to the probability of failure or pass for the vehicles studied.

4.2 Performance and Emission Characteristics of SI Engines

The engine used for simulation of field data was a 4-cylinder four-stroke engine, 2500 cc displacement, 110 kW output power at 5400 rpm and maximum rated torque was 220 Nm at 3800 rpm. The effect of engine speed on engine performance was determined by varying engine speed from 1000 rpm to 5000 rpm. The ignition timing was initially set at 10^o BTDC and excess air factor at 0.95. The tests were carried out at various throttle settings and results were recorded in Table 4.7. As shown in Fig. 4.2, the engine speed was varied over the full range and engine characteristics were plotted on a base of engine speed. It can be seen that as speed increases from the lower value of 1000 rpm, brake power increased from 17 kW to a maximum value of 67 kW at an engine speed of 4000 rpm. Beyond 4000 rpm, the power output decreased. This is because volumetric efficiency decreases with increase in speed caused by gas temperature, valve timing and valve mechanism dynamics as described in Heywood (1998) and Stone (1999). Torque increased with speed to a maximum value of between 2000 rpm and 3000 rpm after which it decreased with increase in speed. The throttle chokes the flow at low speeds as the throttle open area is reduced, increasingly limiting air flow.

Specific fuel consumption (SFC) characteristic showed a minimum value between a speed of 2000 rpm and 3000 rpm. This is the most economical operating speed for engine in terms of the amount of fuel required to produce unit brake power. Around this speed, the thermal efficiency of the engine reaches the maximum value (Heywood, 1998). Speed beyond this value will increase specific fuel consumption as more energy is required to overcome frictional power, which is directly proportional to the speed as stated by Stone (1999). The results obtained from the experiment were different from the engine specifications given due to the conditions of the engine. A maximum output power of 67 kW at 4000 rpm was obtained from the experiment, which was about 61 % of the rated power. Also a maximum torque of 182 Nm was recorded at engine speed of 2500 rpm. The value was 82.7 % of the maximum rated torque. The speed of 5400 rpm for maximum power output was also not achieved because of the condition of the engine.

Table 4.7: Effect of engine speed on SFC, torque and bP

Engine speed (rpm)	Engine performance parameters		
	Torque (Nm)	SFC (g/kWh)	Brake power (kW)
1000	152	487	17
2000	178	409	37
3000	179	431	56
4000	160	457	67
5000	122	545	64

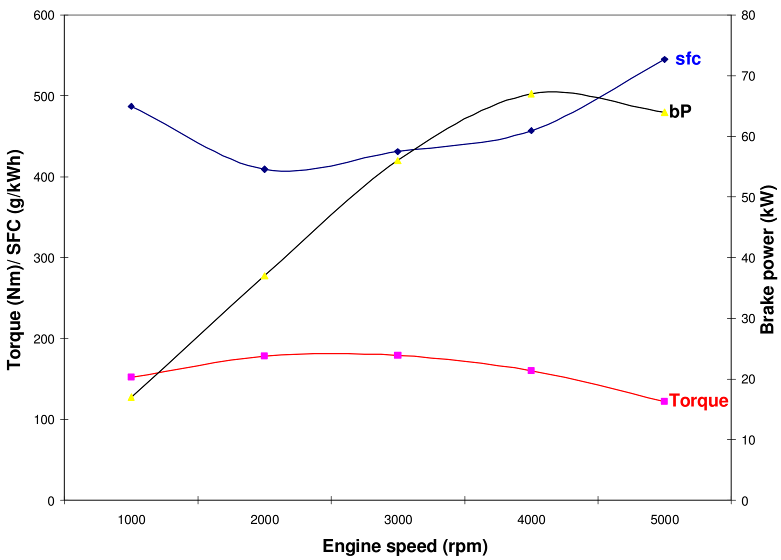

Fig 4.2: Effect of engine speed on SFC, torque and bP

The relationship describing performance curves between the SFC, max Torque and brake power and the speed are given by the equations 4.1a to 4.1c which were obtained using principle of least squares method in Appendix I.

$SFC = 576.142 - 0.122159S + 0.0000236652\ S^2 \qquad (1000 \leq S \leq 5000) \qquad (4.1a)$

$T_{(max)} = 76.5931 + 0.0868755S - 0.0000179363S^2 \qquad (1000 \leq S \leq 5000) \qquad (4.1b)$

$bP = -31.2362 + 0.0486668S - 6.61124\times10^{-6}S^2 \qquad (1000 \leq S \leq 5\,000) \qquad (4.1c)$

Engine performance curves as a function of speed are normally parabolic in nature. At low engine speeds there are relatively large heat losses through the wall thus reducing engine efficiency, while at high speeds, the combustion time becomes unfavorably large as compared to the available interval in the expansion stroke as described by Arsie *et al* (1998). The constants in the equations 4.1a to 4.1c are unique and are determined by engine condition as shown by the range of experimental data. The coefficients of *S* indicate the rate of change

SFC, maximum torque and power with respect to speed. The rate of change of the parameters is not constant, therefore, the coefficients of S^2 indicate the rate of change of the rate of change.

Engine speed has been found to affect engine performance as mixture burning rate is strongly influenced by it. It is well established that the duration of combustion in crank angle degrees only increases slowly with increasing engine speed. The burning rate throughout the combustion process increases, though not quite as rapidly as engine speed as found by Patton *et al* (1989) and Cavina *et al* (2002). It has also been established that engine speed influences frictional losses and volumetric efficiency. The total friction work per cycle and thus the frictional mean effective pressure for given engine geometry vary with engine speed, hence affecting engine performance. The factors were not considered in the study, therefore, the constants and coefficients of speed in equations 4.1a to 4.1c are supposed to take care of the factors.

The variations in the engine performance parameters were due to engine deterioration over time, therefore, justifying the nature of the performance curves in Fig. 4.2. The specifications for tune-up were also expected to be different from the original specifications. The results from the study showed that most of the vehicles in Nairobi were over 10 years with average vehicle activity of over 1400 km/year resulting in most of them failing emission tests. Studies have shown that there is a positive correlation between vehicle emission levels and engine condition (Bureau of Automotive Repair, 2003; Choo *et al*, 2007). There was no specification available on specific fuel consumption (SFC) but a figure of 270 g/kWh can be assumed for a new engine. The results of SFC obtained from the experiments were much higher than 270 g/kWh, an indication that the engine condition was poor. Studies have shown that an important factor in achieving optimal engine performance and reduced emissions is the availability of service and repair facilities with good diagnostic equipment and qualified technical personnel (Bureau of Automotive Repair, 2002; Ale and Nagrkoti, 2003; Ale, 2004). Maintenance of vehicles is crucial as it reduces fuel consumption (Pint *et al*, 2008). Studies have also shown positive correlations between vehicle test results and engine conditions. The high percentage of vehicles failing the emissions tests from the study indicate that the vehicles tested in Nairobi were poorly maintained.

The engine performance curves in Fig. 4.3 were of an engine set at required timing of 10^{o} BTDC. The effect of ignition timing on engine performance was determined by varying the initial setting of crank angle from 5^{o} to 30^{o} BTDC. At each selected crank angle, maximum torque, SFC and corresponding speed were recorded and bP was derived. The results are shown in Table 4.8. The test was carried out at excess air factor of 0.95, spark plug gap of 0.8 mm. From Fig. 4.3, it can be observed that changing ignition timing from the optimal value affected engine performance. There was a reduction in torque from 179 Nm to 176 Nm and an increase in SFC from 407 g/kWh to 429 g/kWh at 3000 rpm when ignition timing was changed from 10^{o} BTDC to 15^{o} BTDC. A similar observation was made when ignition timing was changed to 30^{o} BTDC. The torque reduced from 160 Nm to 136 Nm and SFC increased from 457 g/kWh to 463 g/kWh at 4000 rpm.

Table 4.8: Effect of engine crank angle on SFC, torque and bP

Crank angle (deg. BTDC)	Engine performance parameters		
	Torque (Nm)	SFC (g/kWh)	Brake power (kW)
5	163	454	24
10	182	407	49
15	176	429	55
20	163	443	58
25	150	551	59
30	136	463	57

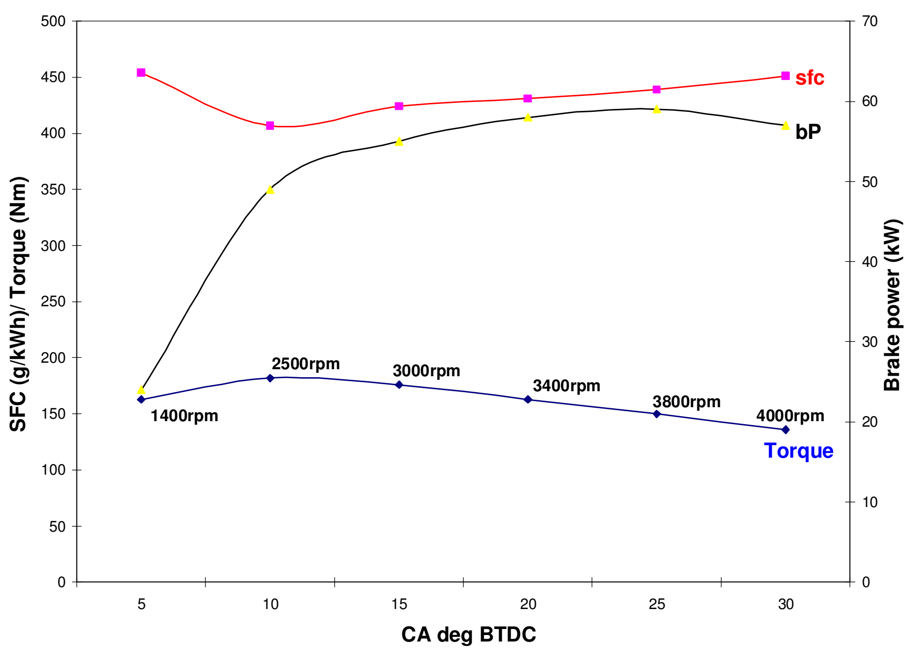

Fig 4.3: Effect of engine crank angle on SFC, torque and bP

The relationship describing performance curves between the SFC, maximum torque and brake power and the crank angle are given by equations 4.2a to 4.2c.

$$SFC = 461.9 - 4.803570\theta + 0.170714\,\theta^2 \qquad (5 \le \theta \le 30) \qquad (4.2a)$$

$$T_{(max)} = 153.9 + 3.43071\,\theta - 0.137857\,\theta^2 \qquad (5 \le \theta \le 30) \qquad (4.2b)$$

$$bP = 4.7 + 5.00643\,\theta - 0.110714\,\theta^2 \qquad (5 \le \theta \le 30) \qquad (4.2c)$$

It can be observed that ignition timing depends on speed. As speed increases, the ignition must be advanced to maintain optimum timing because the duration of the combustion process in crank angle degrees increases (Heywood, 1998; Sierens *et al*, 2005). This is because if combustion starts too early in the cycle, the work transfer is too large, hence increasing frictional forces resulting in torque reduction and increasing SFC. If ignition starts too late, the work transfer is too small due to shortening of the time for combustion. The increase in SFC is due to expulsion of unburned fuel with exhaust gases. Therefore, there exists a particular ignition timing, which gives maximum brake power and minimum fuel consumption at constant speed (Heywood, 1998; Kim *et al*, 2006). However, determination of maximum brake timing is difficult but important because engine performance varies significantly with ignition timing (Stone, 1999; Eichlsender *et al*, 2003).

It is also important to note that as engine conditions deteriorate due to wear, initial engine timing tends to be advanced to give enough time for the pressure build up in the cylinder to improve on its performance. The exact position of ignition advance needs to be determined (Heywood, 1998; Stone, 1999). From the study, it was found out that the average ignition timing for vehicles manufactured before 1986 were about 20^o BTDC. The position could not be ascertained whether it was the position corresponding to maximum torque of the engine.

The effects of air fuel ratio on engine performance parameters were determined by adjusting air fuel ratio from 0.77 to 1.30 excess air factor. The engine was fixed at 10^o BTDC, 2000 rpm and spark plug gap of 0.8 mm. The corresponding values of SFC and torque were recorded while bP and fuel conversion efficiency (η_f) were derived from SFC data as shown in Table 4.9.

Table 4.9: Effect of lambda on SFC, Torque, bP and η_f

Lambda	Engine performance parameters			
	Torque (Nm)	SFC (g/kWh)	bP (kW)	η_f (%)
0.77	158	584	33	14
0.83	210	478	44	17
0.91	191	431	40	19
0.95	178	409	37	20
1.11	172	390	36	21
1.25	167	372	35	22
1.30	148	409	31	20

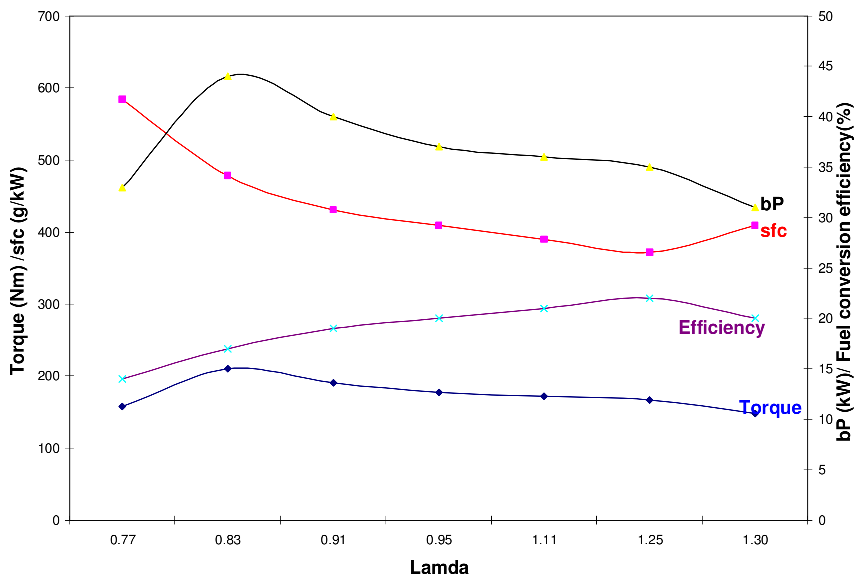

Fig. 4.4 Effect of lambda on SFC, Torque, bP and η_f

The relationship between SFC, maximum torque, brake power and fuel conversion efficiency are given by equations 4.3a to 4.3d

$$SFC = 2318.99 - 3458.63\lambda + 1529.35\lambda^2 \qquad (0.77 \leq \lambda \leq 1.3) \qquad (4.3a)$$

$$T_{(max)} = 43.4225 + 300.712\lambda - 160.486\lambda^2 \qquad (0.77 \leq \lambda \leq 1.3) \qquad (4.3b)$$

$$bP = -21.8802 + 127.711\lambda - 66.7119\lambda^2 \qquad (0.77 \leq \lambda \leq 1.3) \qquad (4.3c)$$

$$\eta_f = -48.4937 + 123.222\lambda - 54.0067\lambda^2 \qquad (0.77 \leq \lambda \leq 1.3) \qquad (4.3d)$$

From Fig. 4.4, it can be observed that SFC reduces as excess air factor increased to a value of 1.25 after which it increased again. This is because a lean mixture burns more slowly and has lower maximum temperature than a less lean mixture (Stone, 1999). The air fuel mixture therefore, affects engine efficiency and power output. At different engine speeds, it can be seen how the SFC (inverse of efficiency) and power output vary.

It can also be seen that maximum power output of 44 kW was achieved with rich mixture of 0.833 excess air factor and beyond this point, power output reduced. Rich mixture containing more than optimum amounts of fuel usually produces more power. Engine power is generally at its maximum when it is about 15 % to 20 % rich (Heywood, 1998). This is because with dissociation occurring, there will be maximum power output with rich mixtures when as much oxygen as is consumed (Stone, 1999; Rahman *et al*, 2009). Conversely, for maximum economy, as much of the fuel should be burned as possible implying a weak mixture with excess oxygen present. The weaker the air fuel mixture, the higher the ideal cycle efficiency. However, when the fuel mixture becomes too weak, as seen in Fig. 4.4, the combustion becomes incomplete and efficiency again falls (Heywood, 1998). It has also been established that lambda affects mixture burning rate. Both flame development and burning angles show minimum for slightly rich mixture (λ = 0.833) and increase significantly as mixture become substantially leaner than stoichiometric (Cavina *et al*, 2002; Rahman *et al*, 2009). The burned gas fraction in the unburned mixture, due to residual gas fraction and any other recycled exhaust gases, affect the burning rate as increasing the burned gas fraction slows down both flame development and propagation.

To maintain the required precision of air-fuel ratio, vehicle fuel systems need to be maintained regularly. From the study, most of the tested vehicles (79.6%) operated on a rich mixture which is an indication of poor maintenance. Studies have shown that vehicles' system performance reduces with time due to repeated usage and poor maintenance (Wenzel and Ross, 1998; Bureau of Automotive Repairs, 2003). Correlation has also been established between vehicle's age, performance, and maintenance levels. Old vehicles tend to perform poorly because of low maintenance levels due to high maintenance costs (Pint, *et al*, 2008).

The spark plug gap was adjusted from 0.6 mm to 1.1 mm and its effects on engine performance determined. The engine timing was initially fixed at 10^{o} BTDC, 2000 rpm and 0.95 excess air factor. The values of SFC were recorded and bP derived as shown in Table 4.10.

Table 4.10: Effect of spark plug gap on SFC, Torque and bP

Engine speed (rpm)	Engine performance parameters		
	Torque (Nm)	SFC (g/kWh)	Brake power (kW)
0.6	172	422	36
0.7	177	416	37
0.8	178	410	37
0.9	177	418	37
1.0	171	426	36
1.1	162	434	34

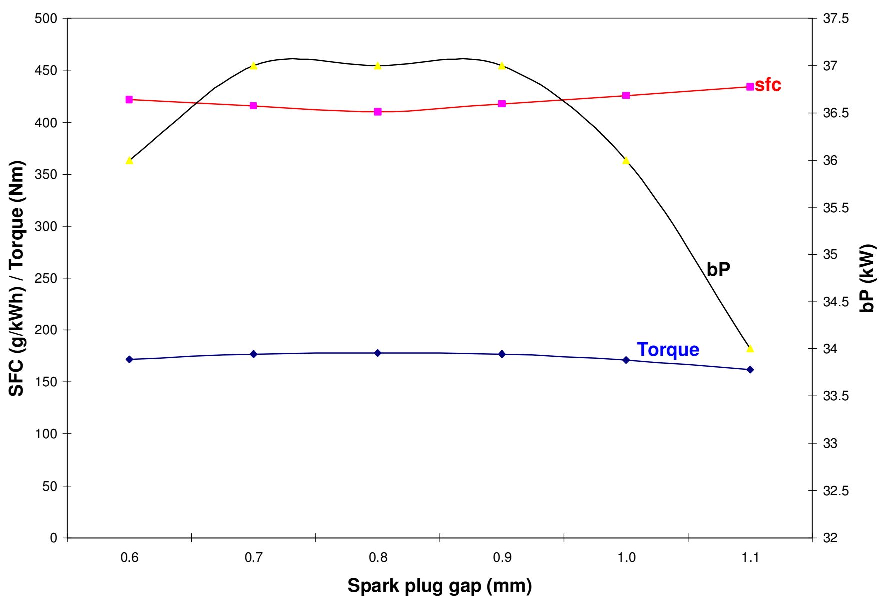

Fig. 4.5: Effect of spark plug gap on SFC, Torque and bP

The relationship between SFC, max torque and brake power with respect to spark plug gap is given by equations 4.4a to 4.4c.

$$SFC = 553.2 - 354.5G + 225G^2 \qquad (0.6 \le G \le 1.1) \qquad (4.4a)$$

$$T_{(max)} = 68.2571 + 277.786G - 175G^2 \qquad (0.6 \le G \le 1.1) \qquad (4.4b)$$

$$bP = 15.8 + 53.9643G - 33.9286G^2 \qquad (0.6 \le G \le 1.1) \qquad (4.4c)$$

From Fig 4.5, minimum SFC and maximum power was obtained at a spark plug gap of 0.8 mm and an increase in SFC and decrease in power was recorded with deviation of spark plug gap from 0.8 mm. This is because a smaller spark plug gap than the recommended size reduces voltage required to produce an arc, hence, the small gap transfer only minimal energy to the mixture resulting in lower energy conversion efficiency. Also, higher voltages are required to support an arc across a larger gap and this normally is limited by a specific ignition design resulting in weaker spark intensity, hence, affecting combustion processes (Adler, 2000). From the study, it was observed that some of the vehicles used spark plugs with big gaps than the recommended size. The big gaps were as a result of improper adjustment or due to wear (electrode erosion). The erosion of plug electrodes over extended mileage increases the gap width and requires a higher break down voltage. Also, spark plug fouling due to deposit build up on the spark plug insulator can result in side tracking of the spark (Adler, 2000). Excessive deposits of carbon also result in smaller spark plug gap in some instances. The spark plug plays greater role in combustion processes in the combustion chamber. The gap specifications given for every engine depend on design and operating parameters. Empirical observations of the spark plug voltage characteristic have shown that variation in engine parameters lead to changes in voltage characteristics (Dziubinski *et al*, 2007). Engine parameters tend to change over time due to deterioration, and hence, different the optimal operating values from the initial ones are likely to exist (Heywood, 1998).

The effects of ignition timing on IC engine emissions was determined by adjusting ignition timing between 5^o to 35^o BTDC and the corresponding values of carbon monoxide (CO), carbon dioxide (CO_2), hydrocarbon (HC) and fuel flow rate (m_f) were recorded in Table 4.11. The engine speed was fixed at 2500 rpm, 0.8 mm spark plug gap and 0.95 excess air factor.

Table 4.11: Effect of engine crank angle on CO, CO_2 , HC and m_f

Crank angle (deg. BTDC)	Engine performance parameters			
	CO (% vol)	CO_2 (% vol)	HC (ppm)	m_f (g/s)
5	6.3	9.1	514	6.1
7	5.7	9.4	489	5.7
10	5.2	9.8	473	5.5
15	4.8	10.3	479	5.6
20	4.7	10.4	486	5.7
25	4.7	10.1	593	5.8
30	4.7	9.9	501	5.9
35	4.7	9.7	507	6.2

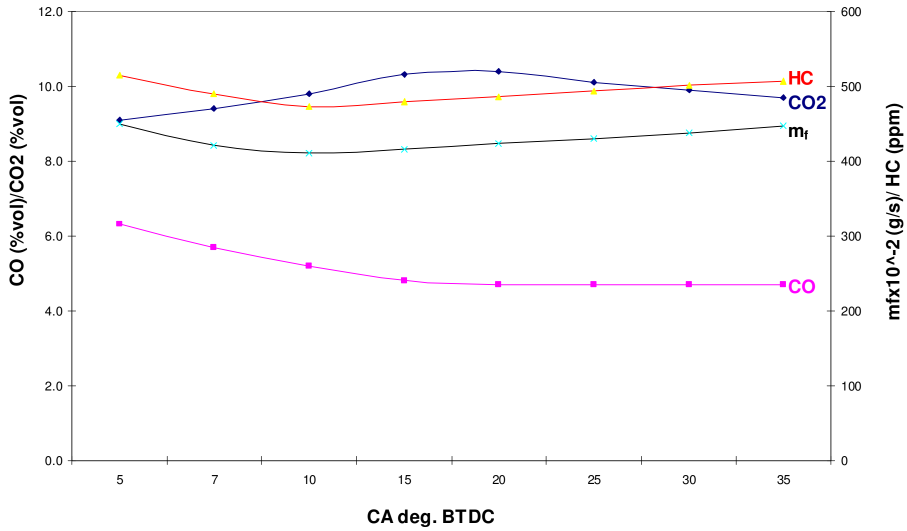

Fig 4.6. Effect of engine crank angle on CO, CO_2 , HC and m_f

The relationship between CO, CO_2 and HC with respect to crank angle is given by equations 4.5a to 4.5c.

$$CO = 6.93145\text{-}0.182943\theta + 0.00350551\,\theta^2 \qquad (5 \le \theta \le 35) \qquad (4.5a)$$

$$CO_2 = 8.35879 + 0.12585\,\theta - 0.00422943\,\theta^2 \qquad (5 \le \theta \le 35) \qquad (4.5b)$$

$$HC = 516.49\text{-}3.94114\theta + 0.109729\theta^2 \qquad (5 \le \theta \le 35) \qquad (4.5c)$$

It can be observed from Fig 4.6 that retarding ignition timing from 10^o BTDC increased CO, HC concentrations and reduced CO_2 concentrations from 5.4% vol. to 6.3% vol. 473 ppm to 514 ppm and 9.8% vol. to 9.1% vol., respectively. It also increased m_f from 5.7 g/s to 6.2 g/s. This is because retarding ignition timing decreases the peak cylinder pressure causing incomplete combustion resulting in the increase of CO and HC concentrations while reducing CO_2 concentrations (Heywood, 1998). However, advancing ignition timing beyond 10^o affected the concentrations of CO from 5.2% vol. to 4.7% vol. while there was an increase of HC concentrations from 473 ppm to 507 ppm and increasing of CO_2 concentrations from 9.8% vol. to a maximum of 10.4% vol. at 20^o BTDC before reducing to 9.7% vol. There was also an increase in m_f from 5.4 g/s to 6.2 g/s.

Advancing ignition timing so that ignition occurs earlier in the cycle increases peak cylinder pressure because more of the fuel burns after TDC, hence, an increase in SFC. Higher peak pressures result in higher temperatures and with sufficient oxygen, complete combustion occurs (Stone, 1999; Cavina, 2002). However, the increase of HC concentrations is as a result of high cylinder pressure due to advancing ignition timing. During compression and combustion, the increase in cylinder pressure forces some of the gas in the cylinder into the crevices, or narrow volumes connected to the combustion chamber, volume between piston, rings, and cylinder wall. Most of this gas is unburned air fuel mixture and much of it escapes the primary combustion process because the entrance to these crevices is too narrow for flame to enter. The gas which leaves these crevices later in the expansion and exhaust processes is a source of HC emissions as observed by Heywood (1998) and Stone (1999).

One of the most important variables in determining spark ignition engine emissions is air fuel ratio which is expressed in excess air factor (λ) in this study. Lambda was varied from 0.83 to 1.3 and the corresponding values of CO, CO_2, HC and m_f were recorded in Table 4.12. The ignition timing was fixed at 10^o BTDC, engine speed at 2500 rpm and spark plug gap at 0.8 mm.

Table 4.12: Effect of lambda on CO, CO_2 , HC and m_f

Lambda	Engine performance parameters			
	CO (% vol)	CO_2 (% vol)	HC (ppm)	m_f (g/s)
0.83	7.6	7.9	631	7.3
0.88	6.7	8.3	581	6.7
0.93	5.9	9.1	516	6.3
0.95	5.2	9.8	473	5.5
1.00	4.1	10.9	414	4.8
1.10	3.8	10.4	433	5.1
1.20	3.8	10.3	447	5.2
1.30	3.8	10.2	561	5.4

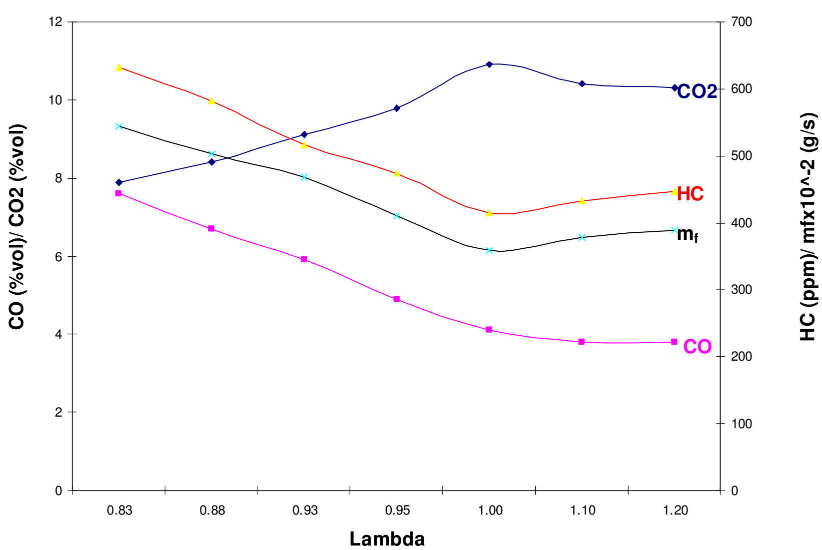

Fig 4.7: Effect of lambda on CO, CO_2 , HC and m_f

Equations 4.6a to 4.6c give the relationship between CO, CO_2, HC and m_f with respect to lambda.

$$CO = -0.741057+16.7074e^{-\lambda} \qquad (0.83 \le \lambda \le 1.3) \qquad (4.6a)$$

$$CO_2 = -13.5521+39.2993\lambda -15.9306\lambda^2 \qquad (0.83 \le \lambda \le 1.3) \qquad (4.6b)$$

$$HC = 2396.46-3310.26\lambda+1384.9\lambda^2 \qquad (0.83 \le \lambda \le 1.3) \qquad (4.6c)$$

From Fig. 4.7, it can be seen how CO, CO_2, HC and m_f vary with this parameter. The spark ignition engine is normally operated close to stoichiometric or slightly rich fuel to ensure smooth and reliable operation (Heywood, 1998). Leaner mixtures give lower emissions until combustion quality becomes poor and eventually misfiring occurs causing HC emissions to rise sharply. It can be observed that there is a direct relationship between m_f and CO and HC emissions. It can also be noticed that CO_2 levels are a function of complete combustion rather than the amount of fuel consumed. The amount of CO and CO_2 concentrations is limited by the amount of fuel available in the mixture and because of excess air, complete combustion will always be achieved (Stone, 1999). Studies have shown that for rich mixtures, incomplete combustion and water/gas shift reactions substantially reducing thermodynamics efficiency (Cavina *et al*, 2002). For mixtures slightly leaner, sufficient oxygen is available for complete combustion and efficiency is not affected by changing lambda. However, for very lean mixtures, misfiring start to occur causing a rise in levels of CO and HC in the exhaust. The findings therefore, support the field results where high exhaust emission levels were recorded among vehicles operating with rich mixture.

The effects of spark plug gap on spark ignition engine emissions was determined by adjusting the gap between 0.6 mm to 1.1 mm and the corresponding values of CO, CO_2, HC and m_f were recorded in Table 4.13. The ignition timing was set at 10^o BTDC, excess air factor at 0.95 and engine speed at 2500 rpm.

Table 4.13: Effect of spark plug gap on CO, CO_2 , HC and m_f

Spark plug gap (mm)	Engine performance parameters			
	CO (% vol)	CO_2 (% vol)	HC (ppm)	m_f (g/s)
0.6	6.1	8.8	496	5.7
0.7	5.6	9.7	478	5.6
0.8	5.3	9.9	473	5.5
0.9	5.5	9.8	476	5.6
1.0	5.8	9.4	481	5.6
1.1	6.1	8.9	487	5.7
1.2	6.3	8.5	493	5.8

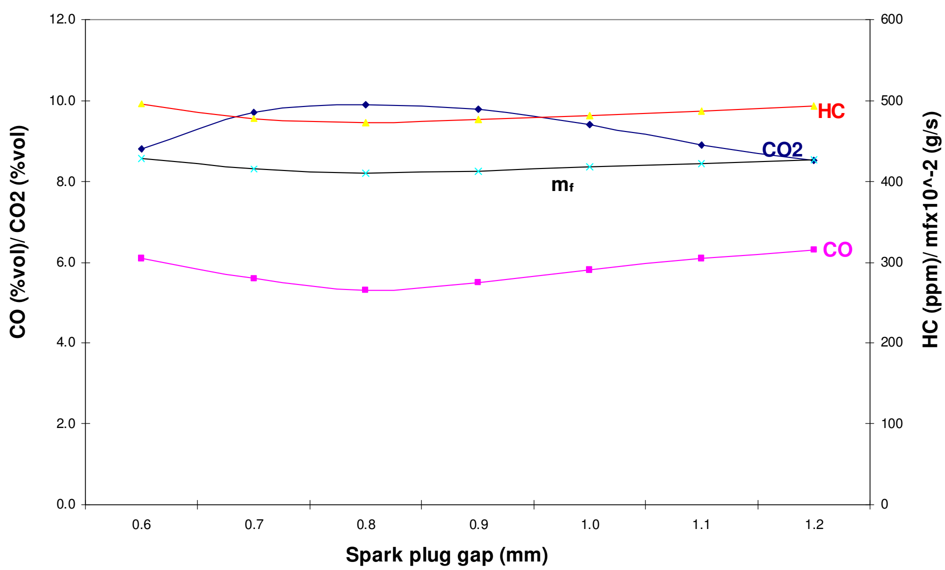

Fig 4.8: Effect of spark plug gap on CO, CO_2 , HC and m_f

The relationships between CO, CO_2 and HC with respect to spark plug gap are given by equations 4.7a to 4.7c.

$$CO = 5.59048-1.82143G+2.02381G^2 \quad (0.6 \le G \le 1.2) \quad (4.7a)$$

$$CO_2 = 0.533333+21.6429G-12.619G^2 \quad (0.6 \le G \le 1.2) \quad (4.7b)$$

$$HC = 642.048-377.5G +213.095G^2 \quad (0.6 \le G \le 1.2) \quad (4.7c)$$

From Fig. 4.8, it can be observed that there was marginal change of CO concentration over the whole range of spark plug gap, while CO_2 concentrations were maximum at 0.88 mm and started reducing when the gap was reduced or increased beyond 0.88 mm. HC emissions followed the same trend with minimum concentrations at 0.8 mm and increased on either side of gap adjustment. Minimum m_f was also recorded at 0.8 mm. A smaller spark plug gap reduces voltage required to produce an arc. Hence, the small gap transfers only minimal energy to the mixture resulting in lower energy conversion efficiency resulting in increased HC and reduced CO_2 concentrations. If a spark plug operates for excessively long periods of time without the removal of the deposits of carbon formed in the central electrode, a high voltage may track to earth over the dry or sometime wet deposits so that insufficient energy will be available to jump the spark gap of the electrodes. Also, higher voltages are required to support an arc across a larger gap and this is normally limited by a specific ignition design resulting in weaker spark intensity, hence, affecting combustion processes and may result in misfiring (Alder, 2000; Dziubinski *et al*, 2007).

From Fig. 4.2 to Fig. 4.10, it was observed that the engine performance parameters were different from the originally designed parameters due to the conditions of the engine. The emission levels in Fig. 4.6, Fig. 4.7 and Fig. 4.8 can be attributed to the conditions of the engine. Studies have shown that high exhaust emission levels are attributed to poor engine performance resulting in poor combustion characteristics and high fuel consumption (UNEP, 1999; Bureau of Automotive Repair, 2002; Choo *et al*, 2007). The high exhaust emission levels from the study can then be linked to poor vehicle maintenance. Vehicles entirely depend on properly functioning components to keep emission levels low (UNEP, 1999). Minor malfunctions in the air and fuel or spark management systems can increase emission levels significantly and major malfunctions can cause emissions to skyrocket. A relatively small number of vehicles with serious malfunctions frequently cause the majority of the vehicle related pollution problems. Unfortunately, it is rarely obvious which vehicles fall into this category as the emissions themselves are not noticeable and emission control malfunctions do not necessarily affect vehicles' driveablity. It is only well established inspection and maintenance programs that can control this (UNEP, 1999). It is also important to note that the results obtained from the engine test bed may not directly reflect the conditions of the engines in the field as engine input parameters were controlled during tests giving specific performance trends. The input parameters in the field were of varied combination resulting in a more complex relationship as opposed to what was given by laboratory tests. However, the results can be used to support the arguments presented in the discussion.

4.3 Development of SI Engine Performance and Emission Models

The model approach used in the study employed the non-linear regression technique because the performance and emission curves were parabolic and exponential in nature. The models are characterized by very limited computational demand, exhibit excellent performance over a large experimental data with less than ten parameters as found by Arsie *et al* (1996a), Arise *et al* (1998). The development of the models does not require an accurate description of physical phenomena responsible for performance and emission. It allows reducing substantially the computational time and reaching precise levels compatible with physical models as used by Saidur *et al* (2008); Kumar and Antony (2008). The models developed in the study predicted specific fuel consumption, SFC (g/kWh), maximum torque, T (Nm) and brake power, bP (kW) from engine input variables namely, engine speed, S (rpm), crank angle/ignition angle, θ (^{o} BTDC), lambda, λ and spark plug gap, G (mm). The other models predicted carbon monoxide, CO (% vol), carbon dioxide, CO_2 (% vol) and hydrocarbon, HC (ppm) from crank angle/ignition angle, θ (^{o} BTDC), lambda, λ and spark plug gap, G (mm). Speed was excluded in the emission models because emission tests were carried out at constant speed. Since it was not possible to develop the models by varying all the engine input variables at the same time, sub-models as given in equations 4.1a through 4.7c were developed by varying each of the engine variables at a time while keeping the rest constant. The sub-models were then combined to form superimposed models.

The sub-models in equations 4.1a, 4.2a, 4.3a, and 4.4a predict SFC from speed, crank angle, Lambda and spark plug gap respectively. Equations 4.1b, 4.2b, 4.3b, and 4.4b predict maximum torque based on the same input variables while equations 4.1c, 4.2c, 4.3c, and 4.4c predict brake power within the same range of input variable. The prediction of CO, CO_2 and HC from crank angle, lambda, and spark plug gap is given by the sub-models in equations 4.5a through 4.7c. The constants in the equations are dependent on the input variables, while the coefficients have linear and quadratic effects. When the quadratic effect is small, the function approaches linear. The models developed predict the performance and emission characteristics of the engine used in the experiment and any other SI engine with the same performance curves. SI engine is designed to operate at optimal values of operating parameters which when plotted over the whole range give engine performance and emission curves. The curves change over time due to engine wear. But for any given engine conditions, there exists optimal values that give the best engine performance.

The overall models that can predict SFC, T_{max}, and bP, from speed (S), crank angle (θ), lambda (λ) and spark plug gap (G) and CO, CO_2 and HC from crank angle (θ), lambda (λ) and spark plug gap (G) are given in equations (4.8 to 4.13). The function within the bracket is an expression of individual sub-models which can independently predict the engine output parameter. The constant outside the bracket in each equation gives a linear combination of the sub-models and there is an overall constant of the superimposed model. The models are valid within the following ranges of independent variables; $1000 \leq S \leq 5000$ rpm, $5 \leq \theta \leq 35$ deg BTDC, $0.77 \leq \lambda \leq 1.3$ and $0.6 \leq G \leq 1.2$.

SFC model:

$$SFC = \psi(S,\theta,\lambda,G) = 1.78158(576.142 - 0.122159S + 0.0000236652S^2)$$
$$-0.68929(461.9 - 4.883357\theta + 0.170714\theta^2) + 0.980964(2318.99 - 3458.63\lambda$$
$$+1529.35\lambda^2) + 0.759604(553.2 - 354.5G + 225G^2) - 767.893 \tag{4.8}$$

Maximum torque model:

$$T_{max} = \psi(S,\theta,\lambda,G) = 1.18874(76.5931 + 0.0868755S - 0.0000179363S^2)$$
$$+0.0957315(153.9 + 3.430071\theta - 0.137857\theta^2) + 1.42897(43.4225 + 300.712\lambda$$
$$-160.486\lambda^2) + 1.31692(68.2571 + 277.786G - 175G^2) - 511.983 \tag{4.9}$$

Brake power model:

$$bP = \psi(S,\theta,\lambda,G) = 1.0876(-31.2362 + 0.0486668S - 6.61124 \times 10^{-6}S^2)$$
$$-0.0623495(4.7 + 5.006430\theta - 0.110714\theta^2) + 0.90759(-19.8802 + 127.711\lambda$$
$$-66.7119\lambda^2) + 1.69442(15.8 + 53.9643G - 33.9286G^2) - 102.231 \tag{4.10}$$

Carbon monoxide model:

$$CO = \psi(\theta,\lambda,G) = 1.08589(6.93145 - 0.1829430\theta + 0.003505510\theta^2)$$
$$+0.938307\left(-0.741057 + 16.7074e^{-\lambda}\right)$$
$$+0.548638(11.281 - 13.6071G + 7.97619G^2) - 8.74393 \tag{4.11}$$

Carbon dioxide model:

$$CO_2 = \psi(\theta, \lambda, G) = 1.0744(8.35879 + 0.182585\theta - 0.00422943\theta^2)$$
$$+0.93717(-13.5521 + 39.2993\lambda - 15.93\lambda^2)$$
$$+0.826081(0.533333 + 21.6429G - 12.619G^2) - 17.7725 \quad (4.12)$$

Hydrocarbon model:

$$HC = (\psi, \theta, \lambda, G) = 0.789274(516.49 - 3.9411\,\theta + 0.109729\theta^2)$$
$$+1.00435(2396.46 - 3310.26\,\lambda + 1384.9\lambda^2)$$
$$-0.130947(642.048 - 377.5G + 213.095G^2) - 333.197 \quad (4.13)$$

The proposed regression models are new and unique to the engine performance curves and can predict the performance and emission characteristics of the engine or any other SI engine with similar performance curves. The model is simple and powerful and the approach can enable researchers to replicate and develop generalized models for all possible performance curves.

4.4 Validation of Performance and Emission Models

The model validation approach employed in this research involved the splitting of data into two separate sets of engine operating parameters. The first set of data in Table 4.7 to Table 4.13 was used for models calibration, while the second set of data in Table 4.14 and Table 4.15 was used for model validation. The performances of the models were first evaluated by comparing model predicted and actual measurements of engine output parameters namely; SFC, maximum torque, brake power, CO, CO_2, and HC in Table 4.15. The performances of the models were evaluated in terms of precision and accuracy which is indicated by the percentage error (Neter and Kutner, 1996; Chunxia, 2007).

Table 4.14: Effect of engine input variables on engine parameters

Engine operating variables				Engine operating parameters					
Speed (rpm)	CA (^oBTDC)	Lambda	Spark plug gap (mm)	SFC (g/kWh) Obs.	SFC (g/kWh) Pred.	Torque (Nm) Obs.	Torque (Nm) Pred.	Power (kW) Obs.	Power (kW) Pred.
1200	10	0.95	0.8	478	483	154	154	19	15
1500	10	0.95	0.8	446	440	168	168	26	29
2500	10	0.95	0.8	409	406	187	186	49	49
3500	10	0.95	0.8	436	439	173	161	61	59
4500	10	0.95	0.8	531	558	129	94	66	54
-	7	0.95	0.8	423	413	174	182	36	39
-	12	0.95	0.8	412	405	183	184	52	52
-	17	0.95	0.8	431	415	173	176	56	55
-	22	0.95	0.8	446	434	158	162	58	57
-	27	0.95	0.8	457	460	139	142	57	58
2000	10	0.85	0.8	472	486	182	180	38	38
2000	10	0.88	0.8	451	462	183	181	39	38
2000	10	0.93	0.8	428	429	186	182	38	39
2000	10	0.97	0.8	402	407	185	182	38	39
2000	10	1.15	0.8	366	370	175	171	36	36
2000	10	1.25	0.8	386	389	168	166	35	35

In Fig 4.9 to Fig. 4.11 the results of predicted and the observed values of SFC, maximum torque and brake power are shown. The SFC characteristic was almost uniform through the speed range. The model predicted well the maximum torque and brake power at lower engine speed, however, at high engine speed, the predicted values were lower than the observed values. This conforms to the study done by Sitthiracha (2006). But when considering percentage errors, overall errors were between -1.0% and 5.0% for SFC, -37.0% and 0.0% for max torque and – 27.0% and 10% for brake power. Although the percentage error greater than 10% may be considered high, the average percentage errors for each of the three parameters were less than 10%; an indication that the model predicted well engine performance parameters.

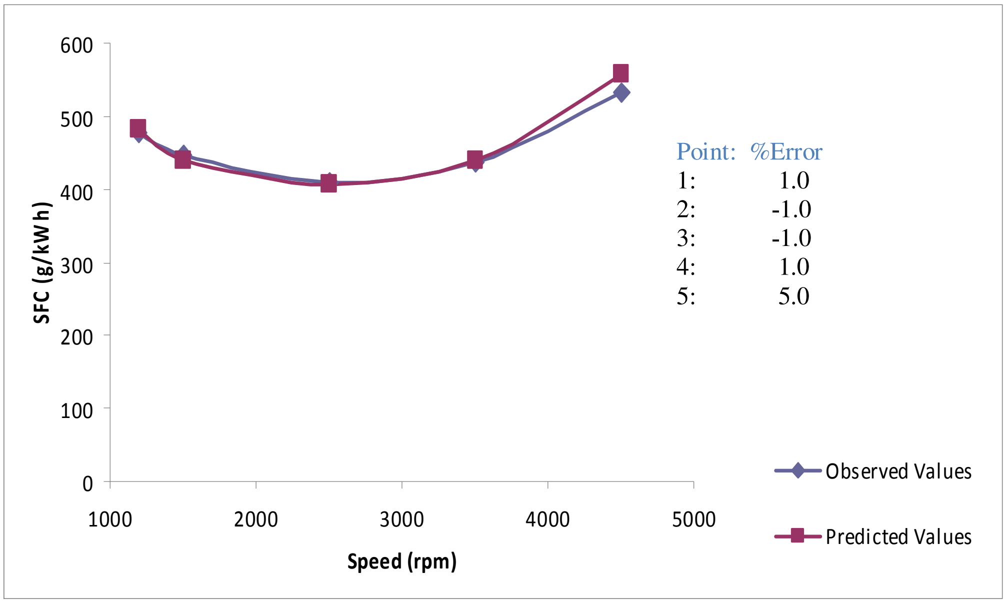

Fig. 4.9: Comparison between predicted and observed SFC versus speed

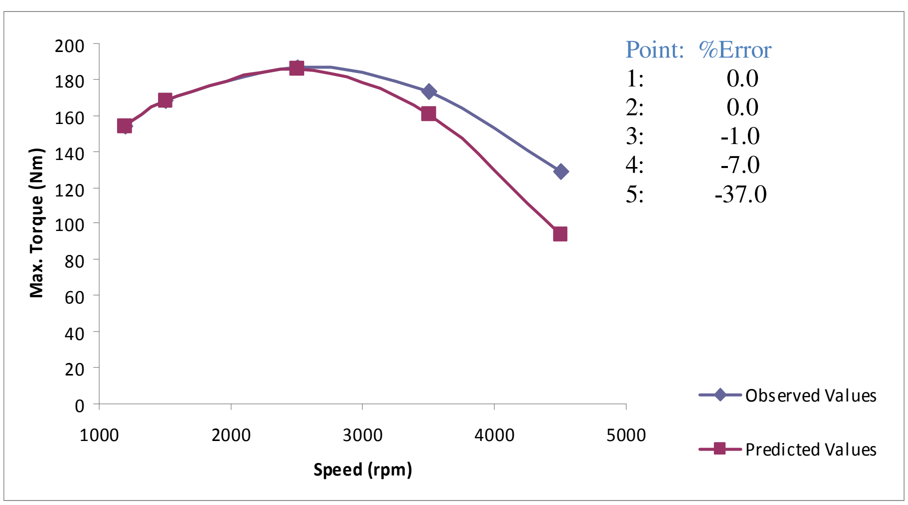

Fig. 4.10: Comparison between predicted and observed max Torque versus speed

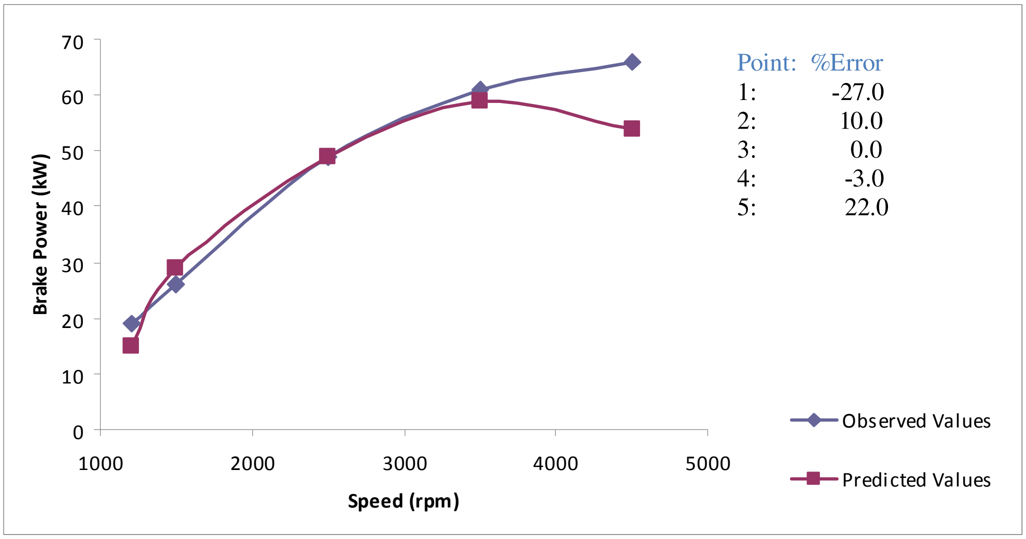

Fig. 4.11: Comparison between predicted and observed brake power versus speed

According to Fig. 4.12, the trend of observed and the predicted values of SFC as a function of crank angle was the same. However, predicted SFC was under-estimated as compared to observed values below 25^o BTDC. The model over-estimated predicted torque at lower values of crank angle below 15^o BTDC but the characteristics were almost similar over the remaining range of crank angle as seen in Fig. 4.13. The same observation was made in the case of predicted and observed brake power in Fig. 4.14. The percentage errors were between – 4.0% and 1.0% for SFC, 1.0% and 4.0% for maximum torque, -2.0% and 8.0% for brake power. The percentage errors were within the acceptable limits for any good model as stated by Su *et al* (2002).

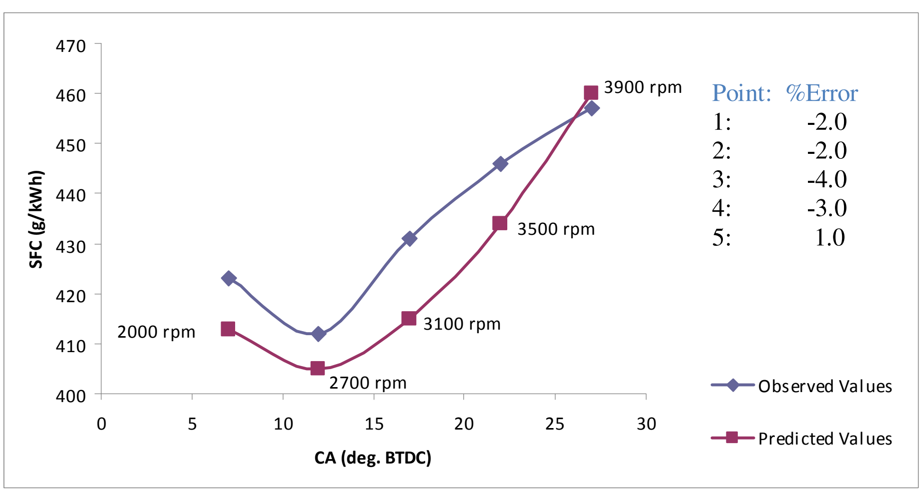

Fig. 4.12: Comparison between predicted and observed SFC versus crank angle

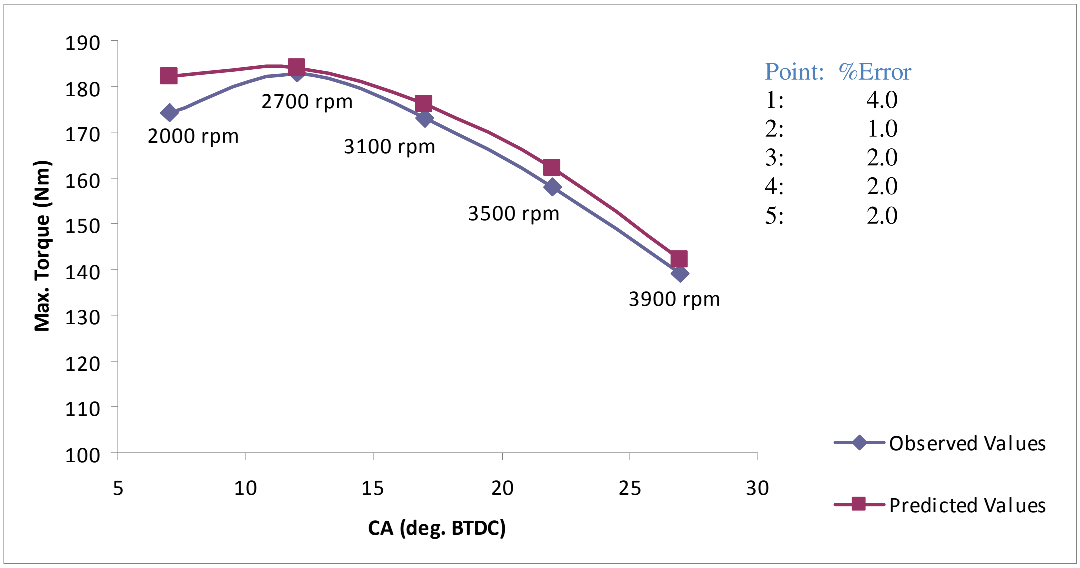

Fig. 4.13: Comparison between predicted and observed max Torque versus crank angle

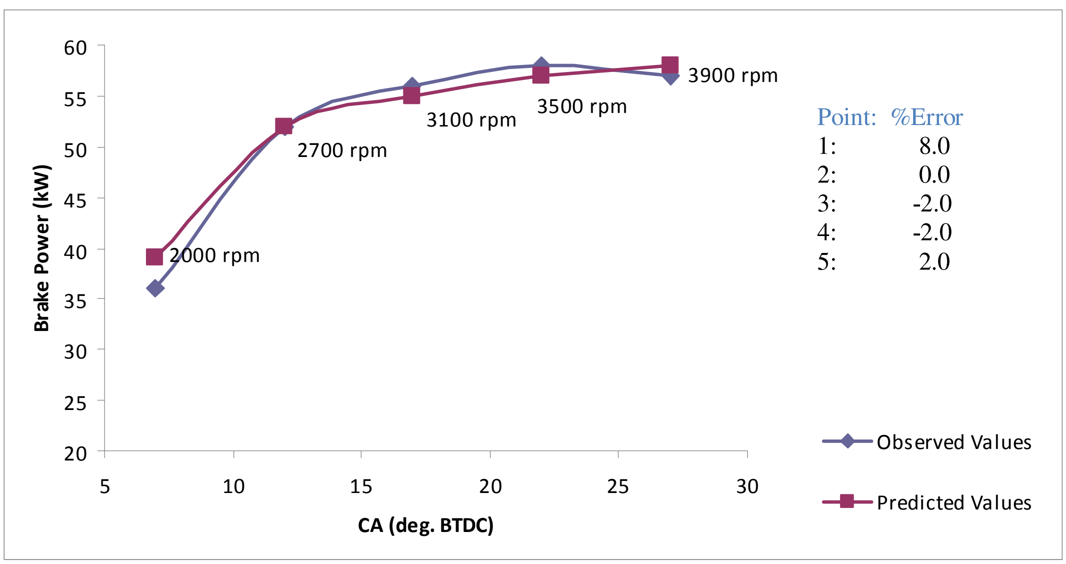

Fig. 4.14: Comparison between predicted and observed power versus crank angle

In Fig. 4.15, predicted and observed curves of SFC as a function of lambda were similar, above 0.85 lambda. Same observation was made in predicted and observed torque in Fig. 4.16 with predicted torque being lower than the observed torque over the whole range of lambda values. The predicted and the observed brake power were the same at lambda of 0.85. However, the model over-estimated the predicted values over the remaining range of lambda values. The percentage errors were between -2.0% and 3.0% for the three parameters showing good predictive trend. However, the variation in both predicted and observed values can be attributed to conditions like running the engine for a long period under a rich mixture that may result in carbon formation inside the combustion chamber which has been known to affect combustion processes inside the cylinder. Spark plug fouling can also occur under rich mixture affecting spark discharge, and also spark plug condition may change with time which will also affect the accuracy of the results.

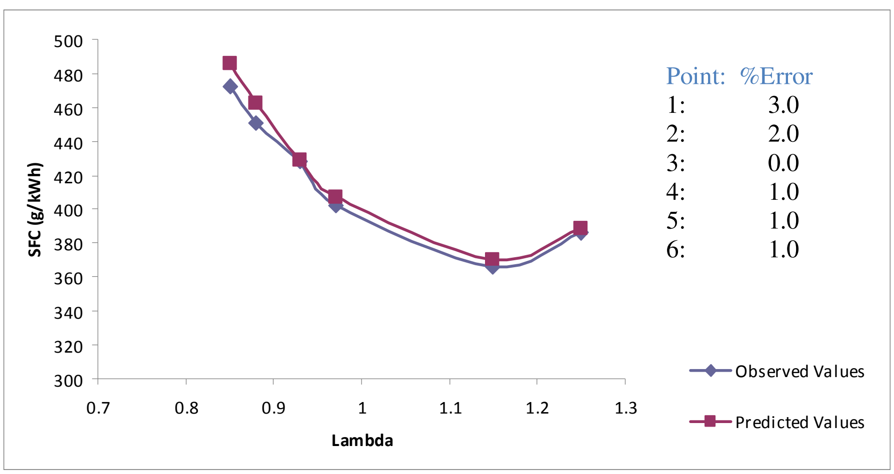

Fig. 4.15: Comparison between predicted and observed SFC versus lambda

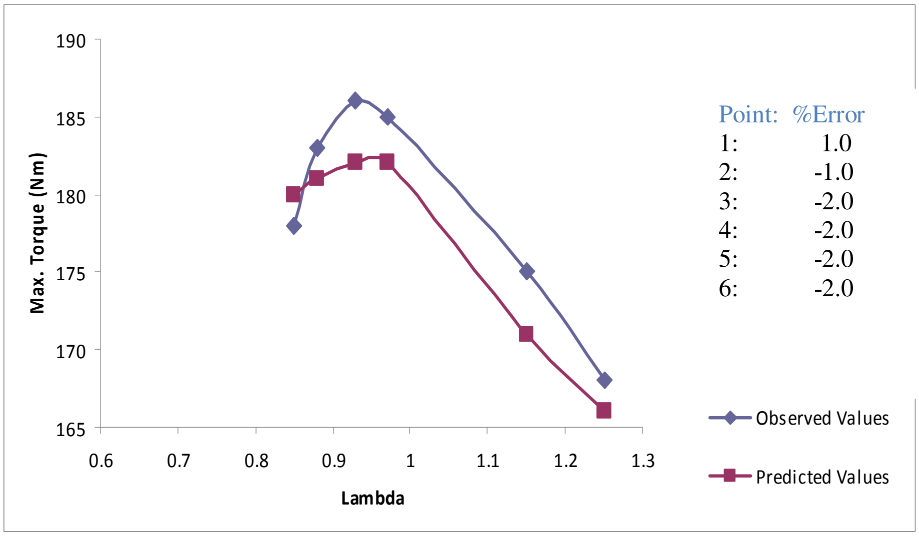

Fig. 4.16: Comparison between predicted and observed max Torque versus lambda

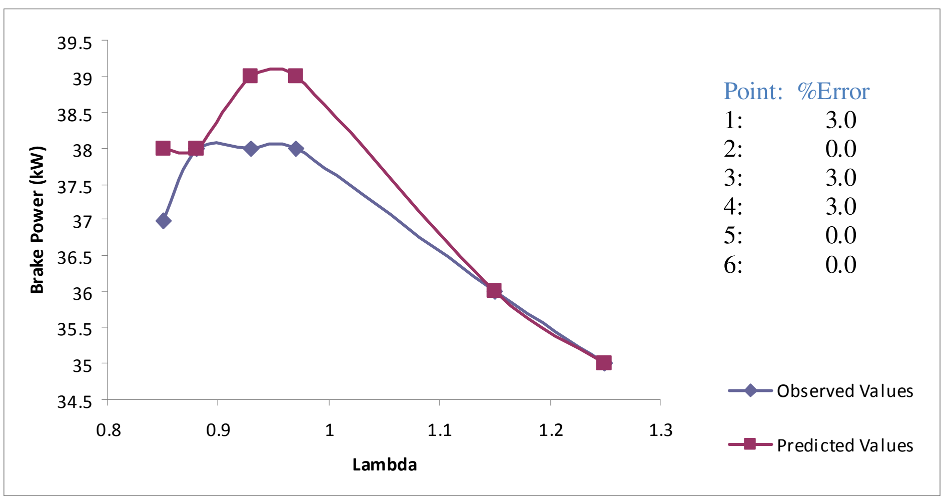

Fig. 4.17: Comparison between predicted and observed brake power versus lambda

According to the results, the models can predict the engine performance well. The high percentage errors realized can be associated with inadequate information like effects of frictional losses and volumetric efficiency associated with engine speed, pressure and temperature changes and fraction of burnt mass in the cylinders associated with crank angle position; burning rate and flame propagation associated with lambda (Heywood, 1998; Stone, 1999; Guzzella and Onder, 2004).

Both predicted and observed data from the analysis of CO, CO_2 and HC are presented in Table 4.15. In Figure 4.18, the model over-estimated the CO at crank angle less than 20^o BTDC and under-estimated at advance angle of over 20^o BTDC. The percentage errors were between -4.0% and 2.0%. By comparing results of predicted and observed CO, it can be observed that with the proposed modeling approach, the percentage error values were within acceptable limits for most of the exhaust emission models. However, the differences realized might have been caused by the assumption of a chemical kinetically controlled CO formation process described by Ramos (1989), which in turn requires a detailed description of burned gases thermal field being modeled physically with the previous multi-zone model. This condition was not factored in this model and this contributed to the variation of the values of predicted and observed CO. This is because the formation and decomposition of carbon monoxide is mainly controlled by chemical reactions, though influenced by high temperatures as discussed by Arsie (1998) and Jobson *et al* (1996).

Table 4.15: Effect of engine operating variables on exhaust emission

Engine operating variables			**Exhaust emission**					
CA	Lambda	Spark plug	CO (% vol.)		CO_2 (% vol.)		HC (ppm)	
(^{o}BTDC)		gap (mm)	Obs.	Pred.	Obs.	Pred.	Obs.	Pred.
10	0.85	0.8	6.9	6.2	8.1	8.6	597	575
10	0.90	0.8	6.1	5.9	9.4	9.1	535	531
10	0.97	0.8	4.8	5.4	10.4	9.8	459	480
10	1.15	0.8	3.8	4.5	10.3	10.7	435	413
10	1.25	0.8	3.8	4.2	10.2	10.8	452	414
12	0.95	0.8	5.3	5.3	10.1	9.8	486	491
17	0.95	0.8	4.8	4.9	10.3	10.1	482	488
22	0.95	0.8	4.7	4.6	10.2	10.2	485	489
27	0.95	0.8	4.6	4.6	10.1	10.1	496	494
33	0.95	0.8	4.7	4.8	9.9	9.6	504	507

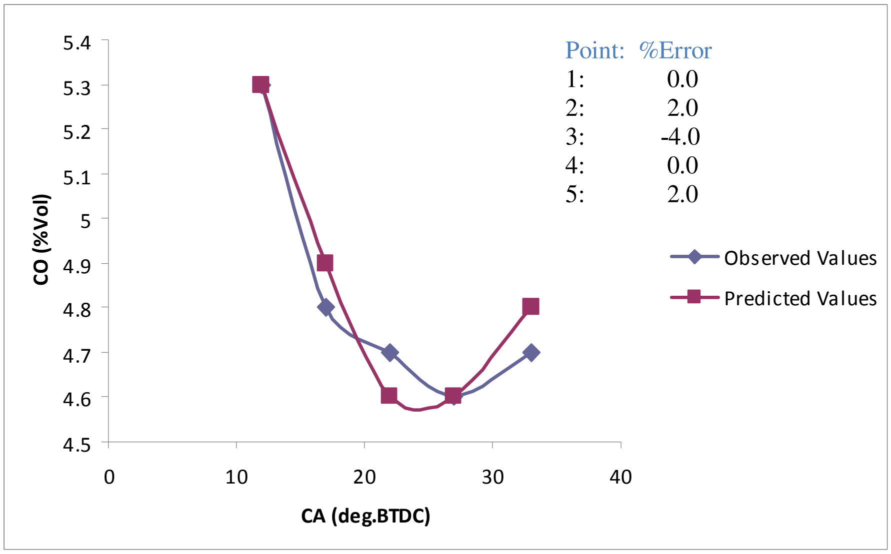

Fig. 4.18: Comparison between predicted and observed CO versus crank angle

Predicted and observed values of CO_2 are presented in Figure 4.19. The CO_2 model under-estimated the predicted values except 22^o BTDC and 27^o BTDC which gave similar values for both predicted and observed values. Percentage errors were between -3.0% and 0.0%. The formation of CO_2 depends on complete combustion and the amount of fuel available for combustion. The CO_2 levels are normally influence by both the CO and HC emission levels through combustion processes and therefore the factors that influence CO and HC formation were expected to affect the CO_2 (Jobson *et al*, 1996).

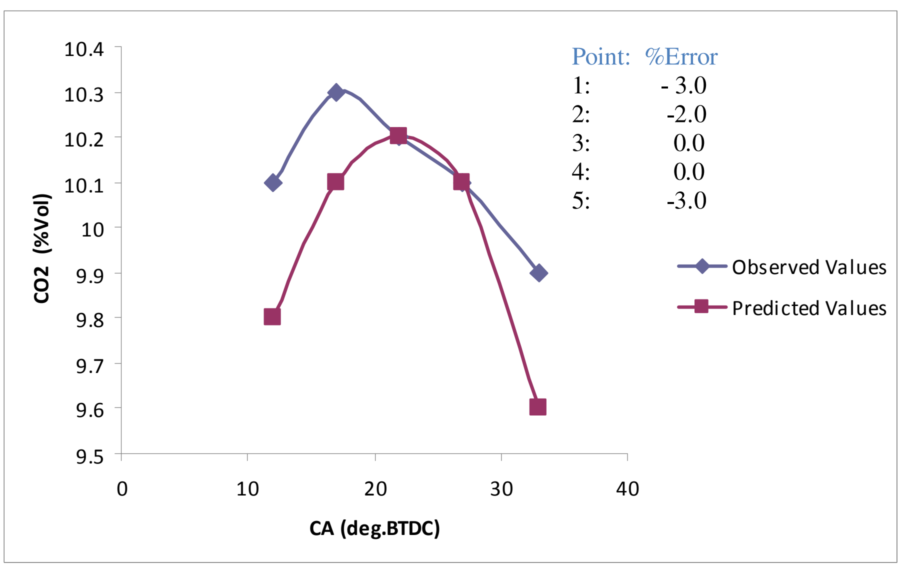

Fig. 4.19: Comparison between predicted and observed CO_2 versus crank angle

Both predicted and measured HC data from the model analyses are presented in Figure 4.20. The model overestimated the predicted HC as compared to the observed HC at the crank angles (CA) less than 20° BTDC and at crank angle over 30° BTDC. Percentage errors were between -0.4 % and 1.0 % which showed good predictive trend. It is worth stating that the mechanisms which mainly influence the HC emissions are the absorption and deabsorption of unburned hydrocarbons into the wall wetting oil layer, the inflow and outflow from the crevices and the post-flame oxidation as explained by Schramm (1990), Heywood (1998), and Stone (1999). These factors are normally associated with crank angle which influence cylinder pressure. The conditions were not considered in the model development as it was not possible to access inside the cylinder while the engine was in operation. This influenced reliability of the HC model developed in this study (Zuo *et al*, 2008; Guzella and Onder, 2004; Jobson *et al*, 1996).

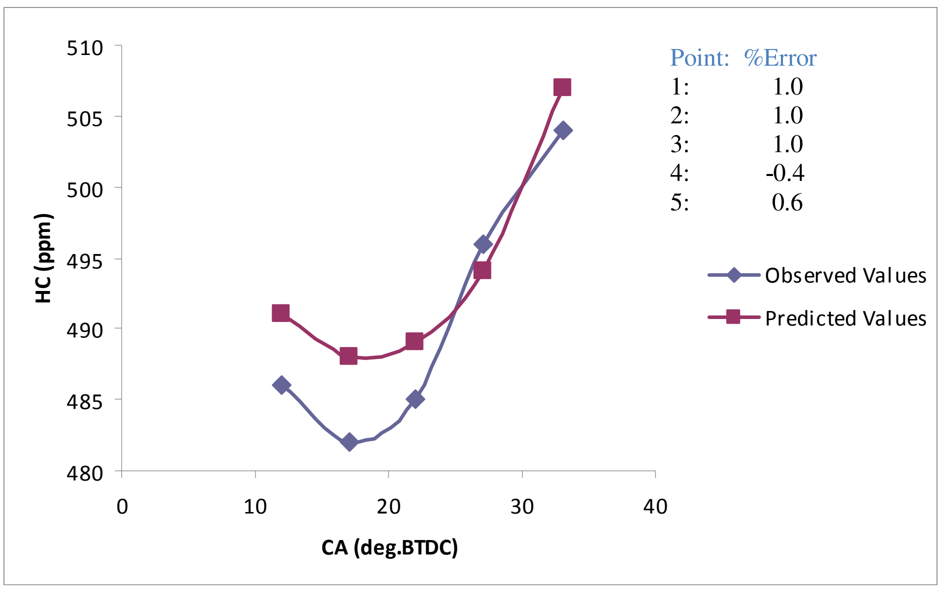

Fig. 4.20: Comparison between predicted and observed HC versus crank angle

Fig. 4.21, Fig. 4.22 and Fig. 4.23 show the prediction and observed CO, CO_2 and HC as a function of lambda. The model under-estimated the CO at lower lambda of less than 0.9 while over-estimating at lean mixture as shown in Fig. 4.21. According to Fig. 4.22, there was over-estimation of CO_2 at lower and high lambda values. The HC model predicted well HC at rich mixtures, while there was over-estimation between lambda values of 0.9 and 1.1

and under-estimation at lean mixtures. The percentage errors were between -11.0 % and 16.0% for CO, -6.0% and 6.0% for CO_2 and -5.0% and 4.0% for HC. Apart from the values for CO, the rest of the values conform to the studies by Atkison *et al* (1998) and Su *et al* (2002). The CO model enhancement should be pursued through a more detailed description of model parameters' dependencies with respect to engine operating conditions in order to minimize the percentage error.

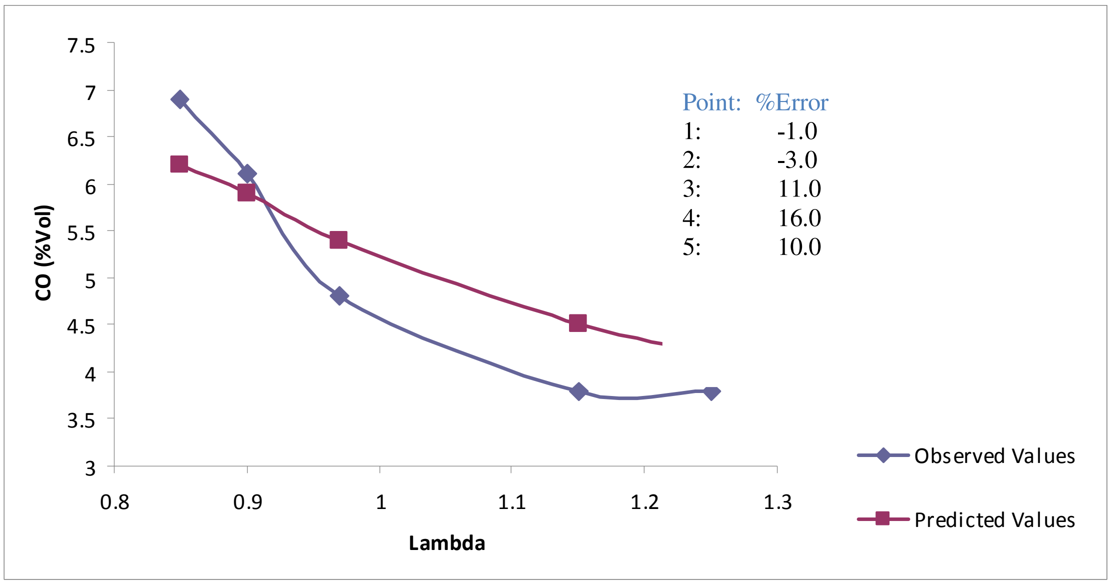

Fig. 4.21: Comparison between predicted and observed CO versus lambda

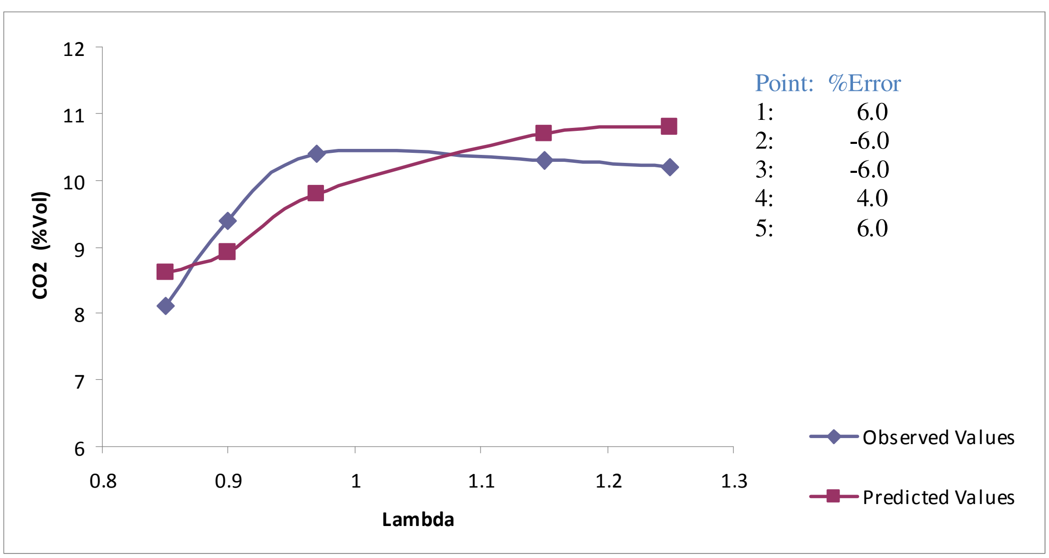

Fig. 4.22: Comparison between predicted and observed CO_2 versus lambda

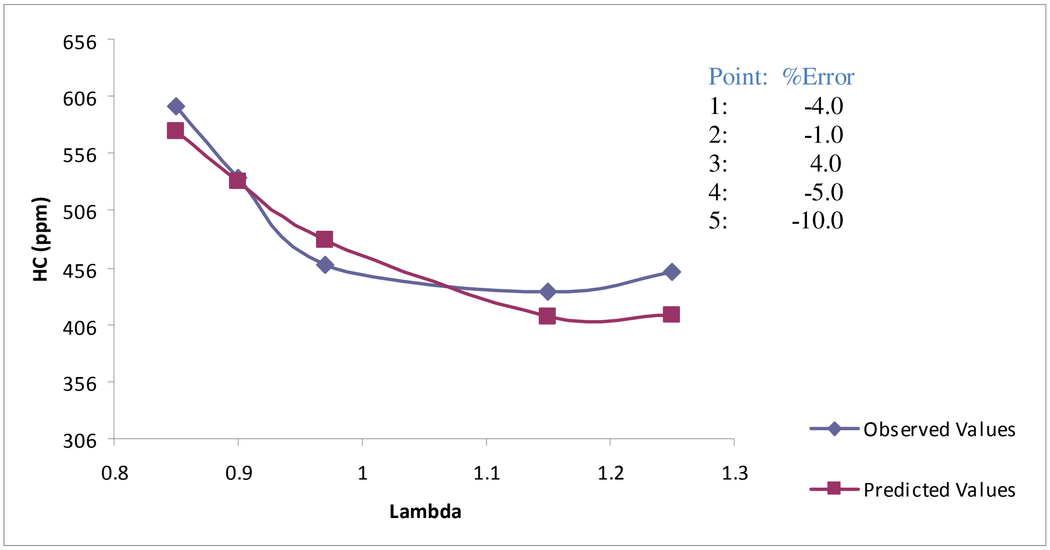

Fig. 4.23: Comparison between predicted and observed HC versus lambda

4.5 Optimization of Engine Performance and Emission Models

To determine optimal engine operating conditions, the SFC model was optimized against maximum torque, brake power, carbon monoxide, carbon dioxide and hydrocarbon models as constraints. The maximum attainable torque was obtained by differentiating equations 4.1b, 4.2b, 4.3b, and 4.4b with respect to S, θ, λ, and G, respectively, and equating it to zero. The obtained values of S, θ, λ and G when substituted in the superimposed model equation (4.9) gave a value of 185 Nm. The constraint function for maximum torque was taken as ϕ (S, θ, λ, G) = 185 Nm. The maximum power was obtained by differentiating equations (4.1c, 4.2c, 4.3c, and 4.4c with respect to S, θ, λ and G and equating to zero. The resulting values of S, θ, λ and G were substituted in the superimposed model in equation (4.10) giving a value of 63 kW. Also, maximum attainable CO_2 was obtained by differentiating equation 4.5b, 4.6b and 4.7b and equating it to zero. The resulting value of θ, λ and G when substituted in equation (4.12) gave a CO_2 value of 10.8 (% vol). Minimum HC was obtained by differentiating equation (4.7) with respect to θ, λ and G and equating to zero. Substituting the values of θ, λ and G in the superimposed model in equation (4.13) gave a minimum value of 405 ppm HC. Minimum attainable CO was obtained by differentiating equations 4.5a, 4.6a and 4.7a. However, the relation given by equation 4.6a is exponential in nature and for it to be optimized it had to be linearized before differentiating using the

Taylor's series by expanding exp $(-\lambda)$ in the neighborhood of 1 which is the ideal value for lambda. Substituting the obtained values in the superimposed model in equation (4.11) gave minimum CO value of 4.1 (% vol).

The determined engine performance and emission parameters were substituted in equation 3.5 to solve for Langragian multipliers $\gamma'_1, \gamma'_2, \gamma'_3, \gamma'_4$ and γ'_5 using *Mathematica*® software Version 5.2. The resulting values of γ'_1= -3.8607, $\gamma'_2 = 0.4$, $\gamma'_3 = -0.013$, $\gamma'_4 = 0.13$ and $\gamma'_5 = 0.015$ gave the values of S, θ, λ and G which were within the experimental range. Using the value of γ', the optimal values of S, θ, λ and G were found to be 2839 rpm, 16° BTDC, 1.05 and 0.8 mm respectively. The independent variables gave the optimal values of SFC = 368 g/kWh, max torque = 179 Nm, bP = 58 kW, HC = 433 ppm, CO_2 = 10.8% vol. and CO = 4.4% vol. The same initial setting of θ, λ and G gave optimal brake power of 58 kW at 3690 rpm. Although maximum torque and brake power were reduced by 1.6% and 13.4% respectively, SFC, CO and HC were reduced by 9.6%, 15.4% and 8.5%, respectively as shown in Table 4.16. From the results the choice of merit function is important. In this case, SFC, CO and HC were reduced at expense of maximum torque and brake power. High values of unburned HC and CO in relation to their reference values were found in the engine. The HC and CO levels in the exhaust directly affect SFC and therefore the objective function should contain the parameter. The optimal values were different from the original tuning values and the adjustments were meant to take care of the current engine conditions.

Table 4.16: Comparison of initial engine tuning and optimal parameters

Initial settings	**Optimal settings**
S = 2500 rpm	S = 2839 rpm
θ = 10° BTDC	θ = 16° BTDC
λ = 0.95	λ = 1.05
G = 0.8 mm	G = 0.8 mm
SFC = 407 g/kWh	SFC = 368 g/kWh
T_{max} = 182 Nm	T_{max} = 179 Nm
bP_{max} = 67 kW (at 4000 rpm)	bP_{max} = 58 kW (at 3690 rpm)
CO = 5.2% vol.	CO = 4.4% vol
CO_2 = 9.8% vol.	CO_2 = 10.9% vol.
HC = 473 ppm	HC = 433 ppm

Studies have shown that engine torque at given engine speed and intake manifold pressure conditions vary as spark timing is varied relative to top dead centre. If the start of combustion process is progressively advanced before top dead centre, the compressive stroke work transfer increases and if the end of combustion is progressively delayed by retarding the spark timing, the peak pressure occurs later in the expansion stroke and is reduced. The optimal setting depends on the rate of flame development and propagation, the length of flame travelled path across the combustion chamber, and the details of flame termination process after it reaches the wall. These depend on engine design and operating conditions and the properties of fuel, air, burned gas mixture, and the intensity of the spark discharged (Heywood, 1998; Stone, 1999; Guzzella and Onder, 2004).

4.6 Sensitivity Analysis of Performance and Emission Models

The sensitivity analysis was done by determining which of the independent variables influenced the models' output variances most. The optimal independent variable values used from the Langragian functions were; S = 2839 rpm, $\theta = 16^o$ BTDC, $\lambda = 1.05$, and G = 0.8. The optimal values were changed by ±10 % and the resulting SFC and maximum torque were computed as shown in Table 4.17. From the table, it can be observed that λ was the most sensitive variable because when it was reduced by 10%, it caused the largest increase of 10.9% in SFC. Lambda also reduced maximum torque by 4.47% when it was reduced by 10%. There was no change in maximum torque when crank angle was changed by 10%, while there was a marginal change of ±0.27% in SFC when crank angle was changed by ±10%.

Table 4.17: Sensitivity of engine variables on SFC and Torque

Engine variables	limits	% change in SFC	% change in Torque
Speed (rpm)	$S_H = 3123$	+2.72	-3.91
	$S_L = 2555$	-0.54	+2.23
CA (^{o}BTDC)	$\theta_H = 18$	-0.27	0.0
	$\theta_L = 14$	+0.27	0.0
Lambda (λ)	$\lambda_H = 1.16$	-2.72	-4.47
	$\lambda_L = 0.95$	+10.9	+1.68
Spark plug gap (mm)	$G_H = 0.9$	+0.54	+0.56
	$G_L = 0.7$	+ 0.54	-1.12

The same observation was made in Table 4.18 when lambda was increased by 10% , brake power reduced by 3.45% Engine speed and spark plug gap reduced brake power by 1.72 % when they were changed by ±10 % implying lambda was the most sensitive of the four variables.

Table 4.18: Sensitivity of engine variables on SFC and brake power

Engine variables	limits	% change in bP
Speed (rpm)	$S_H = 4059$	-1.72
	$S_L = 3321$	-1.72
CA (^{o} BTDC)	$\theta_H = 18$	0.0
	$\theta_L = 14$	0.0
Lambda (λ)	$\lambda_H = 1.16$	-3.45
	$\lambda_L = 0.95$	0.0
Spark plug gap (mm)	$G_H = 0.9$	-1.72
	$G_L = 0.7$	- 1.72

Table 4.19 shows the sensitivity analysis of input variables on CO, CO_2 and HC formation. Lambda was found to be more sensitive as a reduction of lambda by 10% from the optimal value increased the formation of CO and HC by 13.6% and 12.4%, respectively, while reducing the CO_2 levels by 8.33%. Also, increasing the value by 10% towards the lean mixture caused an increase of HC formation by 3.9%. The other two variables had similar effects when changed by ± 10%. Changing crank angle by ±10% affected CO formation by ±4.55% while the effect on CO_2 was marginal and no effect in HC.

Table 4.19: Sensitivity of engine variables on SFC and CO, CO_2 and HC optimization

Engine variables	limits	% change in CO	% change in CO_2	% change in HC
CA (^{o} BTDC)	$\theta_H = 18$	-4.55	+0.93	0.0
	$\theta_L = 14$	+4.55	-1.85	+0.46
Lambda (λ)	$\lambda_H = 1.16$	-13.6	+1.85	-6.45
	$\lambda_L = 0.95$	+13.6	-8.33	+12.4
Spark plug gap (mm)	$G_H = 0.9$	+0.0	+0.0	0.0
	$G_L = 0.7$	+2.27	-2.78	0.0

Studies have shown that given a constant set of engine conditions, an increase in lambda results in increased pressure at ignition which has been attributed to an increase in the ratio of specific heats (gamma ratio) for air fuel mixture and an increase in gas pressure resulting in a rise in break down voltage (Parshley, 1997; NGK, 1991). Changes in lambda and therefore in break down voltage leads to substantial changes in overall wave form. Given a constant ignition energy, an increase in breakdown voltage results in more energy available for following phases of spark (arc and glow discharge). Changes in lambda would be expected to influence both breakdown voltage and the voltage characteristics which affect the engine performance and emission characteristics.

4.7 Computer Program for Performance and Emission Prediction Models

From the model equations (4.8 to 4.13), a Visual Basic Program was developed to predict engine performance and emissions from the engine input variables. The system uses procedure-driven control operations. In the system, operations wait for user input whenever they need data from either a user or from a database. When running the models the performance and emission models variance, it was began by inserting the engine speed, S (rpm) in the "S" column of the spread sheet, crank angle, θ (^{o} BTDC) in the "θ" column, lambda, λ in "λ" column and spark plug gap, G (mm) in "G" column. The source code for the computer program is presented in Appendix III.

4.7.1 System Capability List

The system can be run by selecting the following menu to enable one enter the engine input variables;

- Select parameter to measure( )
- Enter observed values( )
- Compute predicted values( )
- Save models( )
- Display on list( )
- Print graph( )

The flow chart for the operation and execution is shown in Fig. 4.24.

Flowchart

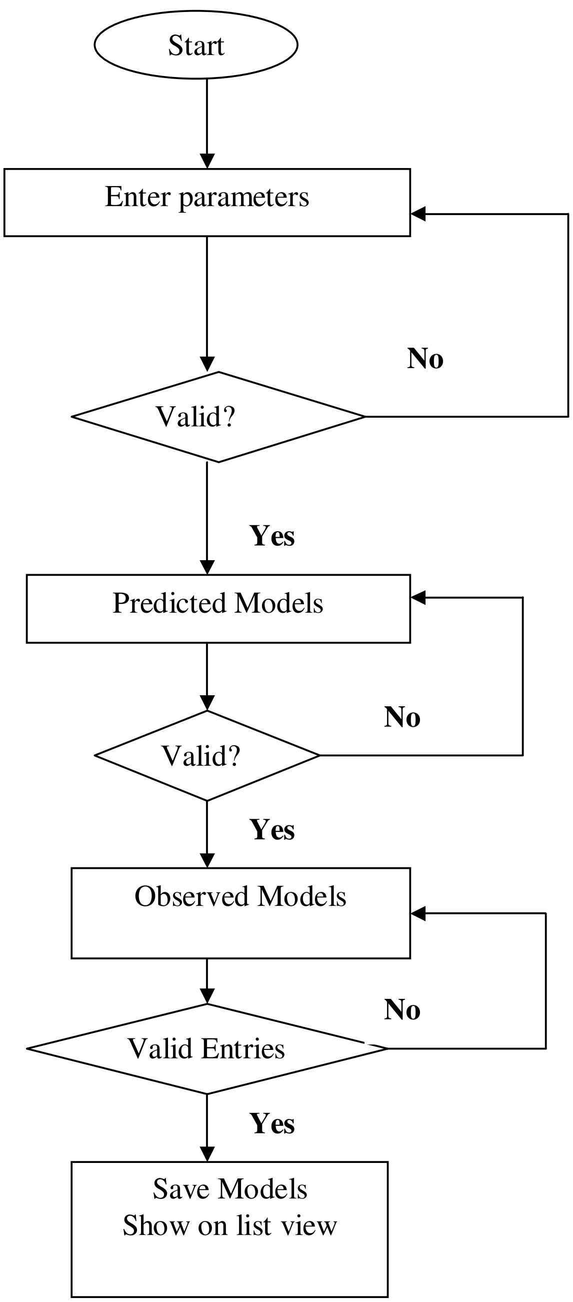

Fig.4.24 System flow chart

4.7.2 Analysis Subsystem Capability List

The system developed is capable of performing the following:

a. Compare Predicted values against Observed values for SFC

b. Compare Predicted values against Observed values for Tmax

c. Compare Predicted values against Observed values for bP

d. Compare Predicted values against Observed values for CO

e. Compare Predicted values against Observed values for CO2

f. Compare Predicted values against Observed values for HC

g. Print a graphical model for SFC against Speed, Theta and Lambda

h. Print a graphical model for T_{max} against Speed, Theta and Lambda

i. Print a graphical model for bP against Speed, Theta and Lambda

j. Print a graphical model for bP against Theta and Lambda

k. Print a graphical model for CO against Theta and Lambda

l. Print a graphical model for CO_2 against Theta and Lambda

m. Print a graphical model for HC against Theta and Lambda

n. Composite analysis

4.7.3 Database Management

This system is based on a Relational Database Management System (RDBMS). The server used is MySQL because of its capabilities to work on the network especially on future advancements (currently the system is meant to be a desktop application).

Database requirements shall include:

MySQL server version 5

MySQL Administrator

MySQL database migration tool kit

MySQL Query browser

The relational structures used are well normalized to ensure persistent data management.

4.7.4 Database Design and Hardware Requirements

The program developed requires the following for installation;

A minimum 800 MHz processor

Minimum of 256 MB RAM

Minimum disk space (for database storage) of about 2GB

4.7.5 Platform and Technology Requirements

The program requires the following operating systems;

Windows 2000/Xp/2000/Me/Vista

Visual studio 2005(.NET)

Optimum operations are however achieved with Windows XP (having service pack2) or Windows Vista

4.7.6 User Interface Design

Some of the interfaces used in this system are shown below.

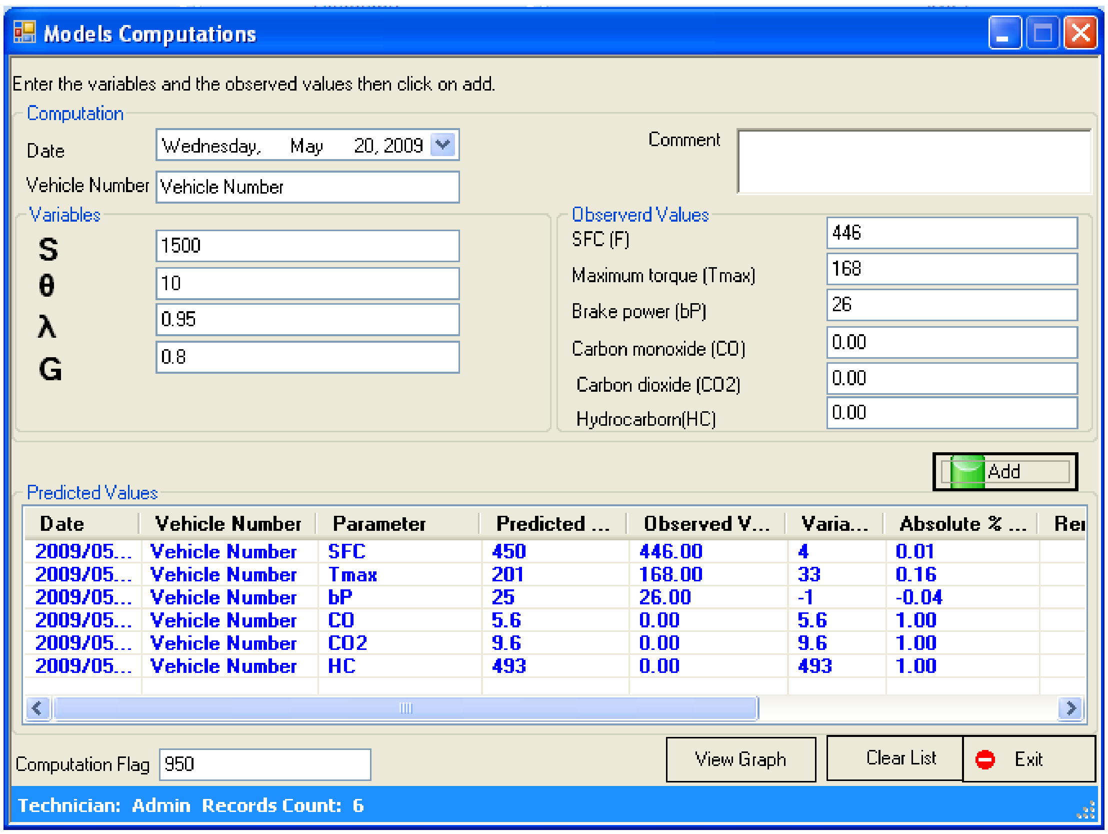

Fig.4.25 Computations Screen

The computer program developed uses the in the model equations and is capable of calculating predicted specific fuel consumption (SFC), maximum torque (T_{max}), brake power (bP), carbon monoxide (CO), carbon monoxide (CO_2) and hydrocarbon (HC) for given engine parameters. It can also plot graphs and calculate percentage errors when observed values of engine performance and emission parameters are given for each engine input variables.

CHAPTER FIVE

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

The overall emission test results showed that 69.5% of petrol vehicles failed the emissions tests. The mean vehicle usage ranged between 14327.9 km/yr and 19640.2 km/yr and the lowest compression pressure of 6.8 bar, was recorded in the non-catalytic vehicles manufactured before 1986. The average spark plug gap for non-catalytic vehicles was 0.9 mm which was higher than the recommended gap of 0.8 mm for most of the vehicles in this category. They also operated at a rich mixture. The *Chi*-square tests for all categories of vehicles showed that there were statistically significant differences between the measured and standard values of CO, HC and CO_2.

Logistic regression model showed that the coefficients of the engine parameters namely; vehicle usage, compression pressure, ignition angle, engine speed, spark plug gap, and lambda were statistically significant in contributing to the probability of failing or passing of a vehicle. The assessment of the logistic model showed good fit as fitted model minimized the AIC and SC while maximizing the likelihood ratio relative to the null model. The null hypothesis of no significant regression was strongly rejected for all categories of vehicles at 5% significance level.

From engine test bed, a maximum output power of 67 kW at 4000 rpm was obtained from the experiment, which was about 61% of the rated power. Also, a maximum torque of 182 Nm was recorded at engine speed of 2500 rpm. The value was 82.7% of the maximum rated torque. The speed of 5400 rpm for maximum power output was also not achieved. In general, there were significant changes in engine performance and emission curves over the whole range of the input variables with specific points giving either maximum or minimum output values of SFC, torque, brake power, CO, CO_2, and HC. The minimum and the maximum points were found to change from time to time depending on the engine conditions. This gave a clear understanding of performance and emission of the engine in the field whose parameters were simulated in the engine test bed.

Analytical models of spark ignition engines to predict the performance and emissions were developed based on engine input parameters; namely speed, crank angle, excess air factor (lambda) and spark plug gap. These are parameters which are always checked to make sure that engines operate optimally. The models developed predicted well engine performance and emission characteristics with average percentage errors for all the models being less than 10 %. Some of the instances of high percentage errors might have been caused by some factors like cylinder pressure, temperature, heat transfer, crevices volume, and geometry of combustion chambers which were not considered.

It was found that optimal speed of 2839 rpm, crank angle of 16^o BTDC, excess air factor of 1.05 lambda and spark plug gap of 0.8 mm gave an optimum SFC value of 368 g/kWh, maximum torque of 179 Nm, CO of 4.4% vol., CO_2 of 10.9% vol., and HC of 433 pmm. Optimal bP was also achieved at 3690 rpm. Lambda was found to be the most sensitive of all the four variables as it caused the highest change in engine and emission values when it was changed by ±10%. The highest percentage of 15.03 SFC reduction was recorded when lambda was increased by 10%. The computer program developed is capable of computing predicted engine parameters and can draw performance and emission curves..

5.2 Recommendations

The recommendations made from the study are:

(i) There is need to raise awareness among policymakers and the general public about urban air pollution levels, damages, to specify and promote the roles that transport sector plays.

(ii) The proposed statistical models can be used to predict the performance and emission characteristics of the engine or any other SI engine with similar performance curves.

(iii) The models can be replicated by researchers to develop generalized models for all possible engine performance and emission curves.

5.3 Recommendations for Further Research

(i) There is need to include more test in engine performance parameters like cylinder pressure, temperature, heat transfer, crevices volume, and geometry of combustion chambers so as to understand fully the performance and emission characteristics of the engines.

(ii) There is need to study performance and emission characteristics of diesel engines in order to explain their contribution to air pollution in Nairobi City

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APPENDIX I

Principle of models Development

Principle of Least Squares Method for Sub-models

Considering engine speed (S) as a variable that affects specific fuel consumption (F), a function of F= ψ(S) with n set of observations such that $(S_{i,} F_i)$ for $i = 1,2,...,n$ in an experiment. The task was to fit a parabola and or exponential to the set of experimental data using the principles of least square method. This is because engine performance and emission curves are eithe parabolic or exponential in nature. From the function $F = \mu_o S^2 + \mu_1 S + \mu_2$ where $\mu_{o,}$ μ_1 and μ_2 were parameters to be determined, then for a given F_i, the expected value of F_i was given by $\mu_0 \, S_i^{\,2} + \mu_1 S_i + \mu_{2,}$ while the observed value was F_i. Then the residual e_i was given by;

e_i = observed value – expected value $= F_i - \left(\mu_0 S_i^{\,2} + \mu_1 S_i + \mu_2\right)$

Using the principles of least squares, E was taken as the sum of squares of residuals i.e.

$$E = \sum_{i=1}^{n} \left[F_i - \left(\mu_0 S_i^{\,2} + \mu_1 S_i + \mu_2\right)\right]^2 \qquad \text{(i)}$$

To find analytically the values of the parameters $\mu_{o,}$ μ_1 and μ_2 that minimize E, calculus was employed by partially differentiating with respect to μ_0, μ_1 and μ_2 and setting to zero.

That is for E to be minimum, the necessary conditions were $\dfrac{\partial E}{\partial \mu_0} = \dfrac{\partial E}{\partial \mu_1} = \dfrac{\partial E}{\partial \mu_2} = 0$

Taking $\dfrac{\partial E}{\partial \mu_0} = 0$

The implication is that;

$$\frac{\partial E}{\partial \mu_0} = 2\sum_{i=1}^{n} \left[F_i - \left(\mu_0 S_i^{\,2} + \mu_1 S_i + \mu_2\right)\right]\left(-S_i^{\,2}\right) = 0$$

From which,

$$\mu_0 \sum_{i=1}^{n} S_i^{\,4} + \mu_1 \sum_{i=1}^{n} S_i^{\,3} + \mu_2 \sum_{i=1}^{n} S_i^{\,2} = \sum_{i=1}^{n} S_i^{\,2} F_i \qquad \text{(ii)}$$

Similarly,

$$\frac{\partial E}{\partial \mu_1} = 2\sum_{i=1}^{n}\left[F_i - \left(\mu_0 S_i^{\ 2} + \mu_1 S_i + \mu_2\right)\right]\left(-S_i\right) = 0$$

$$\therefore\ \mu_0 \sum_{i=1}^{n} S_i^{\ 3} + \mu_1 \sum_{i=1}^{n} S_i^{\ 2} + \mu_2 \sum_{i=1}^{n} S_i = \sum_{i=1}^{n} S_i F_i \qquad \text{(iii)}$$

Finally,

$$\frac{\partial E}{\partial \mu_2} = 2\sum_{i=1}^{n}\left[F_i - \left(\mu_0 S_i^{\ 2} + \mu_1 S_i + \mu_2\right)\right](-1) = 0$$

$$\therefore\ \mu_0 \sum_{i=1}^{n} S_i^{\ 2} + \mu_1 \sum_{i=1}^{n} S_i + n\mu_2 = \sum_{i=1}^{n} F_i \qquad \text{(iv)}$$

Combining equations ii, iii and iv the normal equations become;

$$\mu_0 \sum_{i=1}^{n} S_i^{\ 4} + \mu_1 \sum_{i=1}^{n} S_i^{\ 3} + \mu_2 \sum_{i=1}^{n} S_i^{\ 2} = \sum_{i=1}^{n} S_i^{\ 2} F_i$$

$$\mu_0 \sum_{i=1}^{n} S_i^{\ 3} + \mu_1 \sum_{i=1}^{n} S_i^{\ 2} + \mu_2 \sum_{i=1}^{n} S_i = \sum_{i=1}^{n} S_i F_i$$

$$\mu_0 \sum_{i=1}^{n} S_i^{2} + \mu_1 \sum_{i=1}^{n} S_i + n\mu_2 = \sum_{i=1}^{n} F_i$$

This was expressed in matrix form as;

$$\begin{pmatrix} \sum_{i=1}^{n} S_i^4 & \sum_{i=1}^{n} S_i^3 & \sum_{i=1}^{n} S_i^2 \\ \sum_{i=1}^{n} S_i^3 & \sum_{i=1}^{n} S_i^2 & \sum_{i=1}^{n} S_i \\ \sum_{i=1}^{n} S_i^2 & \sum_{i=1}^{n} S_i & n \end{pmatrix} \times \begin{pmatrix} \mu_0 \\ \mu_1 \\ \mu_2 \end{pmatrix} = \begin{pmatrix} \sum_{i=1}^{n} S_i^2 F_i \\ \sum_{i=1}^{n} S_i F_i \\ \sum_{i=1}^{n} F_i \end{pmatrix} \qquad \text{(v)}$$

The elements of the coefficient matrix on the left hand side and the elements of the vector on the right hand side were constants which were derived from the respective tables.

The sum of the squares of the residuals was to be minimum as possible and it was calculated from the relation

$$\text{E}=\sum_{i=1}^{n} F_i^{\ 2} - \mu_0 \sum_{i=1}^{n} S_i^2 F_1 - \mu_1 \sum_{i=1}^{n} S_i F_i - \mu_2 \sum_{i=1}^{n} F_i$$

The matrix equation (v) was used by mathematica software to solve for the parabolic parameters using an in-built function "Fit [ ]". The syntax for a parabolic fit for sub-model was;

Fit [{{θ_1, F'_1}, {θ_2, F'_2}, …{ θ_n, F'_n}}, {1,θ,θ^2}, θ].

Principles of Least Squares Method for Superimposed Models

This was where curvilinear response models were considered and it was expected that the function;

$F^{'} = \psi(S, \theta, \lambda, G)$ was fitted

Earlier F=$\psi(S)$, $F = \psi(\theta), F = \psi(\lambda)$ and F=$\psi(G)$ were developed assuming a linear combination of the functions

$$F = \xi_0 + \xi_1\psi\ (S) + \xi_2\psi(\theta) + \xi_3\psi(\lambda) + \xi_4\psi(G) \qquad \text{(vi)}$$

Where, ξ_0, ξ_1, ξ_2, ξ_3 and ξ_4 were parameters to be determined

The sum of the residuals from least squares was

$$E = \sum_{i=1}^{n} [F_i - (\xi_0 + \xi_1\psi_i(S) + \xi_2\psi_i(\theta) + \xi_3\psi_i(\lambda) + \xi_4\psi_i(G))]^2 \qquad \text{(vii)}$$

For E to be minimum, the following condition was considered;

$$\frac{\partial Q'}{\partial \xi_0} = \frac{\partial Q'}{\partial \xi_1} = \frac{\partial Q'}{\partial \xi_2} = \frac{\partial Q'}{\partial \xi_3} = \frac{\partial Q'}{\partial \xi_4} = 0$$

Differentiating equation (3.11), with respect to ξ_0;

$$\frac{\partial Q'}{\partial \xi_0} = 2\sum_{i=1}^{n} [F_i - (\xi_0 + \xi_1\psi_i(S) + \xi_2\psi_i(\theta) + \xi_3\psi_i(\lambda) + \xi_4\psi_i(G))](-1) = 0$$

$$\therefore n\xi_0 + \xi_1\sum_{i=1}^{n}\psi_i(S) + \xi_2\sum_{i=1}^{n}\psi_i(\theta) + \xi_3\sum_{i=1}^{n}\psi_i(\lambda) + \xi_4\sum_{i=1}^{n}\psi_i(G) = \sum_{i=1}^{n}F_i \qquad \text{(viii)}$$

Also,

$$\frac{\partial Q'}{\partial \xi_1} = 2\sum_{i=1}^{n} [F_i - (\xi_0 + \xi_1\psi_i(S) + \xi_2\psi_i(\theta) + \xi_3\psi_i(\lambda) + \xi_4\psi_i(G))](-\psi_i(S)) = 0$$

$$\therefore \xi_0\sum_{i=1}^{n}\psi_i(S) + \xi_1\sum_{i=1}^{n}\psi_i^2(S) + \xi_2\sum_{i=1}^{n}\psi_i(\theta)\psi_i(S) + \xi_3\sum_{i=1}^{n}\psi_i(\lambda)\psi_i(S) + \xi_4\sum_{i=1}^{n}\psi_i(G)\psi_i(S)$$

$$= \sum_{i=1}^{n}\psi_i(S)F_i \qquad \text{(ix)}$$

Similarly from equation (vii);

$$\frac{\partial Q'}{\partial \xi_2} = 2\sum_{i=1}^{n}\left[F_i - (\xi_0 + \xi_1\psi_i(S) + \xi_2\psi_i(\theta) + \xi_3\psi_i(\lambda) + \xi_4\psi_i(G))\right](-\psi_i(\theta)) = 0$$

$$\therefore \xi_0\sum_{i=1}^{n}\psi_i(\theta) + \xi_i\sum_{i=1}^{n}\psi_i(S)\psi_i(\theta) + \quad \xi_2\sum_{i=1}^{n}\psi_i^{\;2}(\theta) + \xi_3\sum_{i=1}^{n}\psi_i(\lambda)\psi_i(\theta) + \xi_4\sum_{i=1}^{n}\psi_{\;i}(G)\psi_i(\theta)$$

$$= \sum_{i=1}^{n}F_i\psi_i(\theta) \tag{x}$$

Also;

$$\frac{\partial Q'}{\partial \xi_3} = 2\sum_{i=1}^{n}\left[F_i - (\xi_0 + \xi_1\psi_i(S) + \xi_2\psi_i(\theta) + \xi_3\psi_i(\lambda) + \xi_4\psi_i(G))\right](-\psi_i(\lambda)) = 0$$

$$\therefore \xi_0\sum_{i=1}^{n}\psi_i(\lambda) + \xi_i\sum_{i=1}^{n}\psi_i(S)\psi_i(\lambda) + \xi_2\sum_{i=1}^{n}\psi_i(\theta)\psi_i(\lambda) + \xi_3\sum_{i=1}^{n}\psi_i^{\;2}(\lambda) + \xi_4\sum_{i=1}^{n}\psi_i(\lambda)\psi_i(\lambda)$$

$$= \sum_{i=1}^{n}F_i\psi_i(\lambda) \tag{xi}$$

And finally,

$$\frac{\partial Q'}{\partial \xi_4} = 2\sum_{i=1}^{n}\left[F_i - (\xi_0 + \xi_1\psi_i(S) + \xi_2\psi_i(\theta) + \xi_3\psi_i(\lambda) + \xi_4\psi_i(G))\right](-\psi_i(G)) = 0$$

$$\therefore \xi_0\sum_{i=1}^{n}\psi_i(G) + \xi_1\sum_{i=1}^{n}\psi_i(S)\psi_i(G) + \xi_2\sum_{i=1}^{n}\psi_i(\theta)\psi_i(G) + \xi_3\sum_{i=1}^{n}\psi_{\;i}(\lambda)\psi_i(G) + \sum_{i=1}^{n}\psi_i^{\;2}(G)$$

$$= \sum_{i=1}^{n}F_i\psi_i(G) \tag{xii}$$

Therefore combining equations (viii), (ix), (x), (xi) and (xii) give the normal equations for the function

$F = \psi(S, \theta, \lambda, G)$

Dropping the subscripts and superscripts for simplicity and expressing the system in matrix form, we get

$$
\begin{pmatrix}
n & \Sigma\psi_i(S) & \Sigma\psi_i(\theta) & \Sigma\psi_i(\lambda) & \Sigma\psi_i(G) \\
\Sigma\psi_i(S) & \Sigma\psi^2_i(S) & \Sigma\psi_i(\theta)\,\psi_i(S) & \Sigma\psi_i(\lambda)\,\psi_i(S) & \Sigma\psi_i(G)\,\psi_i(S) \\
\Sigma\psi_i(\theta) & \Sigma\psi_i(S)\,\psi_i(\theta) & \Sigma\psi^2_i(\theta) & \Sigma\psi_i(\lambda)\,\psi_i(\theta) & \Sigma\psi_i(G)\,\psi_i(\theta) \\
\Sigma\psi_i(\lambda) & \Sigma\psi_i(S)\,\psi_i(\lambda) & \Sigma\psi_i(\theta)\,\psi_i(\lambda) & \Sigma\psi^2_i(\lambda) & \Sigma\psi_i(G)\,\psi_i(\lambda) \\
\Sigma\psi_i(G) & \Sigma\psi_i(S)\,\psi_i(G) & \Sigma\psi_i(\theta)\,\psi_i(G) & \Sigma\psi_i(\lambda)\,\psi_i(G) & \Sigma\psi^2_i(G)
\end{pmatrix}
\begin{pmatrix}
\xi_0 \\ \xi_1 \\ \xi_2 \\ \xi_3 \\ \xi_4
\end{pmatrix}
$$

$$
= \begin{pmatrix}
\Sigma F_i \\
\Sigma F_i\,\psi_i(S) \\
\Sigma F_i\,\psi_i(\theta) \\
\Sigma F_i\,\psi_i(\lambda) \\
\Sigma F_i\,\psi_i(G)
\end{pmatrix} \qquad \text{(xiii)}
$$

Equation (xiii) was solved for the suitable parameters $\xi_{0,}, \xi_{1,}, \xi_2, \xi_3$ and ξ_4. The elements of the coefficient matrix and that of the vector on the right hand side were obtained from the tables derived from observed data. Since $\psi(S)$, $\psi(\theta)$, $\psi(\lambda)$ and $\psi(G)$ had been developed, and the parameters $\xi_{0,}$, $\xi_{1,}$, ξ_2, ξ_3 and ξ_4 had been calculated, the superimposed model in equation was obtained. Similarly the normal equations were employed in determining the parameters for the other superimposed models. The assumption was that the coefficient matrix was invertible and therefore the solution exists and was unique.

APPENDIX II

Langragian Method for Optimization

The necessary conditions for $Z(S, \lambda, \theta, G)$ to have an optimal value was dZ = 0;

$$dZ' = \left(\frac{\partial \psi}{\partial \mathrm{s}} + \gamma \frac{\partial \Phi}{\partial \mathrm{s}}\right) dS + \left(\frac{\partial \psi}{\partial \lambda} + \gamma \frac{\partial \Phi}{\partial \lambda}\right) d\lambda + \left(\frac{\partial \psi}{\partial \theta} + \gamma \frac{\partial \Phi}{\partial \theta}\right) d\theta + \left(\frac{\partial \psi}{\partial \mathrm{G}} + \gamma \frac{\partial \Phi}{\partial \mathrm{G}}\right) dG = 0$$

$$\therefore \frac{\partial \psi}{\partial \mathrm{s}} + \gamma \frac{\partial \Phi}{\partial \mathrm{s}} = 0, \frac{\partial \psi}{\partial \lambda} + \gamma \frac{\partial \Phi}{\partial \lambda} = 0, \frac{\partial \psi}{\partial \theta} + \gamma \frac{\partial \Phi}{\partial \theta} = 0 \text{ and } \frac{\partial \psi}{\partial \mathrm{G}} + \gamma \frac{\partial \Phi}{\partial \mathrm{G}} = 0 \qquad \text{(xiv)}$$

Since the SFC model and Torque $_{(max)}$ model were of parabolic nature,

$$F' = \psi(S, \lambda, \theta, G) = \xi_0 + \xi_1(a_0 + a_1 S + a_2 S^2) + \xi_2(b_0 + b_1\theta + b_2\theta^2)$$
$$+\xi_3(c_0 + c_1\lambda + c_2\lambda^2) + \xi_4(d_0 + d_1 G + d_2 G^2) \qquad \text{(xva)}$$

$$T_{max} = \emptyset(S, \lambda, \theta, G) = \sigma_0 + \sigma_1(e_0 + e_1 S + e_2 S^2) + \sigma_2(f_0 + f_1\theta + f_2\theta^2)$$
$$+\sigma_3(g_0 + g_1\lambda + g_2\lambda^2) + \sigma_4(h_0 + h_1 G + h_2 G^2) \qquad \text{(xvb)}$$

Where $a_0,\ a_1,\ a_2,\ b_0,\ b_1,\ b_2,\ c_0,\ c_1,\ c_2,\ d_0,\ d_1,\ d_2,\ e_0,\ e_1,\ e_2,\ f_0,\ f_1,\ f_2,\ g_0,$ $g_1,\ g_2,\ h_0,\ h_1,\ h_2$ are sub-model parameters which were determined.

Then equation (xiv) became;

$$\xi_1 a_1 + 2\xi_1 a_2 S + \gamma(\sigma_1 e_1 + 2\sigma_1 e_2 S) = 0 \qquad \text{(xvia)}$$
$$\xi_2 b_1 + 2\xi_2 b_2\theta + \gamma(\sigma_2 f_1 + 2\sigma_2 f_2\theta) = 0 \qquad \text{(xvib)}$$
$$\xi_3 c_1 + 2\xi_3 c_2\lambda + \gamma(\sigma_3 g_1 + 2\sigma_3 g_2\lambda) = 0 \qquad \text{(xvic)}$$
$$\xi_4 d_1 + 2\xi_4 d_2 G + \gamma(\sigma_4 h_1 + 2\sigma_4 h_2 G) = 0 \qquad \text{(xvid)}$$

Solving equation (xv) for S, θ, λ, and G, then

$$s = \frac{-(\xi_1 a_1 + \gamma\sigma_1 e_1)}{2(\xi_1 a_2 + \gamma\sigma_1 e_2)} \qquad \text{(xviia)}$$

$$\theta = \frac{-(\xi_2 b_1 + \gamma\sigma_2 f_1)}{2(\xi_2 b_2 + \gamma\sigma_2 f_2)} \qquad \text{(xviib)}$$

$$\lambda = \frac{-(\xi_3 c_1 + \gamma\sigma_3 g_1)}{2(\xi_3 c_2 + \gamma\sigma_3 g_2)} \qquad \text{(xviic)}$$

$$\mathrm{G} = \frac{-(\xi_4 d_1 + \gamma\sigma_4 h_1)}{2(\xi_4 d_2 + \gamma\sigma_4 h_2)} \qquad \text{(xviid)}$$

For exponential functions taking the constraint optimization of CO against CO_2 be given by the Langrange's function;

$$Z' = \psi(\lambda, \theta, G) + \gamma\Phi(\lambda, \theta, G)$$

$$= \tau_0 + \tau_1(p_0 + p_1 \exp(-\lambda)) + \tau_2(q_0 + q_1\theta + q_2\theta^2) + \tau_3(r_0 + r_1G + r_2G^2)$$

$$+\gamma[\rho_0 + \rho_1(k_0 + k_1\lambda + k_2\lambda^2) + \rho_2(l_0 + l_1\theta + l_2\theta^2) + \rho_3(m_0 + m_1G + m_2G^2)]$$

When the equation was differentiated with respect to lambda, it reduced to;

$$\frac{\partial Z'}{\partial \lambda} = -\tau_1 p_1 \exp(-\lambda) + \gamma\rho_1 k_1 + 2\gamma\rho_1 k_2 = 0 \qquad \text{(xviii)}$$

However, to linearize the equation, lambda as taken as $\lambda = 1+h$, where h is small, then;

$$f(\lambda) = f(1) + f'(1)h + \cdots = \exp(-\lambda)$$

$$= \exp(-1) - \exp(-1)(\lambda - 1)$$

$$= \exp(-1) + \exp(-1)(1 - \lambda)$$

$$\therefore \exp(-\lambda) = \exp(-1)(2 - \lambda)$$

Substituting this in equation (4.31) and solving for λ, then;

$$-2\tau_1 p_1 + \tau_1 p_1\lambda + \exp(1)\gamma\rho_1 k_1 + 2\exp(1)\gamma\rho_1 k_2\lambda = 0$$

$$\lambda = \frac{2\tau_1 p_1 - \exp(1)\gamma\rho_1 k_1}{\tau_1 p_1 + 2\exp(1)\gamma\rho_1 k_2\lambda} \qquad \text{(xixa)}$$

Also $\frac{\partial Z'}{\partial \theta} = \tau_2 q_1 + 2\tau_2 q_2\theta\gamma + \rho_2 l_1 + 2\theta\gamma\rho_2 l_2 = 0$

$$\theta = \frac{-(\tau_2 q_1 + \gamma\rho_2 l_1)}{2(\tau_2 q_2 + \gamma\rho_2 l_2)} \qquad \text{(xixb)}$$

Finally $\frac{\partial Z'}{\partial G} = \tau_3 r_1 + 2\tau_3 r_2 G + \rho_2 l_1 + \rho_3\gamma m_1 + 2\rho_3\gamma m_2 G = 0$

$$G = \frac{-(\tau_3 r_1 + \gamma\rho_3 m_1)}{2(\tau_3 r_2 + \gamma\rho_3 m_2)} \qquad \text{(xixc)}$$

APPENDIX III

Source Code for Engine Performance and Emission Models

```
Public Class Vehicle
    Public Sub computemodels(ByVal Dat As Date, ByVal VehicleNumber As String, ByVal
Parameter As String, ByVal Predictedvalue As Double, ByVal Observerdvalue As Double,
ByVal Variance As Double, ByVal AbsoluteError As Double, ByVal Remark As String)
        Dim con As New Odbc.OdbcConnection("DSN=EPEModels")
        Dim str As String = "insert into computations(Date,VehicleNumber,
Parameter,Predictedvalue, Observerdvalue,Variance,AbsoluteError,Remark)  values('" & Dat
& "','" & VehicleNumber & "','" & Parameter & "','" & Predictedvalue & "','" &
Observerdvalue & "', '" & Variance & "','" & AbsoluteError & "','" & Remark & "')"
        Dim data As New Odbc.OdbcDataAdapter(str, con)
        Dim ds As New DataSet
        data.Fill(ds, "Computations")
    End Sub
End Class
Public Class Form1

    Private Sub Form1_Load(ByVal sender As System.Object, ByVal e As System.EventArgs)
Handles MyBase.Load
        'Initialize the start up parameters to avoid runtime errors
        Me.ToolStripStatusLabel2.Text = MDIParent1.ToolStripStatusLabel4.Text
        Me.TextBox1.Text = "Vehicle Number"
        Me.TextBox2.Text = 0
        Me.TextBox3.Text = 0
        Me.TextBox4.Text = 0
        Me.TextBox5.Text = 0
        Me.TextBox6.Text = FormatNumber(0, 2)
        Me.TextBox7.Text = FormatNumber(0, 2)
        Me.TextBox8.Text = FormatNumber(0, 2)
        Me.TextBox9.Text = FormatNumber(0, 2)
        Me.TextBox10.Text = FormatNumber(0, 2)
        Me.TextBox11.Text = FormatNumber(0, 2)
        Me.ToolStripStatusLabel4.Text = 0
    End Sub

    Private Sub Button4_Click(ByVal sender As System.Object, ByVal e As
System.EventArgs) Handles Button4.Click
        validatenulls() '*******A Public Function called from here
    End Sub
    Public Sub validatenulls()
        'Chech to see that all the required model computation parameters are available i.e not
null/empty.
        '*****Zero valued parameters are allowed if the parameter is not a member of the model
function**********
        If Me.TextBox1.Text = "" Then
            Beep()
            MessageBox.Show("Value cannot be null for vehicle number", "Missing Vehicle
Number", MessageBoxButtons.OK, MessageBoxIcon.Error)
```

```
            Me.TextBox1.Focus()
        Else
            If Me.TextBox2.Text = "" Then
                Me.TextBox2.Text = FormatNumber(0, 1)
            Else
                If Me.TextBox3.Text = "" Then
                    Me.TextBox3.Text = FormatNumber(0, 1)
                Else
                    If Me.TextBox4.Text = "" Then
                        Me.TextBox4.Text = FormatNumber(0, 1)
                    Else
                        If Me.TextBox5.Text = "" Then
                            Me.TextBox5.Text = FormatNumber(0, 1)
                        Else
                            If Me.TextBox6.Text = "" Then
                                Me.TextBox6.Text = FormatNumber(0, 2)
                            Else
                                If Me.TextBox7.Text = "" Then
                                    Me.TextBox7.Text = FormatNumber(0, 2)
                                Else
                                    If Me.TextBox8.Text = "" Then
                                        Me.TextBox8.Text = FormatNumber(0, 2)
                                    Else
                                        If Me.TextBox9.Text = "" Then
                                            Me.TextBox9.Text = FormatNumber(0, 2)
                                        Else
                                            If Me.TextBox10.Text = "" Then
                                                Me.TextBox10.Text = FormatNumber(0, 2)
                                            Else
                                                If Me.TextBox11.Text = "" Then
                                                    Me.TextBox11.Text = FormatNumber(0, 2)
                                                Else
                                                    computemodels()
                                                End If
                                            End If
                                        End If
                                    End If
                                End If
                            End If
                        End If
                    End If
                End If
            End If

        End If
    End Sub
    Public Sub computemodels()
        'compute the SFC Model here which is a function of S,θ,λ and G
        Dim SFCModel As Double = FormatNumber(CDbl(1.49873 * (574.39 - 0.117882 *
Trim(Me.TextBox2.Text) + 0.000021843 * (Trim(Me.TextBox2.Text) *
```

```
Trim(Me.TextBox2.Text))) - 0.364198 * (468.7 - 5.97214 * Trim(Me.TextBox3.Text) +
0.185 * (Trim(Me.TextBox3.Text) * Trim(Me.TextBox3.Text))) + 0.977251 * (2318.99 -
3458.63 * Trim(Me.TextBox4.Text) + 1529.35 * (Trim(Me.TextBox4.Text) *
Trim(Me.TextBox4.Text))) + 0.711741 * (553.2 - 354.5 * (Trim(Me.TextBox5.Text)) - 225 *
(Trim(Me.TextBox5.Text) * Trim(Me.TextBox5.Text))) - 763.419), 2)
    'Compute the Maximum Torque Model here which is a function of S,θ,λ and G
    Dim MaximumTorqueModel As Double = FormatNumber(CDbl(1.18874 * (76.5931 +
0.0868755 * Trim(Me.TextBox2.Text) - 0.0179363 * (Trim(Me.TextBox2.Text) *
Trim(Me.TextBox2.Text))) + 0.0957315 * (153.9 + 3.43071 * Trim(Me.TextBox3.Text) -
0.137857 * (Trim(Me.TextBox3.Text) * Trim(Me.TextBox3.Text))) + 1.42897 * (43.4225 +
300.712 * Trim(Me.TextBox4.Text) - 160.486 * (Trim(Me.TextBox4.Text) *
Trim(Me.TextBox4.Text))) + 1.31692 * (68.2571 + 277.786 * Trim(Me.TextBox5.Text) -
175 * (Trim(Me.TextBox5.Text) * Trim(Me.TextBox5.Text))) - 511.983), 2)
    'Compute the brake power mode which is a function of  S,θ,λ and G. Here ^ means
"raise to the poer of" ****Used for simplicity of code***** ---*** Model Adopted Onwards-
-- *******
    Dim BrakePowerModel As Double = FormatNumber(CDbl(1.0876 * (-31.2362 +
0.0486668 * Trim(Me.TextBox2.Text) - 6.61124 / 10 ^ 6 * Trim(Me.TextBox2.Text) ^ 2) -
0.0623495 * (4.7 + 5.00643 * Trim(Me.TextBox3.Text) - 0.11714 *
Trim(Me.TextBox3.Text) ^ 2) + 0.90759 * (-19.8802 + 127711 * Trim(Me.TextBox4.Text) +
66.7119 * Trim(Me.TextBox4.Text) ^ 2) + 1.69442 * (15.8 + 53.9643 *
Trim(Me.TextBox5.Text) - 33.9286 * Trim(Me.TextBox5.Text) ^ 2) - 102.231), 2)
    'Compute the carbon mono  model which is a function of θ,λ and G. The exponential  E
has a value of  math 2.7182818284590451
    Dim CarbonMonoxideModel As Double = FormatNumber(CDbl(1.07646 * (6.93145 -
0.182943 * Trim(Me.TextBox3.Text) + 0.00350551 * Trim(Me.TextBox3.Text) ^ 2) +
0.933631 * (-0.741057 + 16.7074 / Math.E ^ Trim(Me.TextBox4.Text)) + 0.08576 *
(5.59048 - 1.82143 * Trim(Me.TextBox5.Text) + 2.02381 * Trim(Me.TextBox5.Text) ^ 2) -
10.275), 2)
    'Compute the carbon dioxide model which is a function of θ,λ and G
    Dim CarbonDioxideModel As Double = FormatNumber(CDbl(1.0744 * (8.35879 +
0.182585 * Trim(Me.TextBox3.Text) - 0.00422943 * Trim(Me.TextBox3.Text) ^ 2) +
0.93717 * (-13.5521 + 39.2993 * Trim(Me.TextBox4.Text) - 15.93 * (Me.TextBox4.Text) ^
2) + 0.826081 * (0.533333 + 21.6429 * Trim(Me.TextBox5.Text) - 12.619 *
Trim(Me.TextBox5.Text) ^ 2) - 17.7725), 2)
    'Compute the hydrocarbon model which is a function of θ,λ and G
    Dim HydroCarbonModel As Double = FormatNumber(CDbl(0.789274 * (516.49 -
3.9411 * Trim(Me.TextBox3.Text) + 0.109729 * Trim(Me.TextBox3.Text) ^ 2) + 1.004435 *
(2396.46 - 3310.26 * Trim(Me.TextBox4.Text) + 1384.9 * Trim(Me.TextBox4.Text) ^ 2) -
0.130947 * (642.048 - 377.5 * Trim(Me.TextBox5.Text) + 213.095 *
Trim(Me.TextBox5.Text) ^ 2) - 333.197), 2)
    Dim li As New ListViewItem
    li = Me.ListView2.Items.Add(Me.DateTimePicker1.Value)
    li.SubItems.Add(Trim(Me.TextBox1.Text))
    li.SubItems.Add("SFC")
    li.SubItems.Add(SFCModel)
    li.SubItems.Add(FormatNumber(Trim(Me.TextBox6.Text), 2))
    li.SubItems.Add(SFCModel - FormatNumber(Trim(Me.TextBox6.Text), 2))
    li.SubItems.Add(FormatNumber((SFCModel -
FormatNumber(Trim(Me.TextBox6.Text), 2)) / SFCModel, 2))
```

```
        li.SubItems.Add(Trim(Me.RichTextBox1.Text))
        Dim li2 As New ListViewItem
        li2 = Me.ListView2.Items.Add(Me.DateTimePicker1.Value)
        li2.SubItems.Add(Trim(Me.TextBox1.Text))
        li2.SubItems.Add("Tmax")
        li2.SubItems.Add(MaximumTorqueModel)
        li2.SubItems.Add(FormatNumber(Trim(Me.TextBox7.Text), 2))
        li2.SubItems.Add(MaximumTorqueModel - FormatNumber(Trim(Me.TextBox7.Text),
2))
        li2.SubItems.Add(FormatNumber((MaximumTorqueModel -
FormatNumber(Trim(Me.TextBox7.Text), 2)) / MaximumTorqueModel, 2))
        li2.SubItems.Add(Trim(Me.RichTextBox1.Text))
        Dim li3 As New ListViewItem
        li3 = Me.ListView2.Items.Add(Me.DateTimePicker1.Value)
        li3.SubItems.Add(Trim(Me.TextBox1.Text))
        li3.SubItems.Add("bP")
        li3.SubItems.Add(BrakePowerModel)
        li3.SubItems.Add(FormatNumber(Trim(Me.TextBox8.Text), 2))
        li3.SubItems.Add(BrakePowerModel - FormatNumber(Trim(Me.TextBox8.Text), 2))
        li3.SubItems.Add(FormatNumber((BrakePowerModel -
FormatNumber(Trim(Me.TextBox8.Text), 2)) / BrakePowerModel, 2))
        li3.SubItems.Add(Trim(Me.RichTextBox1.Text))
        Dim li4 As New ListViewItem
        li4 = Me.ListView2.Items.Add(Me.DateTimePicker1.Value)
        li4.SubItems.Add(Trim(Me.TextBox1.Text))
        li4.SubItems.Add("CO")
        li4.SubItems.Add(CarbonMonoxideModel)
        li4.SubItems.Add(FormatNumber(Trim(Me.TextBox9.Text), 2))
        li4.SubItems.Add(CarbonMonoxideModel - FormatNumber(Trim(Me.TextBox9.Text),
2))
        li4.SubItems.Add(FormatNumber((CarbonMonoxideModel -
FormatNumber(Trim(Me.TextBox9.Text), 2)) / CarbonMonoxideModel, 2))
        li4.SubItems.Add(Trim(Me.RichTextBox1.Text))
        Dim li5 As New ListViewItem
        li5 = Me.ListView2.Items.Add(Me.DateTimePicker1.Value)
        li5.SubItems.Add(Trim(Me.TextBox1.Text))
        li5.SubItems.Add("CO2")
        li5.SubItems.Add(CarbonDioxideModel)
        li5.SubItems.Add(FormatNumber(Trim(Me.TextBox10.Text), 2))
        li5.SubItems.Add(CarbonDioxideModel - FormatNumber(Trim(Me.TextBox10.Text),
2))
        li5.SubItems.Add(FormatNumber((CarbonDioxideModel -
FormatNumber(Trim(Me.TextBox10.Text), 2)) / CarbonDioxideModel, 2))
        li5.SubItems.Add(Trim(Me.RichTextBox1.Text))
        Dim li6 As New ListViewItem
        li6 = Me.ListView2.Items.Add(Me.DateTimePicker1.Value)
        li6.SubItems.Add(Trim(Me.TextBox1.Text))
        li6.SubItems.Add("HC")
        li6.SubItems.Add(HydroCarbonModel)
        li6.SubItems.Add(FormatNumber(Trim(Me.TextBox11.Text), 2))
```

```
        li6.SubItems.Add(HydroCarbonModel - FormatNumber(Trim(Me.TextBox11.Text), 2))
        li6.SubItems.Add(FormatNumber((HydroCarbonModel -
FormatNumber(Trim(Me.TextBox11.Text), 2)) / HydroCarbonModel, 2))
        li6.SubItems.Add(Trim(Me.RichTextBox1.Text))
        Me.ToolStripStatusLabel4.Text = Me.ListView2.Items.Count
        Dim _vehicle As New Vehicle
        _vehicle.computemodels(Me.DateTimePicker1.Value, Me.TextBox1.Text, "SFC",
SFCModel, Me.TextBox6.Text, (SFCModel - FormatNumber(Trim(Me.TextBox6.Text), 2)),
(FormatNumber((SFCModel - FormatNumber(Trim(Me.TextBox6.Text), 2)) / SFCModel,
2)), Me.RichTextBox1.Text)
        _vehicle.computemodels(Me.DateTimePicker1.Value, Me.TextBox1.Text, "Tmax",
MaximumTorqueModel, Me.TextBox7.Text, (MaximumTorqueModel -
FormatNumber(Trim(Me.TextBox7.Text), 2)), (FormatNumber((MaximumTorqueModel -
FormatNumber(Trim(Me.TextBox7.Text), 2)) / MaximumTorqueModel, 2)),
Me.RichTextBox1.Text)
        _vehicle.computemodels(Me.DateTimePicker1.Value, Me.TextBox1.Text, "bP",
BrakePowerModel, Me.TextBox8.Text, (BrakePowerModel -
FormatNumber(Trim(Me.TextBox8.Text), 2)), (FormatNumber((BrakePowerModel -
FormatNumber(Trim(Me.TextBox8.Text), 2)) / BrakePowerModel, 2)),
Me.RichTextBox1.Text)
        _vehicle.computemodels(Me.DateTimePicker1.Value, Me.TextBox1.Text, "CO",
CarbonMonoxideModel, Me.TextBox9.Text, (CarbonMonoxideModel -
FormatNumber(Trim(Me.TextBox9.Text), 2)), (FormatNumber((CarbonMonoxideModel -
FormatNumber(Trim(Me.TextBox9.Text), 2)) / CarbonMonoxideModel, 2)),
Me.RichTextBox1.Text)
        _vehicle.computemodels(Me.DateTimePicker1.Value, Me.TextBox1.Text, "CO2",
CarbonDioxideModel, Me.TextBox10.Text, (CarbonDioxideModel -
FormatNumber(Trim(Me.TextBox10.Text), 2)), (FormatNumber((CarbonDioxideModel -
FormatNumber(Trim(Me.TextBox10.Text), 2)) / CarbonDioxideModel, 2)),
Me.RichTextBox1.Text)
        _vehicle.computemodels(Me.DateTimePicker1.Value, Me.TextBox1.Text, "HC",
HydroCarbonModel, Me.TextBox11.Text, (HydroCarbonModel -
FormatNumber(Trim(Me.TextBox11.Text), 2)), (FormatNumber((HydroCarbonModel -
FormatNumber(Trim(Me.TextBox11.Text), 2)) / HydroCarbonModel, 2)),
Me.RichTextBox1.Text)
        MessageBox.Show("Success")
    End Sub

    Private Sub ListView2_SelectedIndexChanged(ByVal sender As System.Object, ByVal e
As System.EventArgs) Handles ListView2.SelectedIndexChanged
        Me.ToolStripStatusLabel4.Text = Me.ListView2.Items.Count
    End Sub

    Private Sub Button1_Click(ByVal sender As System.Object, ByVal e As
System.EventArgs)
    End Sub

    Private Sub Button2_Click(ByVal sender As System.Object, ByVal e As
System.EventArgs) Handles Button2.Click
        Me.ListView2.Items.Clear()
```

```
    End Sub

    Private Sub Button3_Click(ByVal sender As System.Object, ByVal e As
System.EventArgs) Handles Button3.Click
        Me.Close()
    End Sub
End Class
```