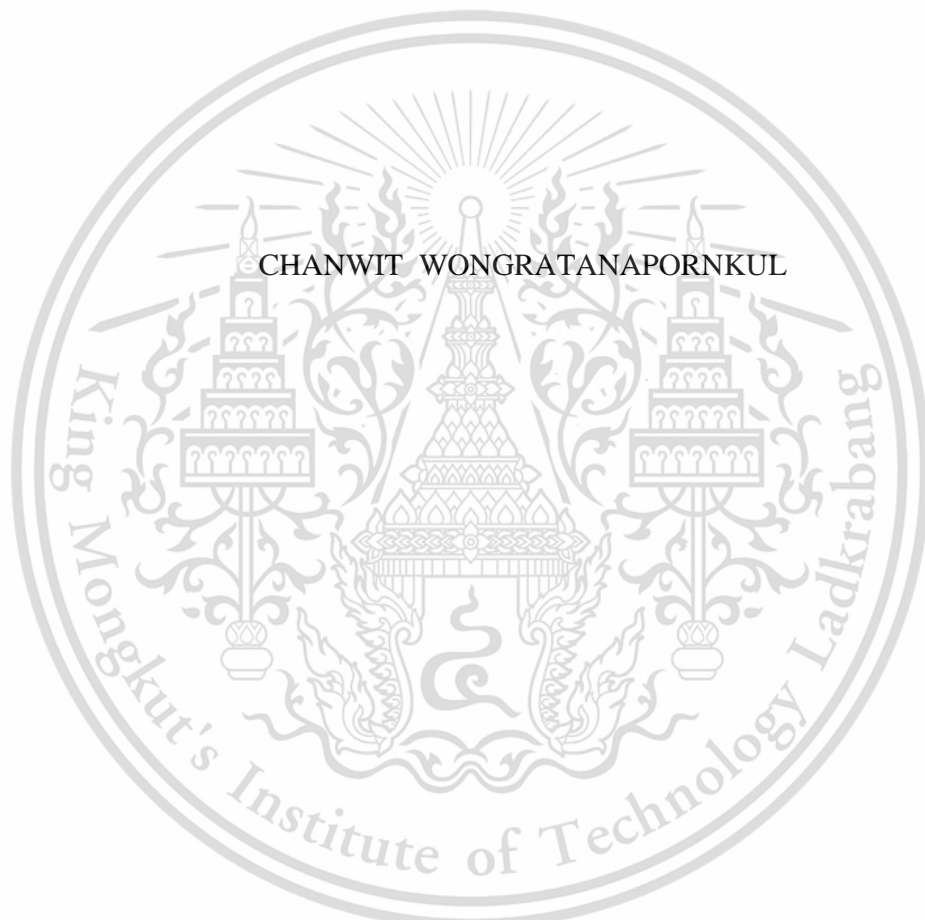


EXPERIMENTAL STUDY OF EXHAUST GAS COMPOSITION  
FROM COMBUSTION IN GASOLINE ENGINE  
WITH POLLUTED AIR INTAKE



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A THESIS REPORT SUBMITTED IN PARTIAL FULFILLMENT  
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THESIS TITLE      EXPERIMENTAL STUDY OF EXHAUST GAS  
COMPOSITION FROM COMBUSTION IN GASOLINE  
ENGINE WITH POLLUTED AIR INTAKE

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### ABSTRACT

This research was an experimental study of exhaust gas composition from combustion in a gasoline engine with polluted air intake. The experimentations were tested at the engine's idle speed with removal of oxygen sensor. Pollution gases composing of carbon monoxide (CO) in range of 0 to 1.40 %vol., carbon dioxide (CO<sub>2</sub>) in range of 0 to 5.0 %vol., and unburned hydrocarbon (HC) in range of 0 to 400 ppm of which proportions were varying was diluted to intake air upstream to the tested engine. Compositions and temperature of the exhaust gas from the exhaust pipe were sampled and analyzed through Exhaust Gas Analyzer to determine the concentrations of CO, CO<sub>2</sub>, HC, and excess O<sub>2</sub>. Experimentation results revealed that the highest values of the exhaust HC and CO were about 18.9 times and 4.6 times of the normal fresh air intake while the highest decrease of the exhaust CO<sub>2</sub> was about 38.3%. The exhaust gas temperature decreased was about 15.73%.

The established mathematical model was able to be used only as a predictor to the trend of the exhaust emission. The model could provide consistent results only for the exhaust HC and CO, but the estimated HC was substantial higher than the measured values due to limitation of HC detector. The estimated CO<sub>2</sub> were lower and the estimated O<sub>2</sub> were higher than the actual values. Therefore, the mathematical model for estimation of the exhaust gas constituent should be improved by considering of the chemical effect, not only considering of the principle of mass conservation.

According to the experimental results, it was found that dilution of the CO<sub>2</sub> to the air intake has evident effect to the exhaust gas constituent. Further testing under the normal operating speed range 1,200-3,000 rpm at fixed throttle position of 30% opening shown that the fuel consumption increased about 23.05%, the maximum engine torque decreased about 25.92%, the maximum brake power decreased about 26.48%, and the brake specific fuel consumption decreased about 9.74% averagely.

## ACKNOWLEDGEMENT

This research is successful with many supports. Firstly, I would like to express my deep gratitude to my thesis advisor, Asst. Prof. Dr.Pheechar Karin of International College at King Mongkut's Institute of Technology Ladkrabang, for the encouragement and advice to get through all obstacles and challenges from the beginning to the end of my study. Secondly, I would like to express my sincere gratitude to all lecturers who give me strong supports and valuable guidance throughout the long journeys. Thirdly, I would like to extend many thanks to my friends for their supports and motivation always. Finally, I would like to give special thanks and love to my son and daughter, Pooh and Noo-dee, who inspire me greatly to overcome all troubles. Because of their love, I always find my strength to fulfil my goal. More importantly, I would like to express the greatest gratitude to my parents who never fail me any supports and motivation. It is certain that my achievement would not be possible without them.

Chanwit Wongratanapornkul



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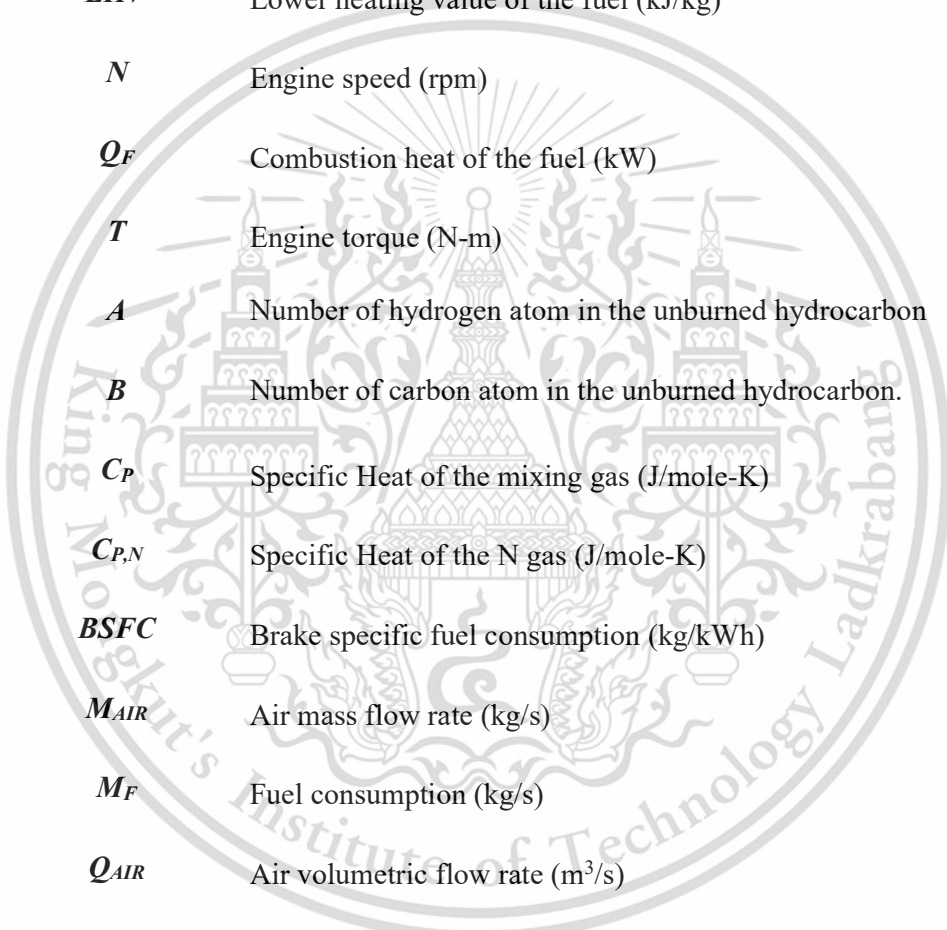
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## LIST OF SYMBOLS



$A/F$	Air-fuel Ratio
$BP$	Engine brake power (kW)
$F$	Net force from spring balance or load cell (N)
$LHV$	Lower heating value of the fuel (kJ/kg)
$N$	Engine speed (rpm)
$Q_F$	Combustion heat of the fuel (kW)
$T$	Engine torque (N-m)
$A$	Number of hydrogen atom in the unburned hydrocarbon
$B$	Number of carbon atom in the unburned hydrocarbon.
$C_P$	Specific Heat of the mixing gas (J/mole-K)
$C_{P,N}$	Specific Heat of the N gas (J/mole-K)
$BSFC$	Brake specific fuel consumption (kg/kWh)
$M_{AIR}$	Air mass flow rate (kg/s)
$M_F$	Fuel consumption (kg/s)
$Q_{AIR}$	Air volumetric flow rate (m <sup>3</sup> /s)
$Q_F$	Volume of fuel used in a period of measurement (L/s)
$Q_{TH}$	Theoretical displacement volume (m <sup>3</sup> /s)
$R$	Load arm length (m)
$Y_I$	Volume fraction of gas
$\lambda_M$	Molecular air-fuel ratio

$\rho_M$  Mean density of the mixing gas ( $\text{kg/m}^3$ )

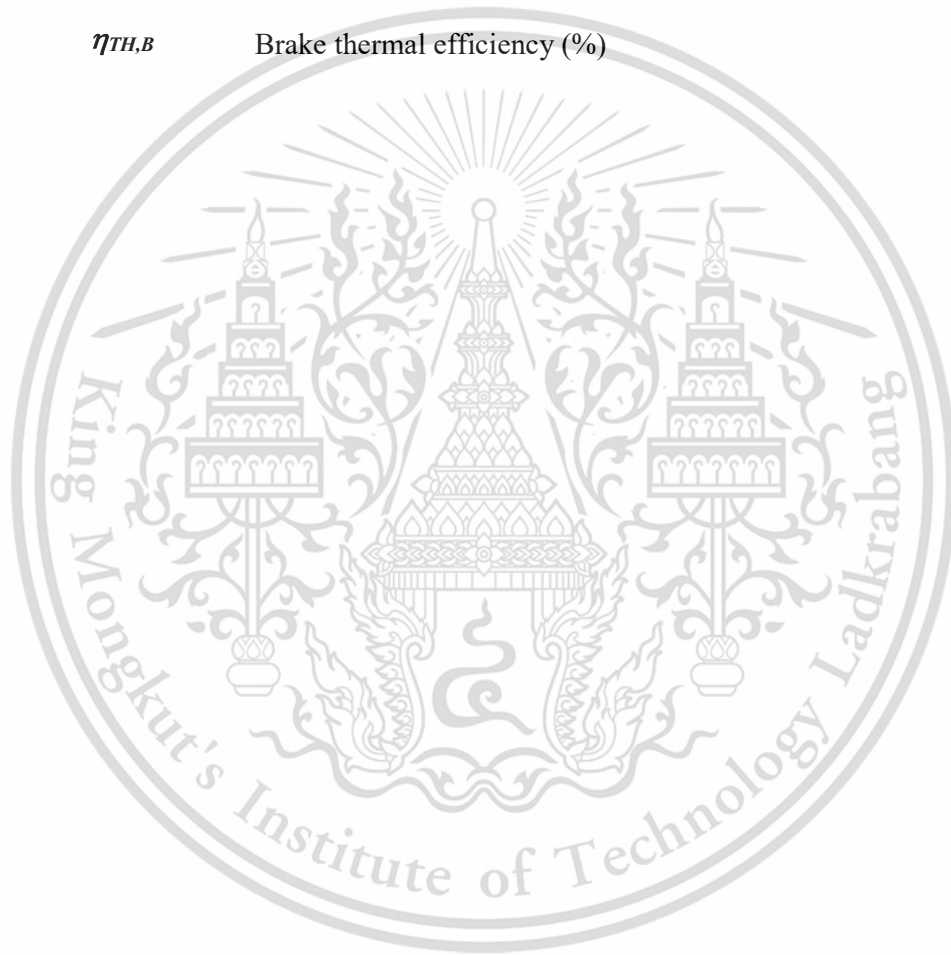
$\rho_N$  Density of the N gas element ( $\text{kg/m}^3$ )

$\rho_{AIR}$  Air density ( $\text{kg/m}^3$ )

$\rho_F$  Fuel density ( $\text{kg/m}^3$ )

$\eta_V$  Volumetric efficiency (%)

$\eta_{TH,B}$  Brake thermal efficiency (%)



## LIST OF DEFINITIONS

BSFC	Brake Specific Fuel Consumption
CO	Carbon Monoxide
CO <sub>2</sub>	Carbon Dioxide
EGR	Exhaust Gases Recirculation
EVAP	Evaporative Emission Control System
FID	Flame Ionization Detection
HC	Unburned Hydrocarbon
H <sub>2</sub>	Hydrogen
KWH	Kilo-watt-hour
LML	Lean-misfire limit
NDIR	Non-Dispersive Infrared
NMHC	Non-Methane Hydrocarbons
NO <sub>x</sub>	Nitrogen Oxides
O <sub>2</sub>	Oxygen
PPM	Parts per million
PM	Particulate Matters
SI	Spark-ignition
THC	Total Hydrocarbon
%VOL	Percentage by Volume

# CHAPTER 1

## INTRODUCTION

### 1.1 Research Background

Exhaust gas emission from the automobiles powered by internal combustion engines, both the gasoline and the diesel engines, operating on conventional petroleum fuels, natural gas and other liquid fuels were identified as the major source to urban air pollution of all large cities where road traffic is heavy. Carbon monoxide (CO), nitrogen oxides (NO<sub>x</sub>), unburned hydrocarbons (HC), and particulate matter (PM) are commonly referred as vehicle emissions while carbon dioxide (CO<sub>2</sub>) being a greenhouse gas contribute to the global warming effect. Exhaust gas constituents from the automobiles can be divided into three classifications: 1) Harmful to health, 2) Objectionable, and 3) Potentially objectionable, while certain constituents may be identified in more than one of the mentioned classifications. Because the vehicular air pollution seriously affects to environment and people health, many countries have tried to curb such problem with the emission control legislations.

All large major cities have been suffering from polluted air; those pollutants of which are mainly exhaust products from both gasoline and diesel engines burning fossil fuels. The automobiles exhaust products are carbon dioxide (CO<sub>2</sub>), carbon monoxide (CO), nitrogen oxides (NO<sub>x</sub>), unburned hydrocarbons (HC), and particulate matters (PM). Carbon dioxide is known well as a greenhouse gas which is responsible for the global warming affecting environmental and human health problems. Global warming effect becomes serious issue for many countries around the world to resolve. Such that are global climate change pacts, strict emission control and legislations. To curb such dangerous climate change, it requires heavy reduction of vehicle emissions and the use of alternatives to fossil fuels worldwide.

Idling emission is considered more harmful because the pollutants cannot be dispersed by the wake created by moving vehicle, while an automobile may have over 25% of its time spent in the idle condition, park-and-wait mode (Ning, 2005). In Bangkok, most drivers spend more than 44 days a year idling in standstill traffic (Lumley, 1999). Although Thai Government has trying to solve traffic congestion problem, the problem seems more complex and difficult. As reported in the CNN

Money website, Bangkok has the worst evening rush hour traffic in the world for the second consecutive year as of 2016, according to GPS manufacturer TomTom. Based on the traffic congestion statistics, Bangkok people spend longer travelling time in morning and evening rush hour traffic about 91% and 118% respectively, comparing to another travelling period. This means that Bangkok people waste average of 64 minutes a day or 244 days per year in traffic congestion during Monday to Friday (<https://www.tomtom.com>).

It is known that the composition of the exhaust gases depends upon many factors such as the relative proportions of fuel and air fed to the engine, fuel composition, and the completeness of combustion process. In addition, local chemistry of the burning process and the chamber geometry also has a profound influence on the final composition of exhaust gases. The emissions from a motor vehicle also vary under different driving conditions and the idle condition is such the specific driving conditions. It is believed that, due to poor combustion conditions inside the engine when the vehicle is operating at idle and the poor dispersion condition, pollutant concentration will be high around the vehicles (Ning, 2005).

Although there have been many experimental investigations on the effects of polluted intake with the engine emission were studied; mostly devoted on varying of the rate of EGR (Exhaust Gases Recirculation), but not much on the effect of the proportion of the pollutant concentration. Though EGR could reduce a large amount of  $\text{NO}_x$  emission, it leads to increase CO and HC concentrations in emission. Besides, the increase of EGR rate results in slow combustion process which causes partial burning or misfire and, subsequently, produces higher CO and HC concentrations.

The aim of this research is to investigate the effect of polluted air intake on the exhaust gases component by varying the polluted air intake composition and mixing exhaust gases to dilute the upstream air intake of a gasoline engine which was running at idle condition in order to simulate the situations of vehicles stopping and waiting along street or in terminal with the engine switched on. Experimental results are measured, analyzed and compared to the exhaust gases of normal fresh air intake. Later, the results are also used to establish a test at partial-load conditions for comparing its performance.

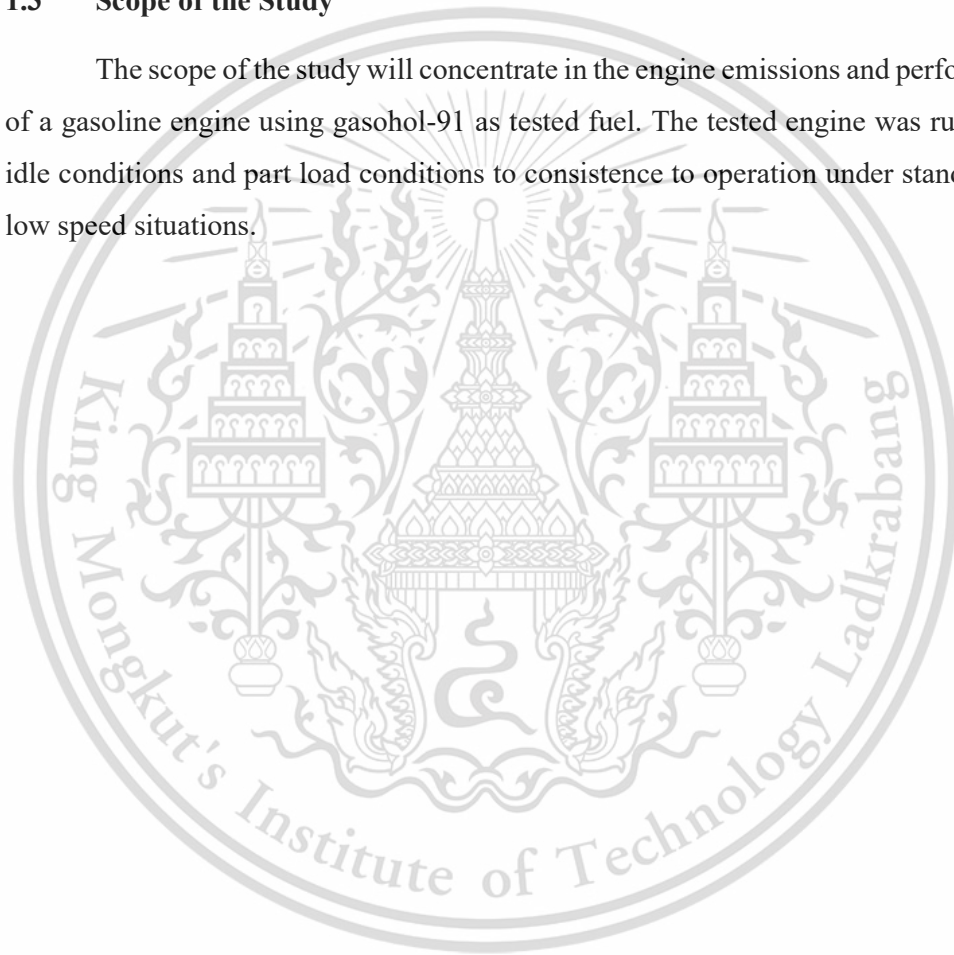
## 1.2 Objective of the Study

The main objectives of this study are:

1. To investigate the effect of pollution gas composition on engine emissions through experimental study.
2. To investigate the effect of pollution gas on engine performance under part load operations.

## 1.3 Scope of the Study

The scope of the study will concentrate in the engine emissions and performance of a gasoline engine using gasohol-91 as tested fuel. The tested engine was run under idle conditions and part load conditions to consistence to operation under standby and low speed situations.



## CHAPTER 2

### LITERATURE REVIEW

#### 2.1 Introduction

The exhaust gases from the internal combustion engines are complex mixtures consisting of the products of complete combustion, small amounts of the oxidation products of sulfur and nitrogen, and compounds derived from the fuel and lubricant (Martin, 2014). Composition of the exhaust gas depends on the relative proportions of fuel and air fed to the engine, fuel composition, and completeness of combustion process (Heywood, 1988). However, local chemistry of the burning process and the chamber geometry will also have a profound influence on the final composition of any exhaust gas (Blair, 1999).

Air-fuel ratio is one of the most important engine variable that affects the exhaust emission from the gasoline engines (Pundir, 2013). Normally, the engine should be operated on lean mixtures that would give low CO and HC emission, and moderate  $\text{NO}_x$  emissions. But presently most gasoline engines are operated very close to stoichiometric conditions for catalytic control of  $\text{NO}_x$  emission. Figure 2.1 demonstrates variation of the exhaust gas composition from typical gasoline engine corresponding to the air-fuel ratio.

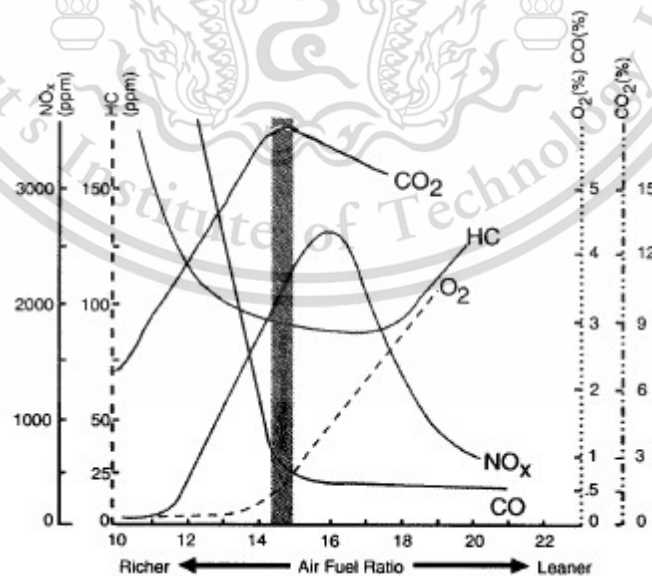


Figure 2.1 Relation between Exhaust Gas and Air-Fuel Ratio

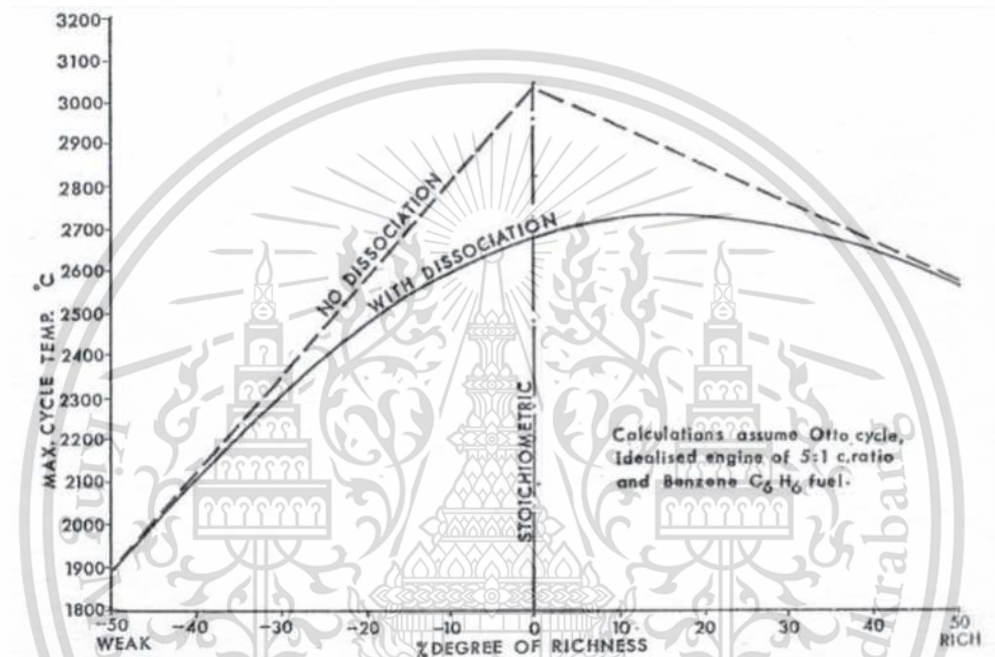
If the air-fuel mixture is too rich, there is insufficient oxygen for complete combustion and some of the fuel will not be burnt or at least only partly burnt. Since hydrogen has a greater affinity for oxygen, the hydrogen will take all the oxygen it needs leaving the carbon with a deficiency of oxygen. Because of the shortage of oxygen, a percentage of the carbon will be converted to carbon monoxide (CO) as well as carbon dioxide (CO<sub>2</sub>), and also, with very rich mixtures, particles of pure unburnt carbon (HC) may be expelled from the exhaust as black smoke. Conversely, if the mixture is too lean, the atomized liquid fuel will be thoroughly mixed throughout the combustion chamber so that slow burning incomplete combustion and misfiring may occurred (Heinz, 1995).

Nitrogen and oxygen will unite to form oxides of nitrogen at rich fuel-air ratio mixture and high temperature while weak fuel-air ratio is needed to control of the HC and CO emissions. Since the fuel-air ratio cannot control the production of the pollutants simultaneously in the internal combustion engines (Tairu, 2017).

Experimental investigation of emission characteristics from a four-stroke motorecycle engine using E-10 gasohol fuel at different driving modes shown that CO and HC emissions were lower with the operation of E-10 compared to the use of unleaded gasoline, whereas the effect of ethanol on NO<sub>x</sub> emissions were not remarkable (Jia, 2005). Another study of emission characteristics of spark ignition engine operating on lower-higher molecular mass alcohol blended gasoline fuels were investigated experimentally (Gravalos, 2013). The results indicated that CO and HC concentration in the engine exhaust gases were reduced with the operation on alcohol gasoline blends, but NO<sub>x</sub> emissions with alcohol gasoline blends were higher than with normal gasoline fuel.

In actual reaction, dissociation reactions take place at elevated temperatures and pressured even under stoichiometric combustions, some free carbon monoxide and hydrogen (H<sub>2</sub>) will be created, so that there would be some carbon monoxide in the existence and minor amount of oxygen (O<sub>2</sub>) and hydrogen (H<sub>2</sub>). As the air-fuel mixture is progressively richer than stoichiometric proportion, the exhaust gas would contain greater amounts of CO and H<sub>2</sub> while would still show little traces of free oxygen O<sub>2</sub>. At the high temperature inside an engine cylinder, there are two principal dissociation reactions involved; 1)  $CO + \frac{1}{2}O_2 \rightleftharpoons CO_2$  and 2)  $H_2O + CO \rightleftharpoons H_2 + CO_2$ . That is, some of the CO<sub>2</sub> splits into CO, O<sub>2</sub>, and O; H<sub>2</sub>O into H<sub>2</sub>, H, O<sub>2</sub> and O, etc. These dissociation

reactions absorb the energy. Maximum temperature of combustion is limited by the first reaction when it is proceeding as energy is released. The second reaction is often called the “water gas reaction”. Figure 2.2 demonstrates effect of dissociation reaction causes that the maximum combustion temperature will occur when the mixture is slightly rich. Therefore, a mixture strength richer than stoichiometric is required for maximum power output (Greene, 1969).



**Figure 2.2** Effect of Dissociation Reaction to the Maximum Combustion Temperature (Greene, 1969)

In addition, incompletely burning of the fuel and non-uniform air-fuel mixture fed to each cylinder are also affect the composition of exhaust gas. Although the combustion equations are useful for determining the combustion products, they do not correspond closely to the actual constituents. For all these reasons, the composition of the exhaust gases cannot be calculated and predicted easily. The best way to determine the composition of the engine exhaust gases is to measure directly (Heywood, 1988).

Effects on combustion and emissions from displacement of  $O_2$  by  $CO_2$  had been studied (Ladommatos, 1998). The displacement of inlet charge with  $CO_2$  and  $H_2O$  can alter the combustion process in several ways. The first major effect is the reduction of  $O_2$  concentration within the engine cylinders that is available for combustion; this is referred to as *dilution effect*. The second major effect is the higher specific heat

capacities of both CO<sub>2</sub> and H<sub>2</sub>O in comparison to that of O<sub>2</sub> being displaced; this is referred to as *thermal effect*. Both CO<sub>2</sub> and H<sub>2</sub>O can, potentially, dissociate at the high temperature prevailing during combustion and the products of dissociation participate in the combustion process; this is referred to as *chemical effect*.

## 2.2 Exhaust Gas Analysis

Experimental work conducted on the motor coaches powered by two-cycle gasoline, diesel, and propane engines had shown that driving condition had a marked effect on the emission rate of all constituents. The highest CO concentration in the exhaust gas occurred under idle condition. The largest NO<sub>x</sub> concentration was at cruise and acceleration conditions. The engine conditions giving highest HC concentration were observed at deceleration and moderate concentration during idle conditions. Under all driving conditions the CO concentrations in the exhausts of the gasoline engines ranged from 1.5 to 12 %vol due to general operations with a deficiency of combustion air, while the exhaust CO from the diesel engines usually operated with a substantial excess of the air contained less than 0.1 %vol. Average oxides of nitrogen (NO<sub>x</sub>) concentrations during cruise and acceleration conditions ranged from about 200 to 2,000 ppm. and were below 60 ppm. during idle and deceleration conditions. And for all types of the engines, less than 2% of the supplied fuel is emitted as unburned HC during cruise and acceleration, whereas more than 34% of the supplied fuel was probably emitted during deceleration conditions. (Martin, 2014).

Effects of intake pollutant to the engine emission were mostly studied through varying of rate of EGR (Exhaust Gases Recirculation), while the effects of proportion of the pollutant concentration have not been studied as much. The exhaust gas mainly consists of CO<sub>2</sub>, N<sub>2</sub>, and water vapor. When a part of this exhaust gas is recirculated to the cylinder, it acts as diluent to the combusting mixture. This diluent reduces the O<sub>2</sub> concentration in the cylinders. The specific heat of the recirculated gas is much higher than the fresh air; hence EGR increases the heat capacity of the intake charge, thus decreasing the temperature rise for the same heat release in the combustion chamber (Meshack, 2015). Applications of EGR could reduce a large amount of NO<sub>x</sub> emission, but it leads to increase of CO and HC emissions (Heywood, 1988) (Jinyoung, 2001). Increase of EGR rate results to slow combustion speeds which lead to partial burning or misfire that produces higher CO and HC (Jinyoung, 2001).

### 2.3 Correlations between the Exhaust Gas Constituents of Gasoline Engines

From various researchers and experimentations, S.H. Graf, G.W. Gleeson, and W.H. Paul (Graf, 1934) reviewed correlations between the exhaust gas constituents of gasoline engines. Figure 2.3 to Figure 2.5 demonstrate some available correlation between  $\text{CO}_2\text{-O}_2$ ,  $\text{CO}_2\text{-CO}$ , and  $\text{CO-H}_2$  respectively.

In Figure 2.3, the correlation between  $\text{CO}_2\text{-O}_2$  was applicable for lean mixtures but for extremely rich mixtures it was advised to measure directly.

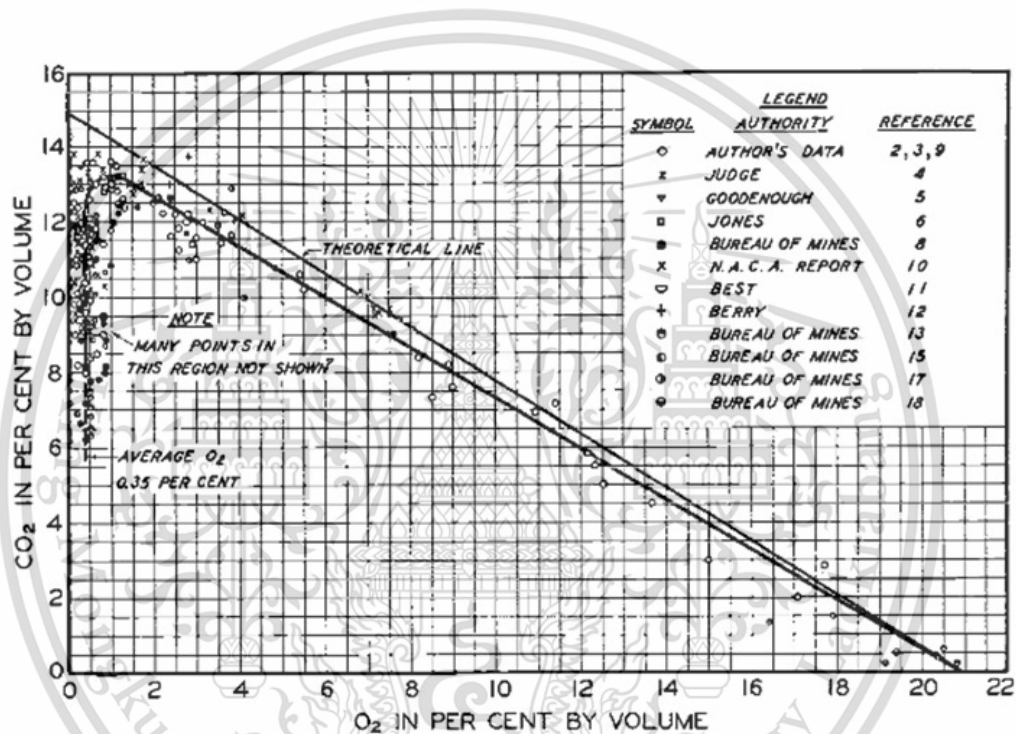


Figure 2.3 Exhaust  $\text{CO}_2$  and  $\text{O}_2$  Correlation (Graf, 1934)

There is a distinct correlation between  $\text{CO}_2\text{-CO}$  from the exhaust gases. Such  $\text{CO}_2\text{-CO}$  relationship lies in a consideration of the completeness of the combustion of carbon content of the fuel. In the absence of any hydrocarbon gases in the exhaust, such as methane ( $\text{CH}_4$ ), all the carbon must appear as the two carbon gases  $\text{CO}_2$  and  $\text{CO}$ . The correlation is recommended for practical use as a time-saving method in estimation of the exhaust gas constituents.

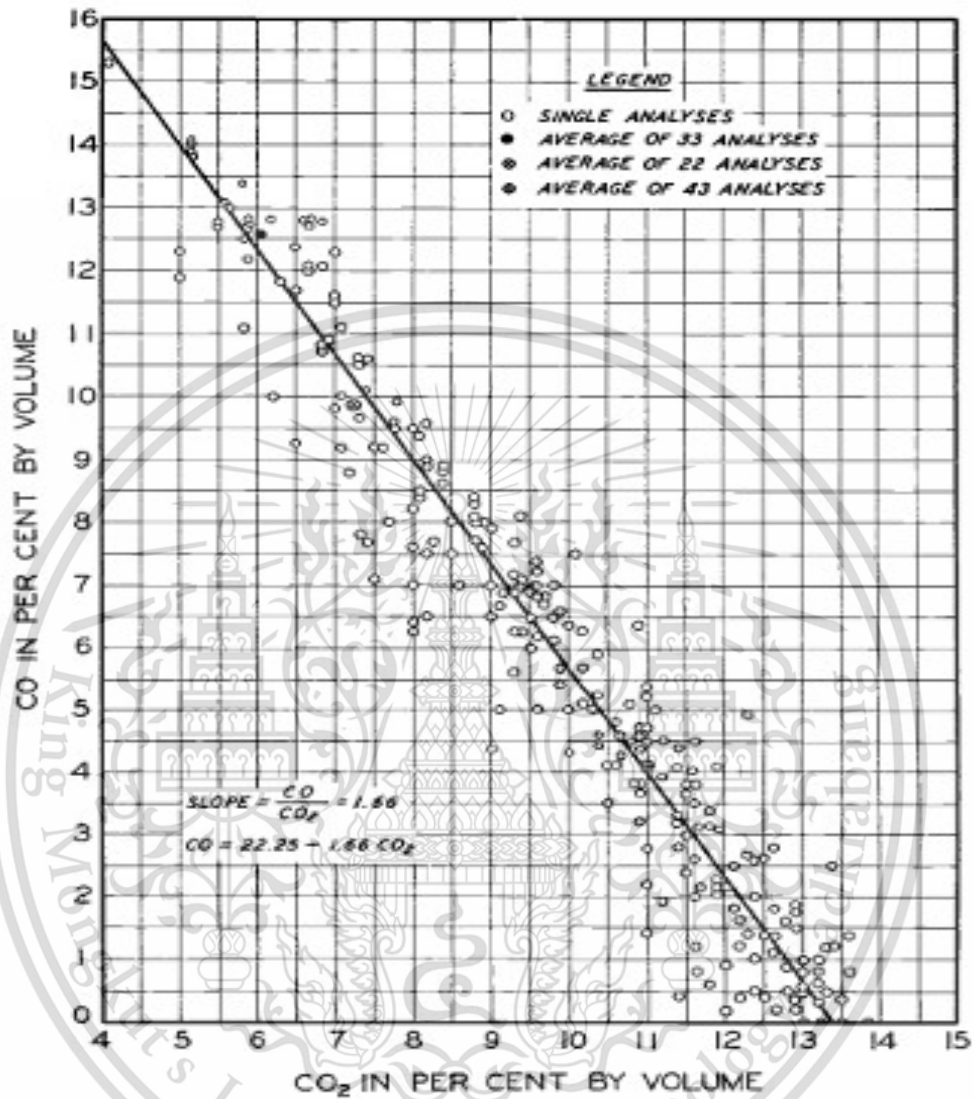


Figure 2.4 Exhaust CO<sub>2</sub> and CO Correlation (Graf, 1934)

The correlation between CO-H<sub>2</sub> described that the ratio/slope of CO to H<sub>2</sub> in percentage by volume was non-linear over the range of values which were varying from 1.96 to 2.9.

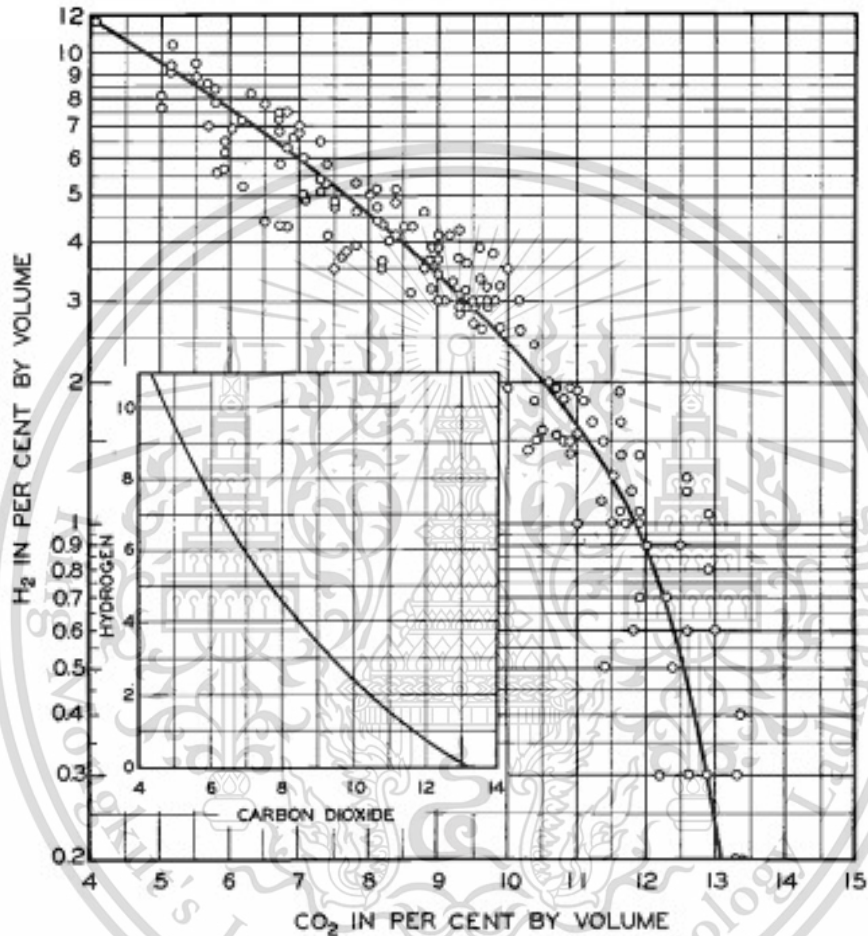
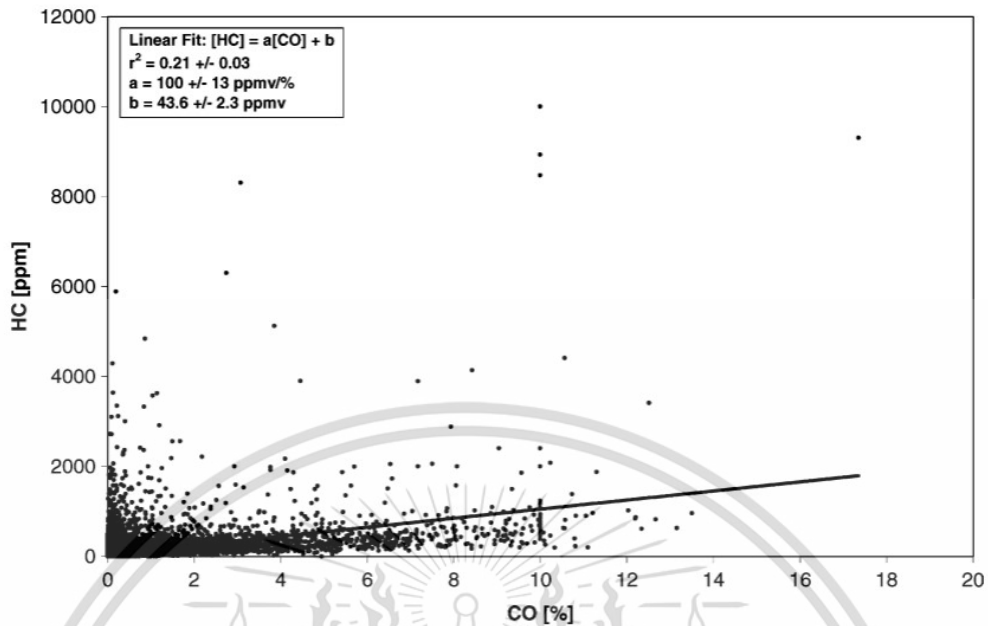


Figure 2.5 Exhaust CO<sub>2</sub> and H<sub>2</sub> Correlation (Graf, 1934)

The studies of relationship between the concentration of HC, CO, NO, and particulate matters on an individual vehicle were poorly correlated. Therefore, one or more pollutants cannot be used as predictor for other pollutants, but it should be measured separately to ensure that the variability of the emissions was properly characterized (Claudio, 2004).



**Figure 2.6** Exhaust HC and CO Correlation (Claudio, 2004)

## 2.4 Reduction of Pollutants in the Gasoline Engine

Minimizing and controlling the exhaust gas composition in the gasoline engine can be attainable by means of various measures from the following strategies. Each measure can be implemented independently, or it also can be combined to achieve the reduction of pollution.

### 2.4.1 Engine-design Measures

#### 2.4.1.1) Mixture Formation

On spark-ignition (SI) engines, air-fuel ratio is set properly by calculation the cylinder air charge and injecting the fuel quantity as correctly as possible. Electronic fuel injection systems allow adjustment of the air-fuel ratio precisely.

The composition of the exhaust emissions depends upon the ratio of the actual air intake to theoretical air, called equivalent air ratio ( $\lambda$ ), of the mixture fed to the engine cylinders. Maximum torque is produced at approximately  $\lambda=0.9$  and enriching the mixture at wide-open throttle condition is practically necessary to lower the exhaust gas temperature to prevent thermal damage to the exhaust valves, exhaust manifold, and catalytic converter.

In the warm-up duration, a certain level of excess air is required for low fuel consumption and low emission until the operating temperature of the engine is achieved and the catalytic converter starts up to lower CO and HC emissions. However, NO<sub>x</sub>

emissions are at the highest level here. Therefore, this operating condition is not allowed in practice with the engine at the operating temperature.

As the mixture becomes progressively leaner, the engine will reach or exceed its lean-misfire limit (LML) results in combustion missing which causes a rapid increase in HC emissions.

The quality of the air-fuel mixture; e.g., patterns, and mixture temperature also are important factors in determining flammability and combustion characteristics which are consequently effects on the exhaust gas constituent. In the engine starting and post-start durations, the air-fuel mixture formation are much poorer due to the prevailing low temperature resulting to manifold-wall fuel condensation increases. This causes much higher concentrations of the pollutants in the exhaust gas.

#### 2.4.1.2) Uniform Mixture Distribution

Maximum engine efficiency can be achieved when both the air intake and the injected fuel are distributed evenly among the individual cylinders. That means every cylinder should be operated with the same air-fuel ratio as much as possible.

#### 2.4.1.3) Exhaust Gas Recirculation (EGR)

EGR system is the effective way to reduce the pollution of oxides of nitrogen without increase in other pollutants in the internal combustion engines. EGR system were introduced in the early '70s and implemented to reduce the exhaust emission when the catalyst technology was not sufficient to reduce  $\text{NO}_x$  (Tairu, 2017). Higher combustion temperature induced an over-proportional increase in  $\text{NO}_x$  formation. As the exhaust gas recirculated back to the combustion chamber, the recirculated exhaust gas does not participate in the combustion and thus does not extract combustion energy, but it forms as an additional thermal mass which means that the combustion energy distributes itself to a higher overall mass, so that the peak combustion temperature in the chamber reduces. As the exhaust gas recirculation reduces the chamber temperature, it represents an effective means of controlling  $\text{NO}_x$  emissions. The exhaust gas recirculation also increases intake-manifold pressure leading to a reduction in the charge cycle work and therefore lowers the fuel consumption. Implementation of the exhaust gas recirculation can be performed both internal and external recirculation approaches by overlapping the intake valve and using EGR valves, respectively.

EPA had suggested the implementation of EGR as a technology to reduce  $\text{NO}_x$  emission, since the use of EGR measure could reduce  $\text{NO}_x$  emission by lowering maximum cylinder temperature and prevent knocking and improve fuel economy. But automobile manufacturers are reluctant to apply EGR measure at high load condition

because of combustion instability and durability problem of EGR system (Jinyoung, 2001).

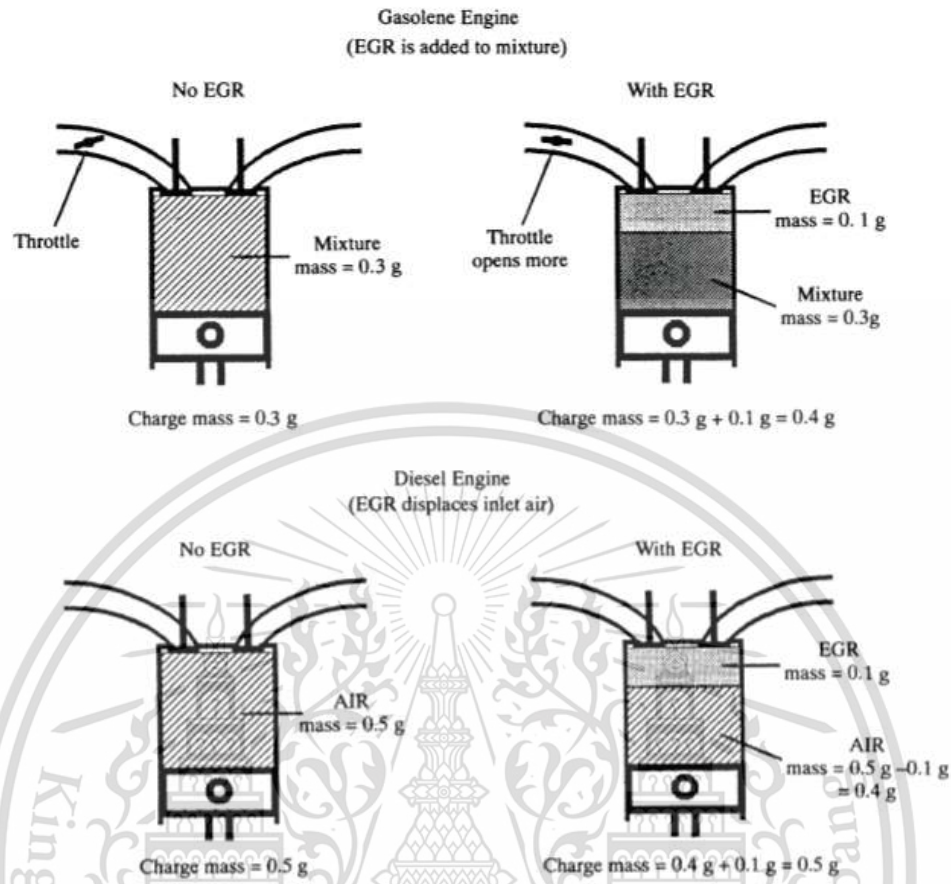
The EGR rate is defined as the mass percent of the recirculated exhaust gas in the total intake mixture. However, several reports also explained that the EGR percentage can be calculated as the recirculated CO<sub>2</sub> fraction (Domenico, 2017) (Masayuki, 2014). The fresh intake air contains negligible amounts of CO<sub>2</sub>, thus, comparing the CO<sub>2</sub> concentrations in the engine exhaust and intake is a practical way to determine the EGR rate.

Application of EGR in the gasoline engines is different from the way in which it is applied to the diesel engines. Such difference is demonstrated in Figure 2.7 (Ladommatos, 1998). Modern gasoline engines are operated with stoichiometric air-fuel mixture. Without implement of EGR, the amount of air-fuel mixture mass admitted to the engine cylinders are used to control the engine torque and power. To maintain the engine torque and power to stay constant when EGR is implemented, the mixture mass trapped in the cylinders must also constant. For this reason, the EGR mass admitted into the engine cylinders must be additional to the trapped air-fuel mixture mass. Thus, the total trapped charge composing of the air-fuel mixture plus EGR must increase. Since the engine displacement is fixed, increase in trapped mass can only be achieved by raising the trapped charge density, which necessitates further opening of the engine air throttle. An incidental benefit of this further opening of the air throttle is a reduction in engine pumping work and some increase in fuel economy.

In contrast, the diesel engines are operated without the inlet air throttle, so that inlet air is admitted into the cylinders as much as it is practicable to trap at a given engine speed. Thus, the implementation of EGR involve displacement of some of the inlet air by EGR. A consequence of this air displacement is a reduction in the available air for combustion. Since for a given torque and power output, the amount of fuel supplied to the engine must stay constant, the reduced air available for combustion lowers the air-fuel ratio at which the engine operates. This reduction in the air-fuel ratio can affect the exhaust emissions substantially.

#### 2.4.1.4) Valve Timing

Although, early opening of the intake valve increases the internal exhaust gas recirculation help to reduce NO<sub>x</sub> emissions, but also displaces the induced fresh air-fuel mixture, leading to reduction in the maximum output torque. In addition, excessively high exhaust gas recirculation, particularly at idle condition, can cause combustion misses resulting to an increase in the HC emissions.



**Figure 2.7** Comparison of the Different Ways in which EGR is used in Gasoline and Diesel Engines (Ladommatos, 1998)

## 2.4.2 Reduction of Engine External Interference

### 2.4.2.1 Crankcase Ventilation (Blowby)

The concentration of hydrocarbons in the crankcase may be many times that found in the engine's exhaust gases. Control systems conduct these gases to a suitable point in the engine's air-intake system, from where they are drawn into the combustion chamber for further burning.

### 2.4.2.2) Evaporative-emissions Control System

The Evaporative Emission Control System (EVAP) is used to prevent gasoline vapors from escaping into the atmosphere from the fuel tank and fuel system. The EVAP system usually requires no maintenance, but faults can turn on the Check Engine light and prevent a vehicle from passing an OBD II plug-in emissions test.

### 2.4.3 Exhaust Gas Treatment

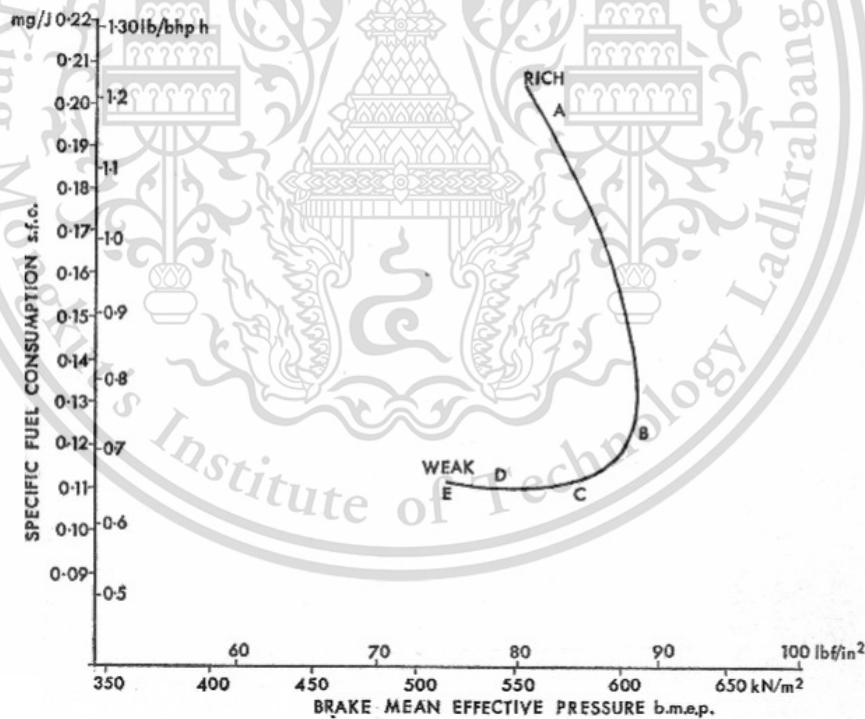
While diesel engine development aims at minimizing the production of pollutants in the combustion process itself. Exhaust gas treatment is the preferred method for gasoline engines. Catalytic converters convert the pollutants produced during combustion into harmless components.

## 2.5 Influence of Engine Factor to the Engine Performance

Several studies and various experimentation results had been carried out to study engine factors which have influent to the engine performance and are summarized as follows:

### 2.5.1 Effect of Mixture Strength upon Power and Economy

Figure 2.8 is termed a consumption loop of an gasoline engine derived from testing an engine which is run at a specific throttle position and at a constant speed while the mixture strength varied from very weak to excessively rich.

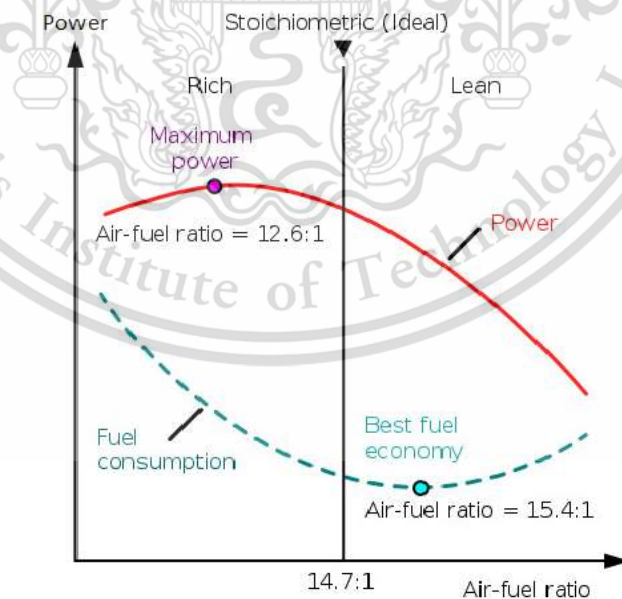


**Figure 2.8** Consumption Loop of a Gasoline Engine (Greene, 1969)

At point *A*, the large mass of fuel added to the air in the engine intake robs the cylinder of air resulting in a lower volumetric efficiency, and also the large mass of unburnt fuel in the engine cylinder cool the combustion process. The net result is a loss

in brake mean effective pressure (b.m.e.p.) or brake power (b.p.) or torque. As the richness is reduced, point *B* is obtained at which the power is the maximum. Point *C* indicates stoichiometric condition and, at a point slightly weaker than stoichiometric, the maximum thermal efficiency of optimum fuel economy is achieved at point *D*. If the mixture strength is weakened still further, the speed of the combustion process is reduced such that at point *E*, burning is taking place throughout the expansion stroke and the subsequent exhaust stroke. Weaker than conditions at *E* results in such poor combustion that a flame is present when the inlet valve opens on the next cycle, resulting in ignition of the new charge as it enters the cylinders. At very weak mixtures, the output power is lost, and the thermal efficiency is low while the exhaust valves run hot, because burning is still taking place at the end of the expansion process and during the exhaust process. Figure 2.8 also informs that the range of power variation (between point *A* to point *E*) in a gasoline engine employing a control of fuel input is substantial small. It is necessary to control the air input also. Hence, the power output of a gasoline engine must be controlled by throttling the air intake and not by mixture control as in a compression ignition engine, while the mixture strength should be near optimum throughout the throttle range.

Mixture strength or the ratio of amount of air to fuel consumption which provide the maximum power output is one of about 20% richer than the mixture proportions giving the complete combustion, that is, of value about 14.7 kg of air to kg of fuel. And the mixture which yields the best fuel economy for a given power output is one of about 13% more air than the complete combustion's mixture strength (Judge, 1943). Figure 2.9 shows the engine power and fuel consumption function of air-fuel ratio.



**Figure 2.9** Engine Power and Fuel Consumption Function of Air-fuel Ratio

Source : <http://www.x-engineer.org>

Figure 2.10 shows the consumption loops at full throttle, half throttle, and at low throttle opening for a gasoline engine comparing with that of the compression ignition engine running at the maximum torque speed. The engine operations are assumed that maximum economy conditions prevail at part throttle with a richening of the mixture strength at full throttle to give a maximum power, so that the specific fuel consumption against b.m.e.p. line is the dotted line  $B, D, D_1, D_2$  in the figure. The line shows the penalty of throttling the air inlet by the progressive drop in power, but the specific fuel consumption at low throttle opening rises to more than its full throttle value. The lower specific fuel consumption of the compression ignition engine is due to the higher compression ratio of the engine compared with the rather low value for the gasoline engine.

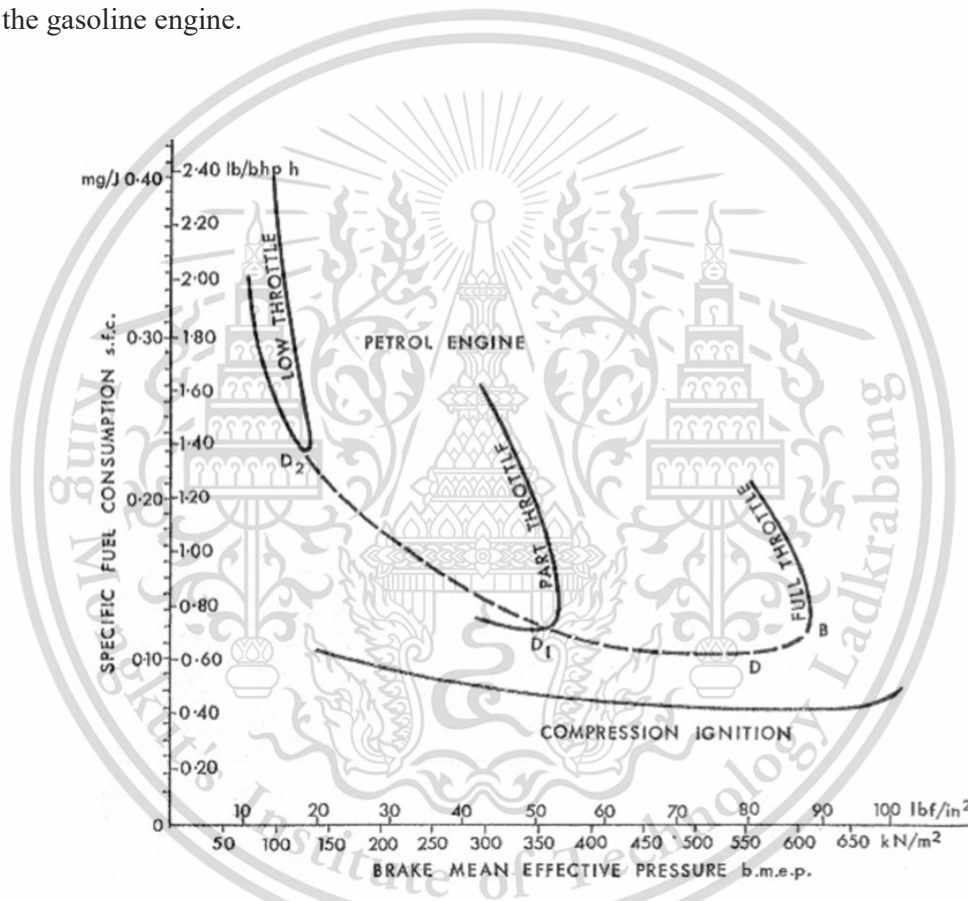
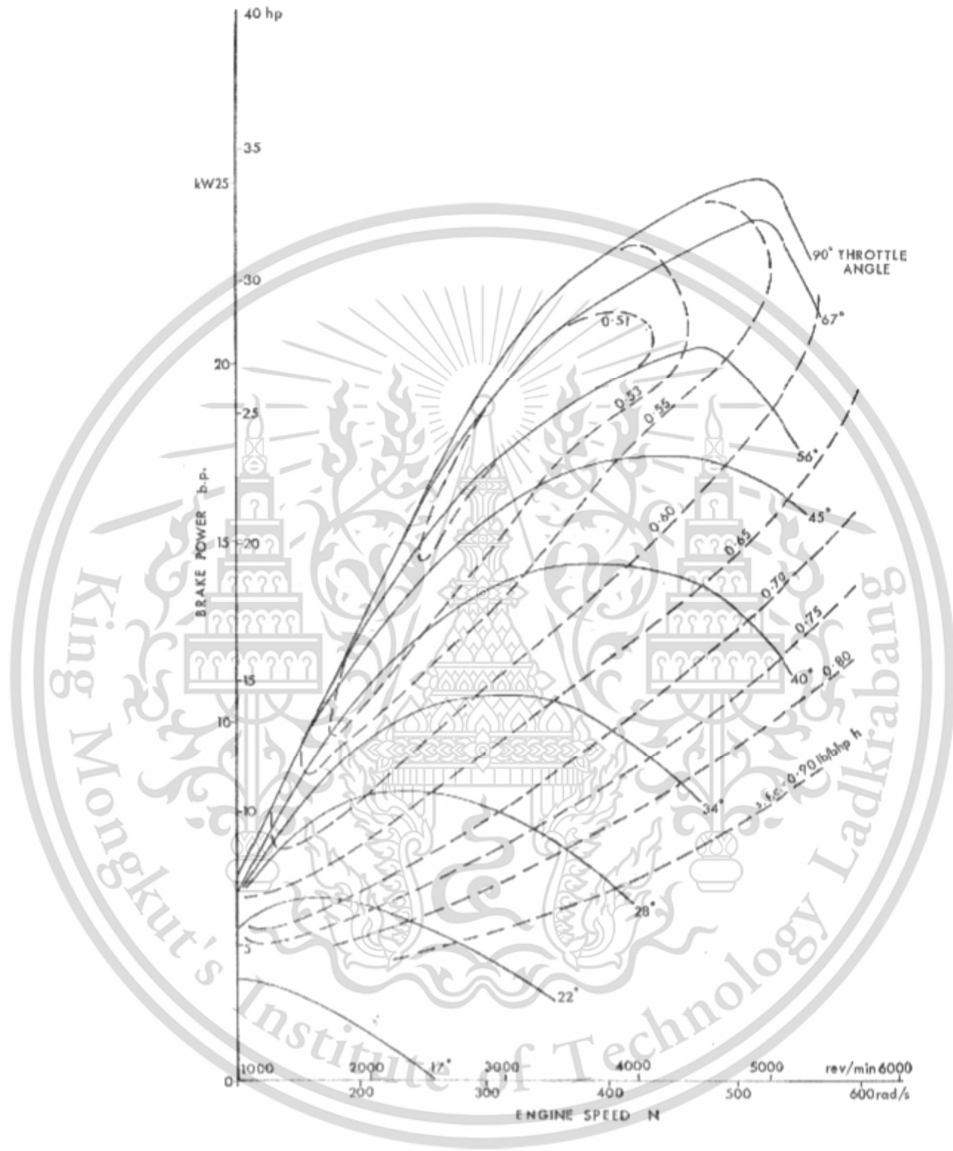
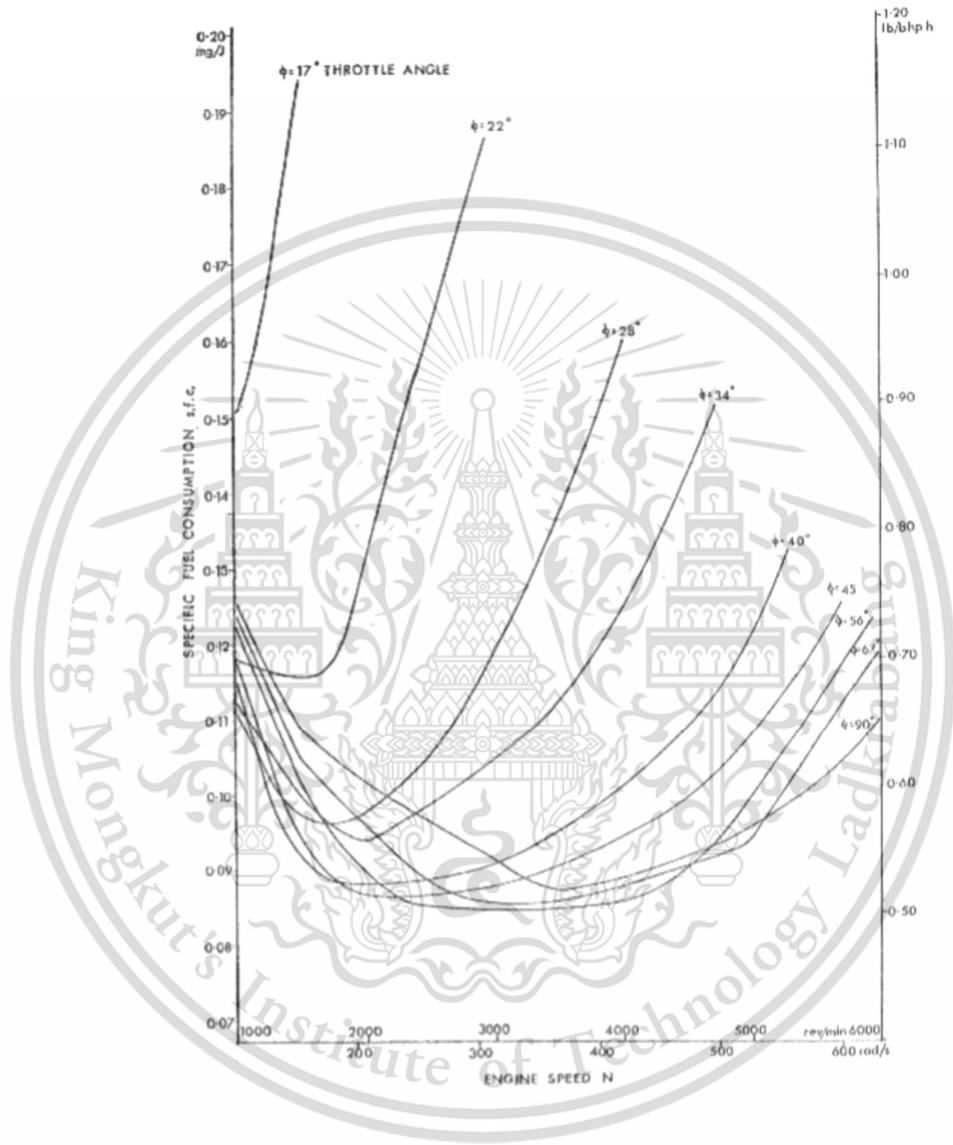


Figure 2.10 Consumption Loop of a Gasoline Engine (Greene, 1969)

Figure 2.11 and 2.12 demonstrate power and specific fuel consumption characteristics at different throttle angles of a gasoline engine. It is worth noting that the effect of throttling the air in the engine intake. By reducing the throttle angle and so reducing the air flow rate into the engine, not only is the power output of the engine reduced, but the maximum brake power speed drops. And reducing the throttle angle also causes increase of pumping work during the induction stroke resulting to the increase of the specific fuel consumption too.



**Figure 2.11** Power Characteristics at Different Throttle Angles of a Gasoline Engine (Greene, 1969)



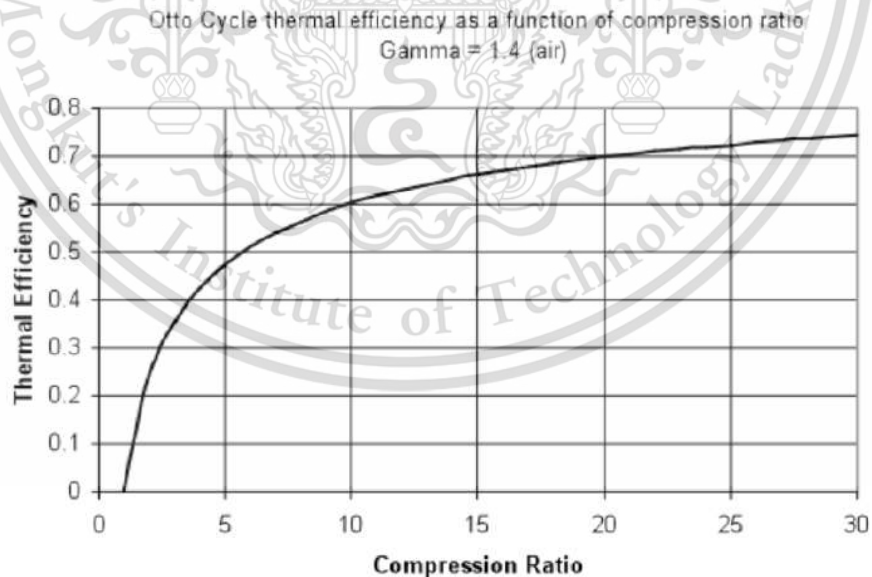
**Figure 2.12** Specific Fuel Consumption Characteristics of a Gasoline Engine (Greene, 1969)

### 2.5.2 Effect of Nature of Fuel

Various in the engine performance and output may be obtained when using different fuels in both the composition and the heating value. Generally, composition of the liquid fuel employed in the internal combustion engines consists of the element of carbon (C), hydrogen (H), and occasionally oxygen (O), with traces of impurities in a few instances. The calorific or heating value of a fuel is the amount of heat given out by the complete combustion of one-unit weight of the fuel with air or oxygen. It was found that the higher the hydrogen content, the higher is the heating value, and that the effect of oxygen content in the fuel is to reduce the available heating value.

### 2.5.3 Effect of Compression

From both theoretical considerations of air cycle efficiencies and practical test results, it has been shown that under normal running conditions, without detonation, thermal efficiency of a gasoline engine depends upon compression ratio. Both the explosion pressure and the mean effective pressure (m.e.p.) increase progressively with the compression pressure, while the brake specific fuel consumption (bsfc) diminishes with increase in the compression pressure. Therefore, the use of higher compression ratio engine is beneficial. Figure 2.13 demonstrates variation of the thermal efficiency and the compression ratio.



**Figure 2.13** Thermal Efficiency and Compression Ratio

Source : <http://www.dukeengines.com>

## CHAPTER 3

### RESEARCH METHODOLOGY

The experiments were performed under idle conditions and partial load conditions. In the idle tests, the tested engine was operated with varying of the diluent's composition to investigate effect of the diluent's composition on the engine emissions. In the partial load tests, the tested engine was operated with diluting the air intake at normal operation speed range to investigate change of the engine performance. The results were compared with the engine operating with its original configuration without the dilution to the air intake. Details of the study were described subsequently as follows.

#### 3.1 Introduction

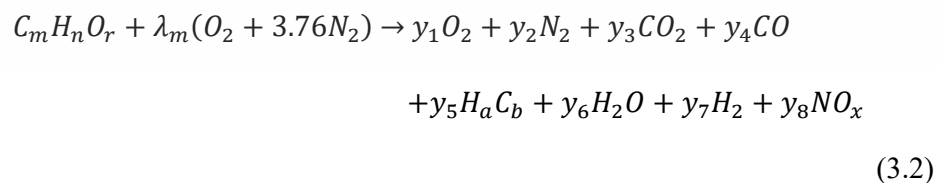
In the gasoline engine, an overall combustion process can be written as the following equation:



Generally, the fuel is hydrocarbon fuel, the oxidizer normally is air ( $O_2 + 3.76N_2$ ), and the products are  $CO_2$ ,  $H_2O$ ,  $CO$ ,  $H_2$ ,  $O_2$ ,  $NO_x$ ,  $N_2$ , unburned hydrocarbon (HC), and soot particles which are mainly solid carbon and usually can be omitted from the analysis. Among of the combustion products,  $CO$ ,  $HC$ ,  $NO_x$ , and  $CO_2$  are considered as pollutants (Blair and Gordon, 1999).

Air-fuel ratio is one of the most important parameter that affects the engine exhaust emissions. Variation in emissions with air-fuel ratio for any typical premixed gasoline engines is demonstrated in reference (<http://www.egr.msu.edu>).

The actual combustion equations including the effects of dissociations for any generic hydrocarbon fuel containing alcohols  $C_mH_nO_r$  in the internal combustion engine can be demonstrated explicitly as



where  $\lambda_m$  is the molecular air-fuel ratio, and  $y_1, y_2, y_3, y_4, y_5, y_6, y_7, y_8$  are the number of mole of the combustion products,  $a$  and  $b$  are the number of hydrogen atom and carbon atom in the unburned hydrocarbon.

To solve for the unknowns in Eq. (3.2), various assumptions are determined, and it is required carbon balance, hydrogen balance, oxygen balance, nitrogen balance, and additional equations to complete the analysis. And it should be pointed out that there are various ways of recording the exhaust emissions as values “equivalent to a reference gas”. In measurement of unburned hydrocarbon (HC), either by an NDIR (Non-dispersive Infrared) device or by a FID (Flame Ionization Detector), the readings are quoted as ppm hexane (C<sub>6</sub>H<sub>14</sub>) or ppm methane (CH<sub>4</sub>), respectively. Therefore, the molecular weights for hexane or methane must be inserted in the appropriated equation if the pollutant gas is to be regarded as equivalent hydrocarbons to either hexane or methane.

Assumption of ideal-gas behavior for all gaseous species and gas mixtures is appropriate for nearly all the systems to be considered. Since the high temperature associated with the combustion process generally result in sufficiently low densities for ideal-gas behavior to be a reasonable approximation (Stephen, 1996).

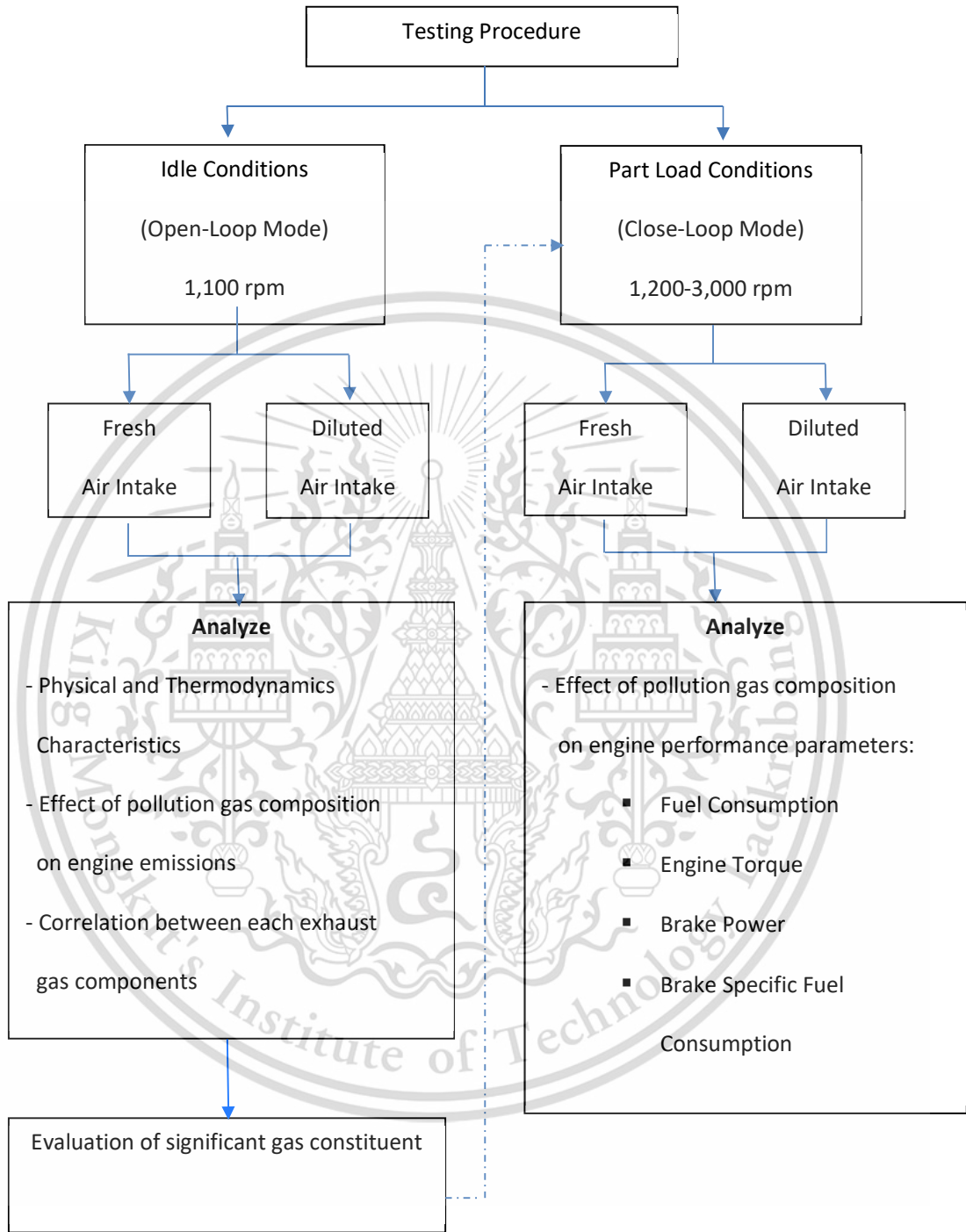
### **3.2 Experimental Methodology**

In this study, the experimentation was carried out in the engine testing which was tested on the engine model: Toyota 5A-FE, 1500-cc engine. There were two conditions selected to test: 1) testing at the engine’s idle speed and 2) testing at the normal operating speed range. Details of the testing procedure of the study were described in Figure 3.1.

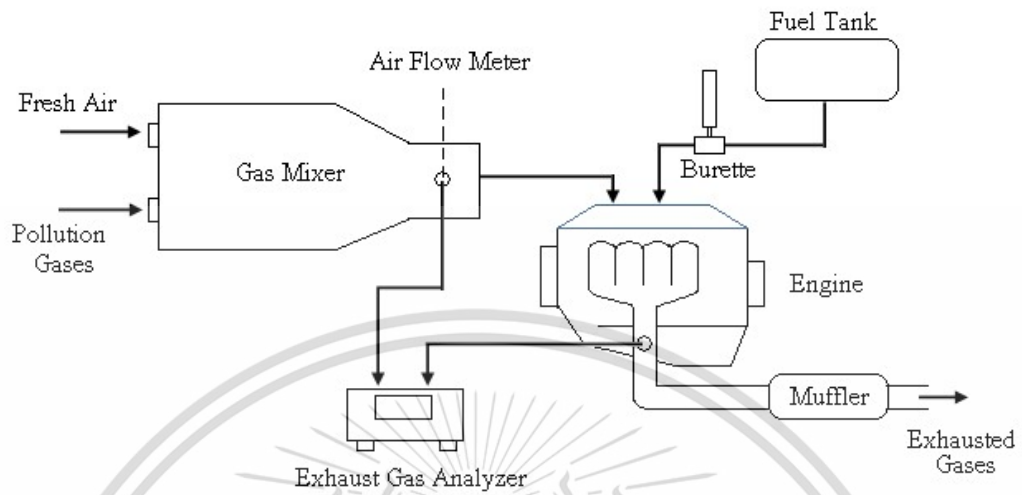
#### **3.2.1 Testing at the engine’s idle condition**

To analyze effects of the intake air’s composition to the exhaust gas’s constituent, the engine’s oxygen sensor had been removed to cut off input signal from the oxygen sensor, so that the engine’s ECU operates in open loop mode and orders the engine to run at the constant fuel injection rate or fixed fuel mixture as the manufacturer’s set fuel injection rate. Two vehicles with and without catalytic converter were employed to generated various composition of CO, CO<sub>2</sub>, and HC content in pollution gases from combusting gasohol-91 fuel. The pollution gases of which various proportion was fed and diluted to the intake fresh air in a mixing tank upstream to the tested engine. Corresponding change in the compositions and temperature of the exhaust gases from the exhaust pipe before entering the catalytic converter were sampled and analyzed through Exhaust Gas Analyzer EMS Model 5002 to determine the concentrations of CO, CO<sub>2</sub>, HC, and excess oxygen O<sub>2</sub>.

Schematic diagram described the details of equipment and sampling measurement points are shown in Figure 3.2 and photography of the experimental apparatus employed for this study are demonstrated in Figure 3.3 as below.



**Figure 3.1** Flow Chart of the Testing Procedure



**Figure 3.2** Schematic Diagram of the Experimentations



**Figure 3.3** Photography of Experimental Apparatus

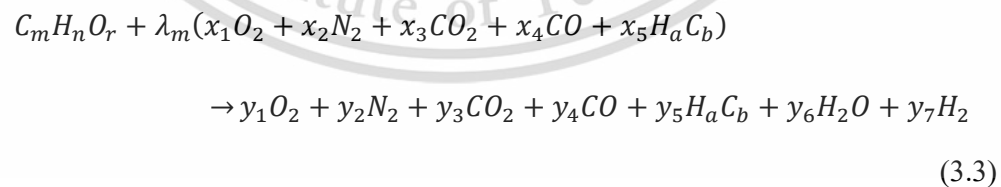
Results of the concentrations of the exhaust CO, CO<sub>2</sub>, HC, and excess oxygen O<sub>2</sub> from the exhaust gas analysis were analyzed and considered to find out the effect of the physical characteristics, density and specific heat, and the composition of the diluted air intake to the emission gas constituent from the tested engine. Then, parameters of the diluted air intake which affect to the constituent of the exhaust gas will be considered and further tested. to calculate amount of the fuel consumption under the normal operating conditions with having the oxygen sensor installed and under closed-loop control condition. Such calculated fuel consumption values will be compared to the values of which the normal fresh air intake was employed to combustion.

### 3.2.2 Testing at the normal operating speed range

After analyzing the experimental results, component of the diluted air intake which has significant effect to the exhaust emission will be further considered. The experimental testing based on the normal operating speed range at fixed throttle position will be performed additionally. Desired speed is maintained by load adjustment and required engine load is obtained through the dynamometer control. Comparison of the engine performance, e.g., output torque (T), brake power (BP), fuel consumption, and brake specific fuel consumption (bsfc) to normal air intake will be carried out and discussed subsequently.

### 3.3 Mathematical Model for Estimation of Exhaust Gas Composition

To predict the exhaust gas composition, mathematical model is established. It is well aware that using the exhausted gas as EGR is purposely to reduce NO<sub>x</sub> formation. However, the gasoline engines were typically operated in the fuel-rich condition or close to the stoichiometric condition, so that the amount of NO<sub>x</sub> in the exhaust tails is comparatively much less than the others; CO, CO<sub>2</sub> and HC. Only trace amount of NO was recorded from all tests and the combustion products, NO<sub>x</sub> was assumed to transform to N<sub>2</sub>. Hence, the equation of combustion in the gasoline engine with polluted air can be modified from Eq. (3.2) as follow:



where the molecular air-fuel ratio  $\lambda_m$  is derived from experimental measurement at the engine's idle speed and  $x_1, x_2, x_3, x_4, x_5$  are the mole fraction of the diluted air composing of dry pollutant gases and induced fresh air which are measured from the exhaust gas analyzer.

so that,

$$x_1 + x_2 + x_3 + x_4 + x_5 = 1 \quad (3.4)$$

According to the specification of the gas analyzer, the equipment uses a sensors bench, NDIR for HC, CO, and CO<sub>2</sub> measurement, while HC emissions are measured as hexane (C<sub>6</sub>H<sub>14</sub>), thus,  $a = 14, b = 6$ . Then, there are 7 unknowns to be solved in equation (3.3), e.g.,  $y_1, y_2, y_3, y_4, y_5, y_6, y_7$  which are needed to be solved simultaneously through carbon balance, hydrogen balance, oxygen balance, nitrogen balance, and additional three equations as follows:

Carbon balance:

$$m + \lambda_m(x_3 + x_4 + 6x_5) = y_3 + y_4 + 6y_5 \quad (3.5)$$

Hydrogen balance:

$$n + \lambda_m(14x_5) = 14y_5 + 2y_6 + 2y_7 \quad (3.6)$$

Oxygen balance:

$$r + \lambda_m(2x_1 + 2x_3 + x_4) = 2y_1 + 2y_3 + y_4 + y_6 \quad (3.7)$$

Nitrogen balance:

$$\lambda_m(2x_2) = 2y_2 \quad (3.8)$$

Another three equations implemented to solve the solution derived from the experimental data between the exhaust gas CO<sub>2</sub>-O<sub>2</sub>, the exhaust gas CO-CO<sub>2</sub> and the exhaust gas CO<sub>2</sub>-H<sub>2</sub> [4] are as below

$$[CO_2] = 14.023 - 0.67[O_2] \quad (3.9)$$

$$[CO] = 22.25 - 1.66[CO_2] \quad (3.10)$$

$$[H_2] = 0.4 * \{[CO_2] - 0.22\} \quad (3.11)$$

where [CO], [CO<sub>2</sub>], [O<sub>2</sub>] and [H<sub>2</sub>] are the percentage by volume of CO, CO<sub>2</sub>, O<sub>2</sub>, and H<sub>2</sub>, respectively.

To find out the solutions of the equations, Excel Solver were employed to solve the equations. The Excel Solver is a built-in tool for optimization and curve fitting which is included in Microsoft Excel. The Solver model's single output variable to be minimized or maximized is called the Objective Function. The model's input variables that the Solver will adjust are called the Decision Variables. The Solver will calculate the set of values for all the model's input variables that will result in a minimum or maximum value of its output variable. Problems solved using the Solver normally must have conditions placed on variables within the model, so that the outcome will be reasonable. Such conditions are called Constraints (Harmon, 2012).

### 3.4 Calculations of Density of the Air Intake

Density of ideal gaseous mixture can be calculated from the following equation:

$$\rho_m = \rho_1 + \rho_2 + \dots + \rho_N$$

$$\rho_m = \frac{y_1 PM_1}{R_u T} + \frac{y_2 PM_2}{R_u T} + \dots + \frac{y_N PM_N}{R_u T} \quad (3.12)$$

Where

$\rho_m$  = mean density of the mixing gas (kg/m<sup>3</sup>)

$\rho_i$  = density of the i gas element (kg/m<sup>3</sup>)

$y_i$  = volume fraction of the i gas element

$M_i$  = molecular mass of the i gas element (kg)

$P$  = total absolute pressure (kPa)

$T$  = absolute temperature (K)

$R_u$  = universal gas constant = 8.314 kJ/kmol-K

As the engine operating conditions change, there is a corresponding change in the composition and physical nature of the exhaust gas. When gasoline or diesel fuels are used, the ratio of the added mass and added volume from the combustion will be close to the density of air. If the excess air ratio is larger than 1, the density of emission gas is nearly equal to the air density (Adachi, 2014).

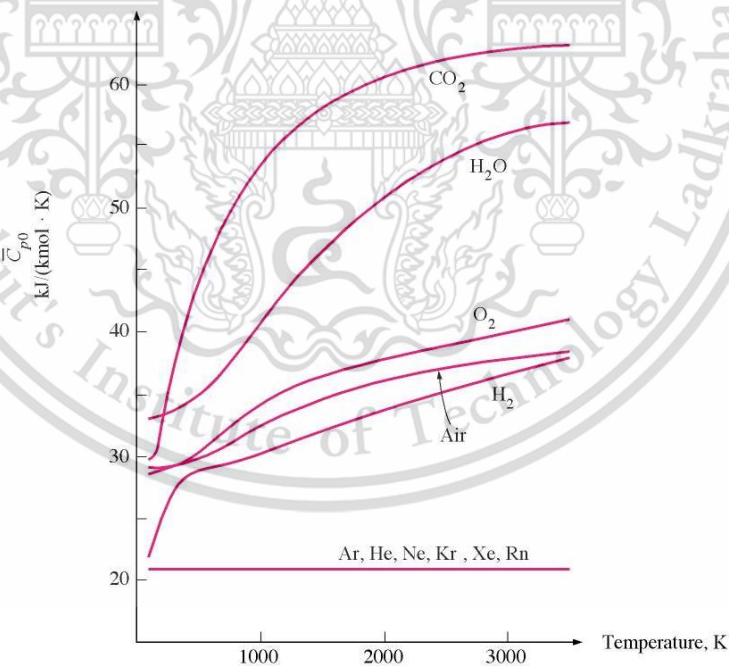
### 3.5 Calculations of Specific Heat of the Air Intake

Specific heat of gaseous mixture can be calculated from the following equation:

$$c_p = y_1 c_{p,1} + y_2 c_{p,2} + \dots + y_N c_{p,N} \quad (3.13)$$

Where  $c_p$  = mean specific heat of the mixing gas (J/mole-K)  
 $c_{p,N}$  = specific heat of the N gas element (J/mole-K)

Figure 3.4 demonstrates the molar specific heat of some idea gases. The value of the mass specific heat can be obtained from the value of the molar specific heat by dividing by its molecular weight (molar mass). It can be observed that the specific heats are constant for monatomic gases and vary more strongly with temperature for triatomic gases, for example CO<sub>2</sub> and H<sub>2</sub>O, than for diatomic gases. The Exhaust Gas Recirculation (EGR) system utilizes this phenomenon by recirculating a portion of the exhaust gas back into the intake manifold to increase the specific heat capacity of the combustion gas and act as a heat sink, consequently lowering the combustion temperature and reduce the amount of NO<sub>x</sub> formation inside the engine cylinder.



**Figure 3.4** Specific Heat of Ideal Gases

Source : <http://www.mhhe.com/engcs/mech/cengel/notes/IdealGases.html>

### 3.6 Calculations of Adiabatic Flame Temperature

We define two adiabatic flame temperatures, one for constant-pressure combustion and one for constant-volume combustion. If a fuel-air mixture burns adiabatically at constant pressure as the gasoline engine, the absolute enthalpy of the reactants at the initial state ( $T=298$  K,  $P= 1$  atm) equals the absolute enthalpy of the products at the final state ( $T=T_{ad}$ ,  $P= 1$  atm) as follow:

$$H_{react}(T_i, P) = H_{prod}(T_{ad}, P)$$

$$\sum_{react} N_i \bar{h}_i = \sum_{prod} N_i [\bar{h}_{f,i}^0 + \bar{c}_{p,i}(T_{ad} - 298)] \quad (3.14)$$

Where

$N_i$  = number of mole (kmol)

$\bar{h}_{f,i}^0$  = enthalpy of formation of specie i (kJ/kmol)

$\bar{c}_{p,i}$  = specific heat (kJ/kmol-K)

### 3.7 Calculations of Engine Performance

Relevant theories and equations for the calculations of engine performance based on the engine test bed experiments are briefly mentioned and demonstrated as follow:

#### 3.7.1 Air Consumption

Air mass flow rate is determined from volumetric flow rate of induction air to the manifold by using a propeller-type air flow rate sensor installed at the inlet of the engine suction duct. As the air flows through the duct, the propeller rotates and generates pulses which can be shown on an indicator. Therefore, the air mass flow rate is:

$$m_{air} = \rho_{air} \cdot q_{air} \quad (3.15)$$

Where

$m_{air}$  = air mass flow rate (kg/s)

$\rho_{air}$  = air density (kg/m<sup>3</sup>)

$q_{air}$  = air volumetric flow rate (m<sup>3</sup>/s)

### 3.7.2 Fuel Consumption

Fuel consumption is obtained through measurements of fuel flow rate. A graduated cylinder and a stop watch is generally used for such measurement of the fuel flow rate, and also, a digital fuel flow sensor is installed in series with the graduated cylinder as an alternate method for measuring the fuel flow rate whose values are shown on an indicator at the control panel. Quantity of the fuel consumption is determined from:

$$m_f = \frac{\rho_f \cdot q_f}{1000} \quad (3.16)$$

Where  $m_f$  = fuel consumption (kg/s)  
 $\rho_f$  = fuel density (kg/m<sup>3</sup>)  
 $q_f$  = volume of fuel used in a period of measurement (L/s)

### 3.7.3 Air-Fuel Ratio

Air-fuel ratio is the ratio of the air mass flow rate to the fuel consumption which can be calculated from:

$$A/F = \frac{m_{air}}{m_f} \quad (3.17)$$

Where  $A/F$  = air-fuel ratio  
 $m_{air}$  = air mass flow rate (kg/s)  
 $m_f$  = fuel consumption (kg/s)

### 3.7.4 Volumetric Efficiency

Volumetric efficiency is the ratio of actual air flow rate to theoretical displacement volume which can be determined from:

$$\eta_v = \frac{q_{air}}{q_{th}} \times 100\% \quad (3.18)$$

Where  $\eta_v$  = volumetric efficiency (%)  
 $q_{air}$  = air volumetric flow rate (m<sup>3</sup>/s)  
 $q_{th}$  = theoretical displacement volume (m<sup>3</sup>/s)

For four-stroke internal combustion engine, the theoretical displacement volume,  $q_{th}$ , can be calculated from:

$$q_{th} = \frac{1}{60} \times \frac{N}{2} \times \text{Total Engine Displacement Volume} \quad (3.19)$$

Where  $N$  = engine speed (rpm)

### 3.7.5 Engine Brake Power

The engine brake power can be determined from following relation:

$$BP = \frac{2\pi TN}{60 \times 1000} \quad (3.20)$$

Where  $BP$  = engine brake power (kW)

$T$  = engine torque (N-m)

$N$  = engine speed (rpm)

In case a load cell with an indicator is employed, the engine torque can be measured directly from the digital torque indicator. In case of Prony brake with a spring balance is used, the engine torque can be derived from:

$$T = F \cdot R \quad (3.21)$$

Where  $F$  = net force from spring balance or load cell (N)

$r$  = load arm length (m)

### 3.7.6 Brake Specific Fuel Consumption

Brake specific fuel consumption is amount of fuel consumption per unit brake power derived from:

$$\text{BSFC} = \frac{3600 \cdot m_f}{BP} \quad (3.22)$$

Where  $bsfc$  = brake specific fuel consumption (kg/kWh)

$m_f$  = fuel consumption (kg/s)

$BP$  = engine brake power (kW)

### 3.7.7 Combustion Heat from Fuel

Combustion Heat of the fuel in the induction mixture is the amount of heat energy available by combustion of the fuel and air which depends upon the amount of oxygen present in the induced air. Estimation of such quantity can be determined from:

$$Q_f = 3600 \times m_f \times \text{LHV} \quad (3.23)$$

Where  $Q_f$  = combustion heat of the fuel (kW)

$m_f$  = fuel consumption (kg/s)

$\text{LHV}$  = lower heating value of the fuel (kJ/kg)

### 3.7.8 Brake Thermal Efficiency

The difference between the fuel energy and the power output from engine is due to heat loss to cooling water and the exhaust gas, heat transfer from the engine to the surrounding air, and also, incomplete combustion process. Brake thermal efficiency is the amount of useful work obtained from the engine in relation to the heat energy supplied by the combustion of the fuel which can be determined from:

$$\eta_{th,b} = \frac{BP}{Q_f} \times 100\% \quad (3.24)$$

Where  $\eta_{th,b}$  = brake thermal efficiency (%)

$BP$  = engine brake power (kW)

$Q_f$  = combustion heat of the fuel (kW)

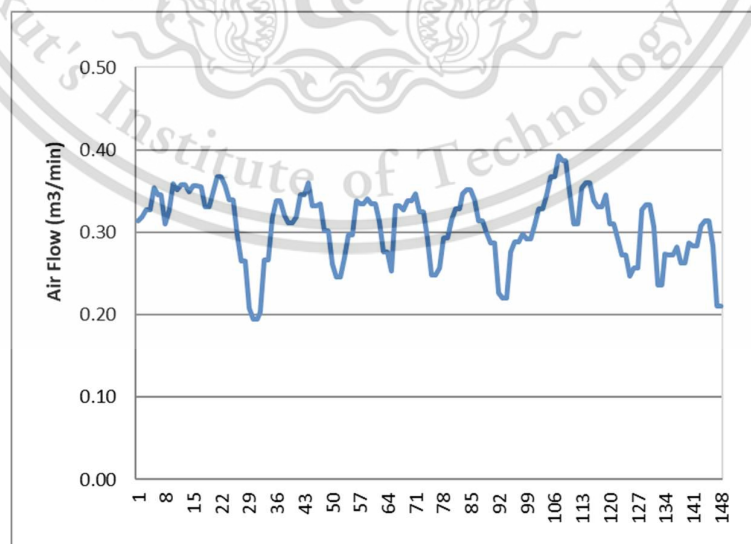
## CHAPTER 4

### RESULTS AND DISCUSSIONS

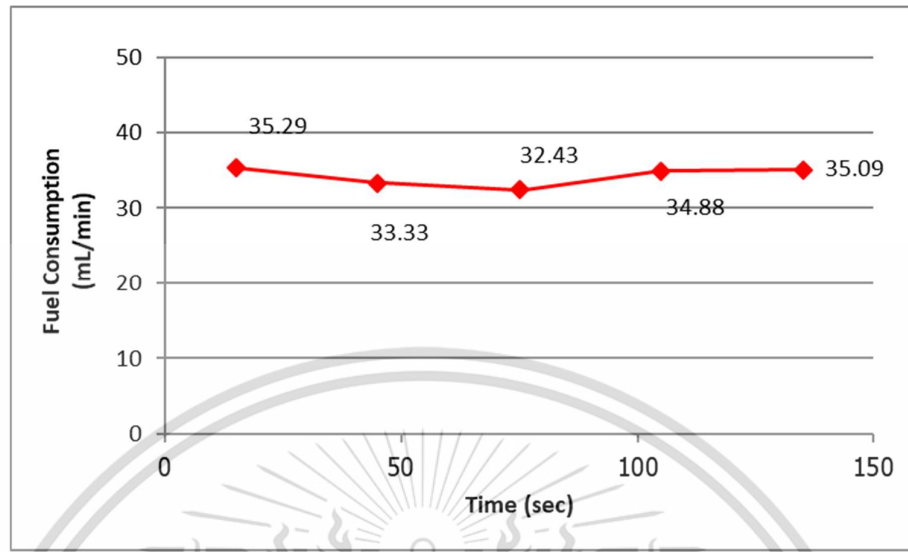
Experimental results of testing at the engine's idle condition were analyzed to investigate the effects of physical properties and composition of the diluted air intake to the emission characteristic of the exhaust gas combusted from the tested gasoline engine. And then, the engine performance such as engine torque, brake power, fuel consumption, and brake specific fuel consumption were measured and compared to the fresh air intake from results of testing the engine at partial load condition. Complete experimental results for each experimentation are presented at the end of the thesis in Appendix D.

#### 4.1 Results of Testing at the Engine's Idle Condition

The effects of the diluted air intake's composition to the exhaust gas's constituent were studied by unplugging the engine's oxygen sensor to operate the engine at open-looped control so that the amount of the fuel is injected at constant rate. The data of the intake air flow rate and the fuel consumption at the idle speed were recorded for confirming the open-looped control condition was illustrated in Figure 4.1 and Figure 4.2, respectively. The measured data were recorded by employing a data-logging, hot-wire anemometer. Both recorded results shown that the intake air flow and the fuel consumption were rather constant. Average value of the air flow rate is  $0.3086 \text{ m}^3/\text{min}$  and average value of the fuel consumption is  $34.21 \text{ mL}/\text{min}$ . Therefore, average value of air-fuel ratio is about 12.994 kg of air per kg of fuel approximately.



**Figure 4.1** Variation of Air Flow Rate at Idle Condition



**Figure 4.2** Variation of Fuel Consumption at Idle Condition

Table 4.1 demonstrates measured data of exhaust gas component from the tested engine which employed fresh air intake. The results show the gas analyzer could measure only 4 exhaust gases, e.g. HC, CO, CO<sub>2</sub>, and O<sub>2</sub>, excepting NO<sub>x</sub>. The exhaust concentration of each gas seems to be constantly. Therefore, scope of the study and subsequent discussion will be limited and considered to these 4 exhaust gases.

**Table 4.1** Exhaust Gas Component of Fresh Air Intake

No.	HC (ppm)	CO (%vol)	CO <sub>2</sub> (%vol)	O <sub>2</sub> (%vol)	NO <sub>x</sub> (ppm)
1	238	1.00	12.6	2.6	0
2	238	1.00	12.7	2.4	0
3	239	0.99	12.7	2.5	0
4	238	1.00	12.6	2.3	0
5	239	0.99	12.6	2.3	0

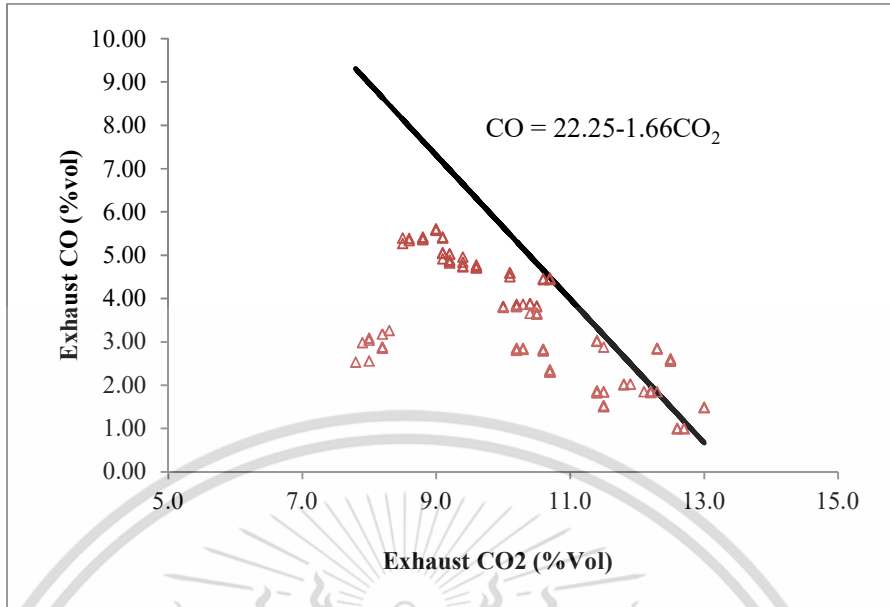
In the first experimentations, various compositions of CO, CO<sub>2</sub>, and HC content in pollution gases were generated from combusting gasohol-91 fuel, and then diluted to the intake fresh air in a mixing tank upstream to the tested engine. In these experimentations, the tested pollution gases had varying concentration of CO in range of 0-1.40 %vol., CO<sub>2</sub> in range of 0-5.0 %vol., and HC in range of 0-400 ppm. Measured data of compositions and temperature of the exhaust gases from the exhaust pipe before entering the catalytic converter are sampled and analyzed through Exhaust Gas Analyzer EMS Model 5002 to determine the concentrations of the exhaust HC, CO, CO<sub>2</sub>, and excess oxygen O<sub>2</sub>. Physical characteristic of the varying composition of the diluted air intake were calculated and analyzed correlation of its values to the variation of the polluted air composition. The effects of varying proportion of each gas element in the polluted air intake and the experimental data derived from analysis of the dry exhaust gas composition, e.g., HC, CO, CO<sub>2</sub>, and O<sub>2</sub> were considered and explained according to each intake elements. And then, component of the exhaust gas composition will be considered for correlation based on prior studies.

#### **4.1.1 Correlation of Exhaust Gas Composition**

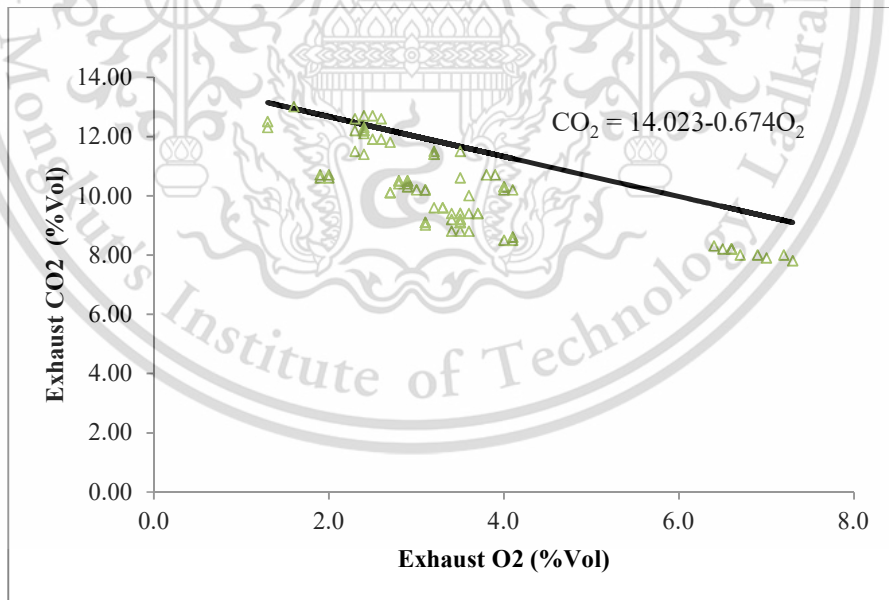
Comparisons of the experimental results to the available correlation of the exhausted CO and CO<sub>2</sub>, and the exhausted O<sub>2</sub> and CO<sub>2</sub> from the prior study in normal fresh air intake (Graf, 1934) are demonstrated as shown in Figure 4.3 to Figure 4.4. The correlations can be approximated to a linear equation as the described relations in the figure.

The amount of CO+CO<sub>2</sub> in the exhaust gas implies total carbon to be burned in the engine. In Figure 4.3, for normal fresh air intake, the concentration of the exhausted CO increases as the exhausted CO<sub>2</sub> decreases corresponding to equation (3.9). For the polluted air intake, such relation is also consistent, but the amount of the exhausted CO is lower than the normal fresh air intake. However, these results could be confirmed that use of equation (3.9) for establishing the mathematical model is reasonable.

In Figure 4.4, for normal fresh air intake, the concentration of the exhausted CO<sub>2</sub> decreases linearly as the exhausted O<sub>2</sub> increases corresponding to equation (3.10). For the polluted air intake, such relation is also consistent, but the decreasing trend of the exhausted CO<sub>2</sub> is lower than the normal fresh air intake. This means the completeness of combustion is lower than the fresh air intake. These results could be confirmed that use of equation (3.10) for establishing the mathematical model is reasonable too.

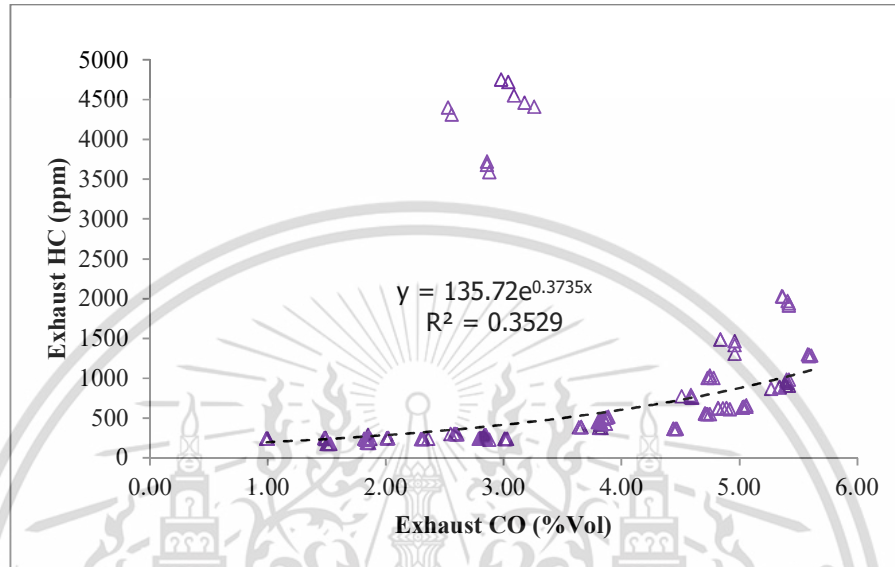


**Figure 4.3** Comparison of the Experimental Data to the Correlation of the Exhaust CO<sub>2</sub> and the Exhaust CO



**Figure 4.4** Comparison of the Experimental Data to the Correlation of the Exhaust O<sub>2</sub> and the Exhaust CO<sub>2</sub>

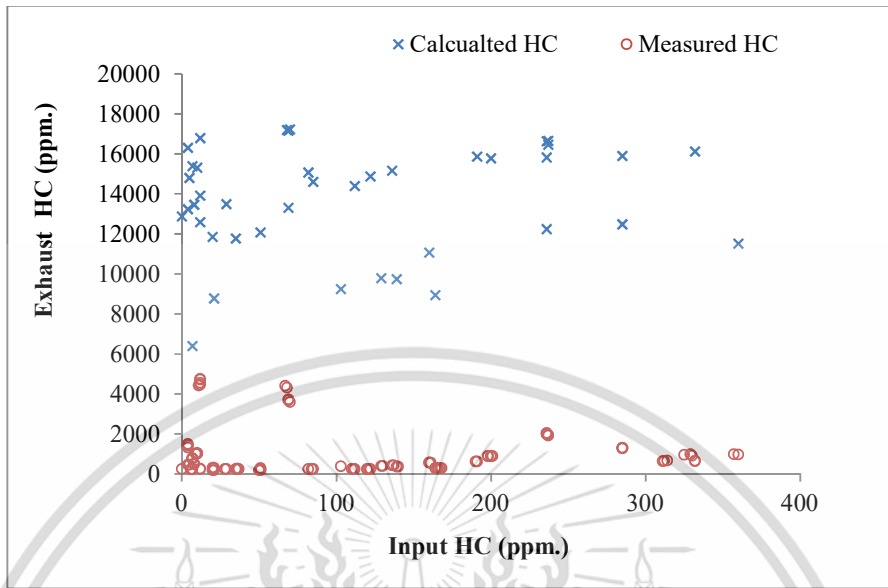
Figure 4.5 demonstrates that the concentration of the exhausted CO and the exhausted HC has no obvious relation which is consistent to the prior study (Claudio, 2004).



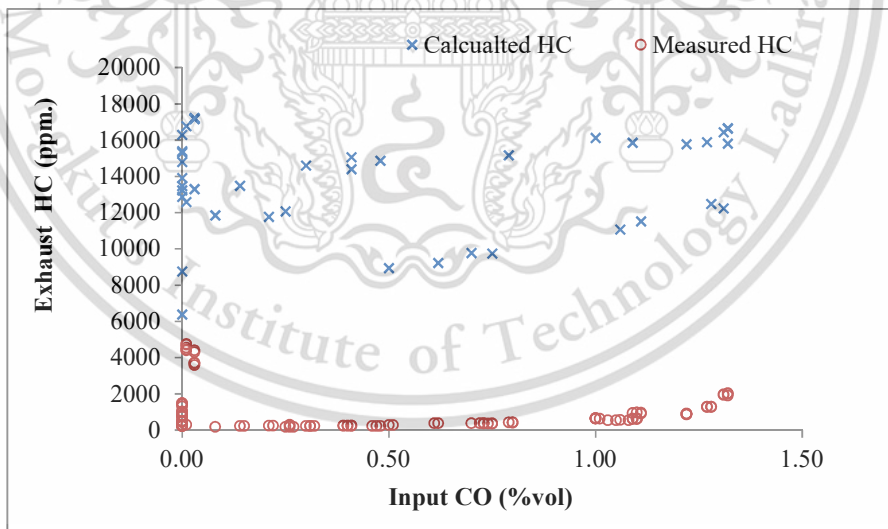
**Figure 4.5** Comparison of the Experimental Data to the Correlation of the Exhaust CO and the Exhaust HC

#### 4.1.2 Estimations of the Exhaust Gas Composition

Figure 4.6-(a) to Figure 4.6-(d) demonstrates comparison of the variation of the exhaust HC to each intake polluted air element  $O_2$ ,  $CO_2$ ,  $CO$ , and  $HC$ , respectively, between the results measured from the experimentations and the values calculated from the mathematical model. The calculated values of HC seem to provide good predictive tendency between the exhausted HC and each intake constituent. However, the calculated values derived from the established model and the measured values are considerably different.



**Figure 4.6-(a)** Exhaust HC corresponding to Input HC



**Figure 4.6-(b)** Exhaust HC corresponding to Input CO

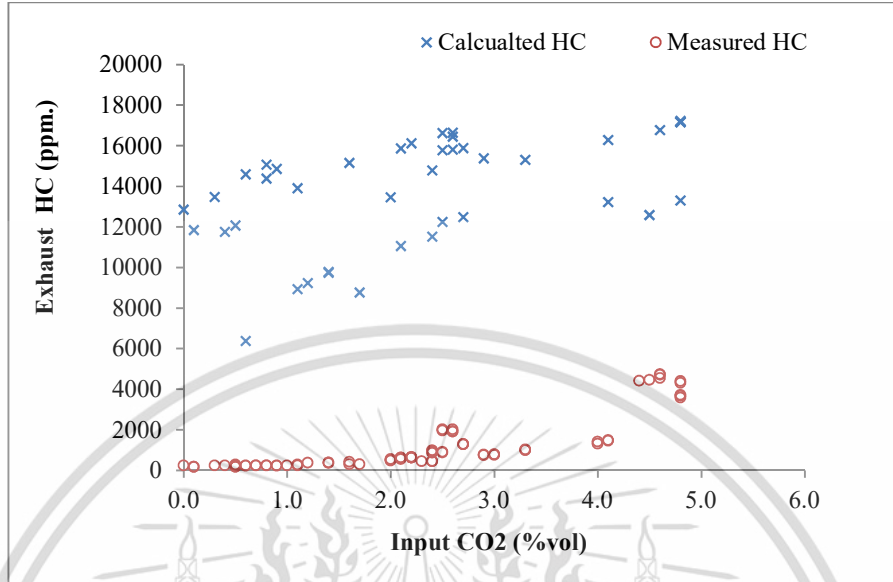


Figure 4.6-(c) Exhaust HC corresponding to Input CO<sub>2</sub>

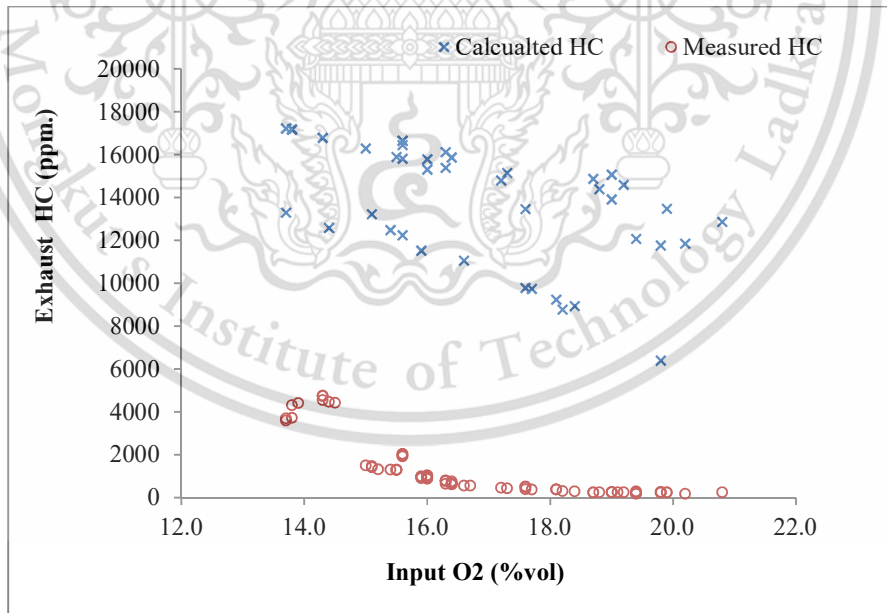
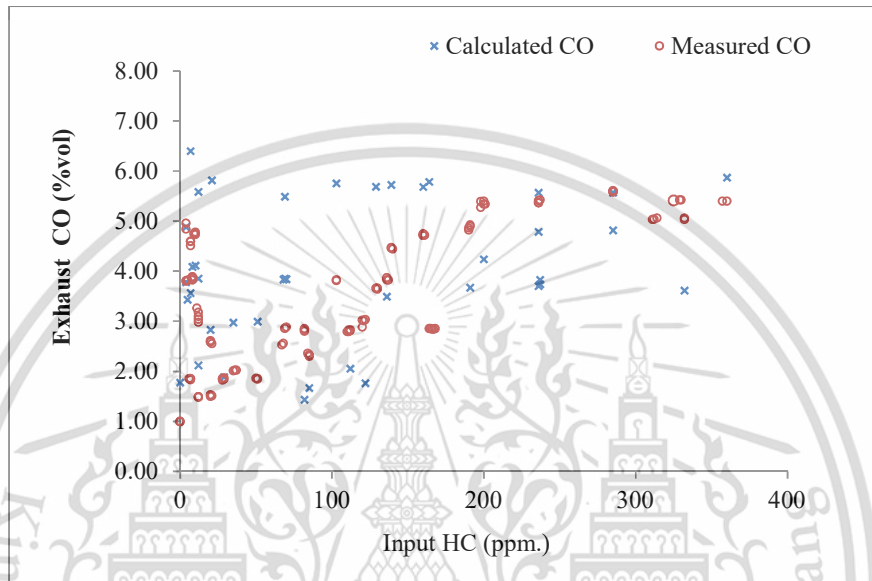
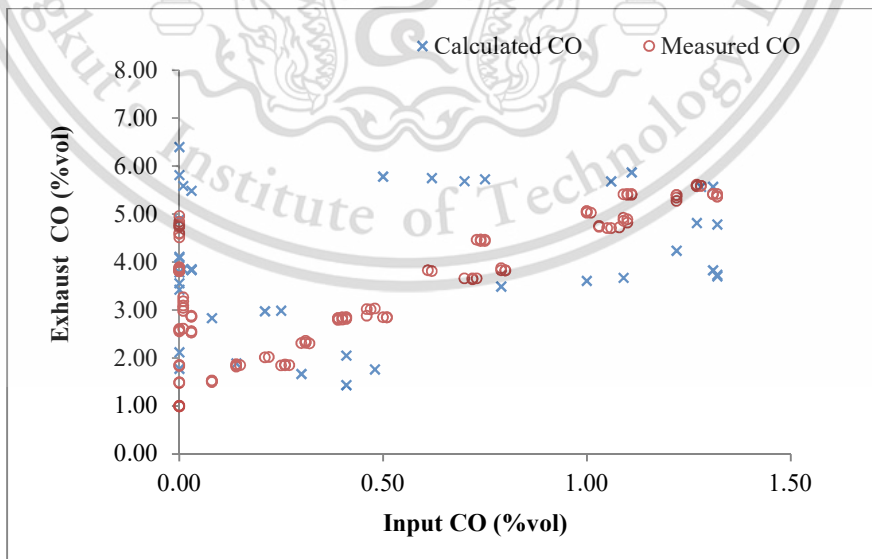


Figure 4.6-(d) Exhaust HC corresponding to Input O<sub>2</sub>

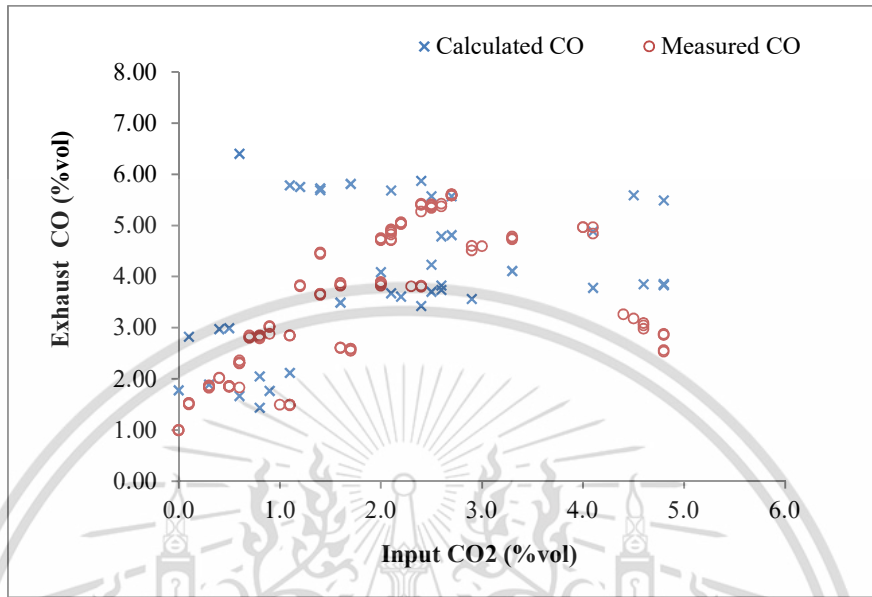
Figure 4.7-(a) to Figure 4.7-(d) demonstrates comparison of the variation of the exhaust CO to each intake polluted air element O<sub>2</sub>, CO<sub>2</sub>, CO, and HC, respectively, between the results measured from the experimentations and the values calculated from the mathematical model. Tendency of the calculated CO from the established model is consistency to the measured values in all the tested range.



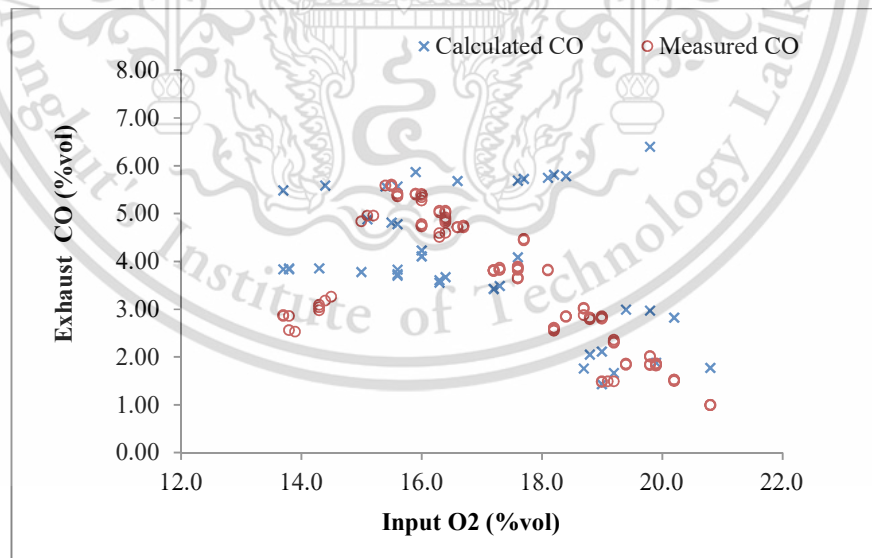
**Figure 4.7-(a)** Exhaust CO corresponding to Input HC



**Figure 4.7-(b)** Exhaust CO corresponding to Input CO

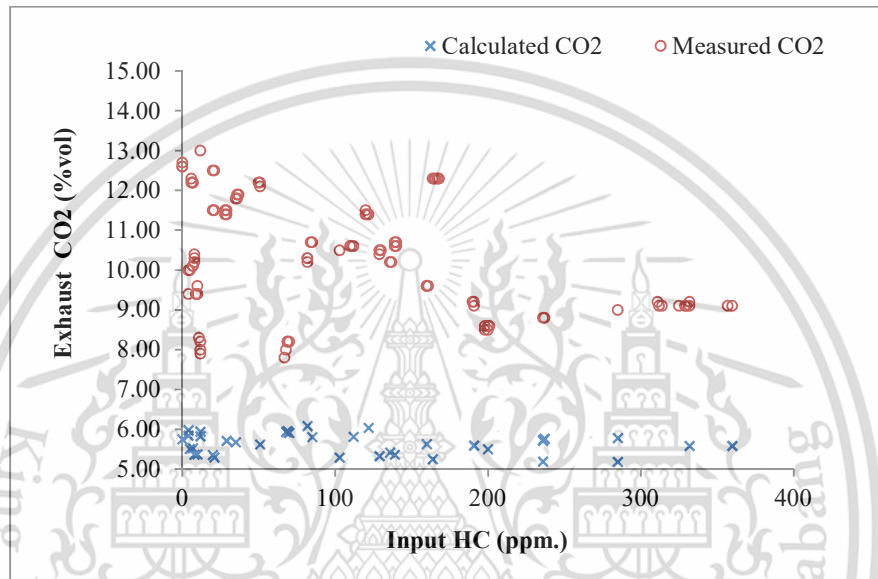


**Figure 4.7-(c)** Exhaust CO corresponding to Input CO<sub>2</sub>

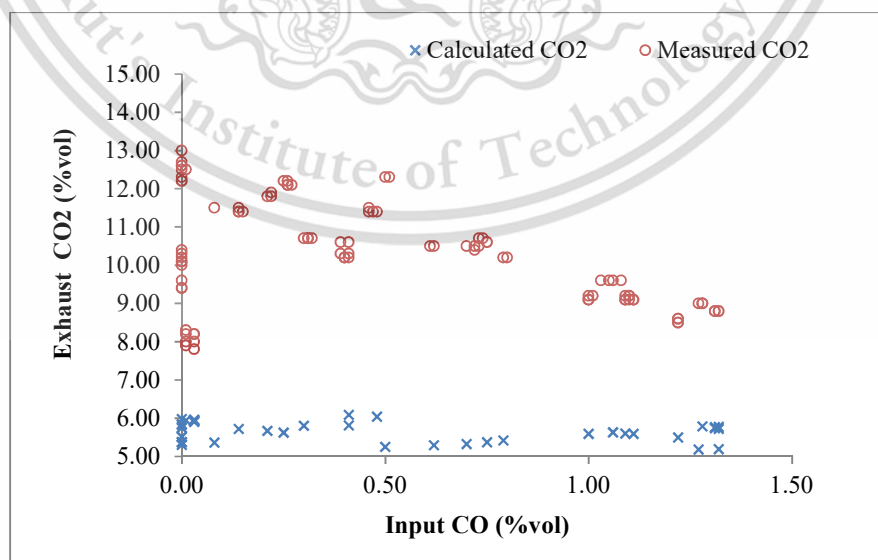


**Figure 4.7-(d)** Exhaust CO corresponding to Input O<sub>2</sub>

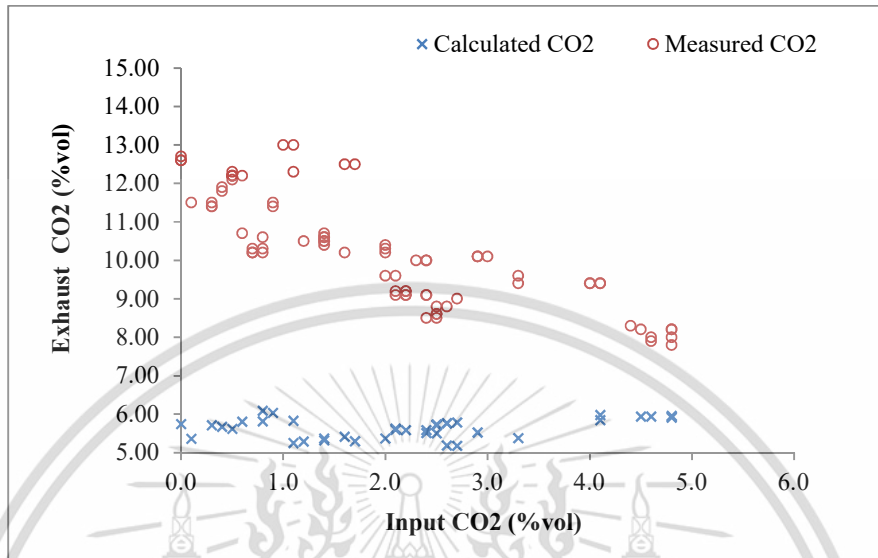
Figure 4.8-(a) to Figure 4.8-(d) demonstrates comparison of the variation of the exhaust CO<sub>2</sub> to each intake polluted air element O<sub>2</sub>, CO<sub>2</sub>, CO, and HC, respectively, between the results measured from the experimentations and the values calculated from the mathematical model. The estimated CO<sub>2</sub> look invariably and are lower than the measured CO<sub>2</sub> in all the tested range. That means the established model predicts lower completeness of the combustion than actual combustion without consider to the chemical effect of the dilution in the actual reaction.



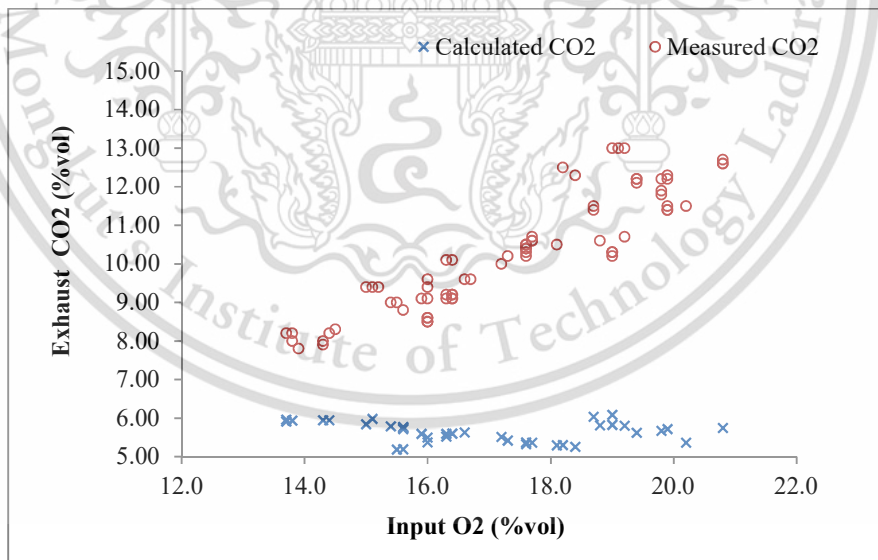
**Figure 4.8-(a)** Exhaust CO<sub>2</sub> corresponding to Input HC



**Figure 4.8-(b)** Exhaust CO<sub>2</sub> corresponding to Input CO



**Figure 4.8-(c)** Exhaust CO<sub>2</sub> corresponding to Input CO<sub>2</sub>



**Figure 4.8-(d)** Exhaust CO<sub>2</sub> corresponding to Input O<sub>2</sub>

Figure 4.9-(a) to Figure 4.9-(d) demonstrates comparison of the variation of the exhaust O<sub>2</sub> to each intake polluted air element O<sub>2</sub>, CO<sub>2</sub>, CO, and HC, respectively, between the results measured from the experimentations and the values calculated from the mathematical model. Trend of the estimated O<sub>2</sub> provides poor correlation to the measured O<sub>2</sub>. The values of estimated O<sub>2</sub> are lower than the measured O<sub>2</sub> in all the tested range.

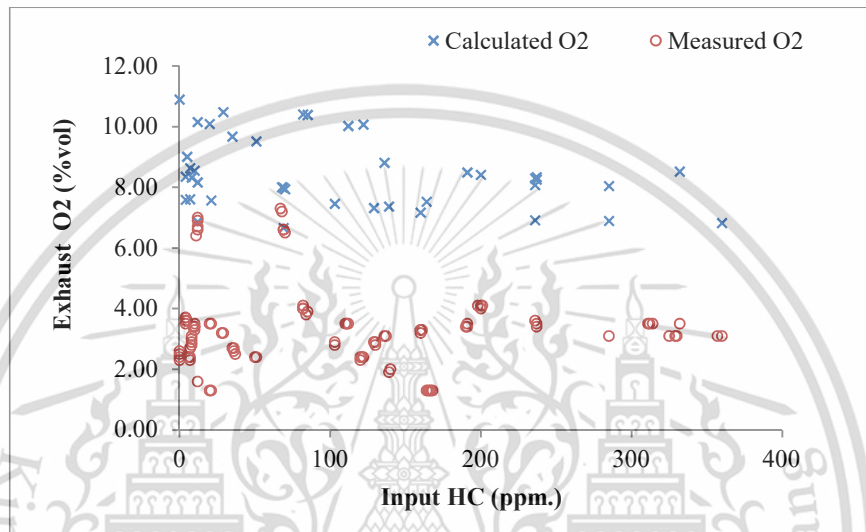


Figure 4.9-(a) Exhaust O<sub>2</sub> corresponding to Input HC

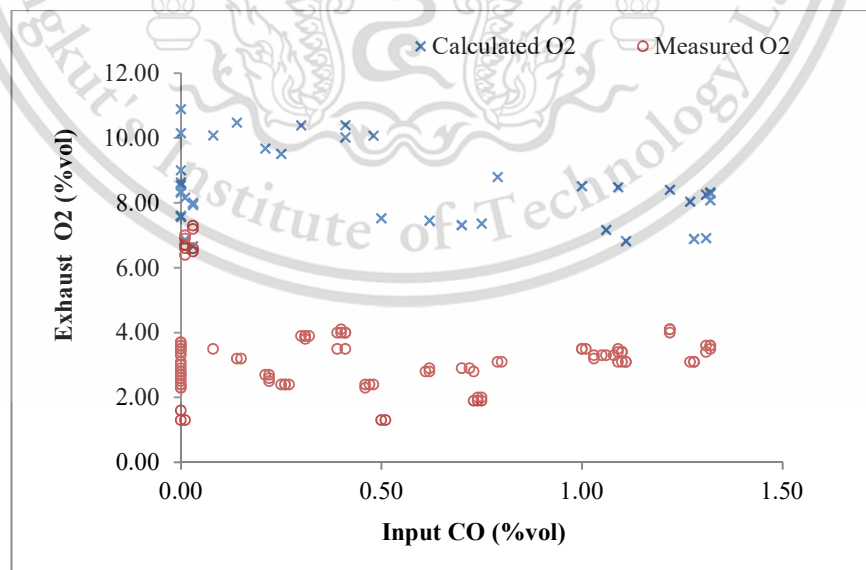
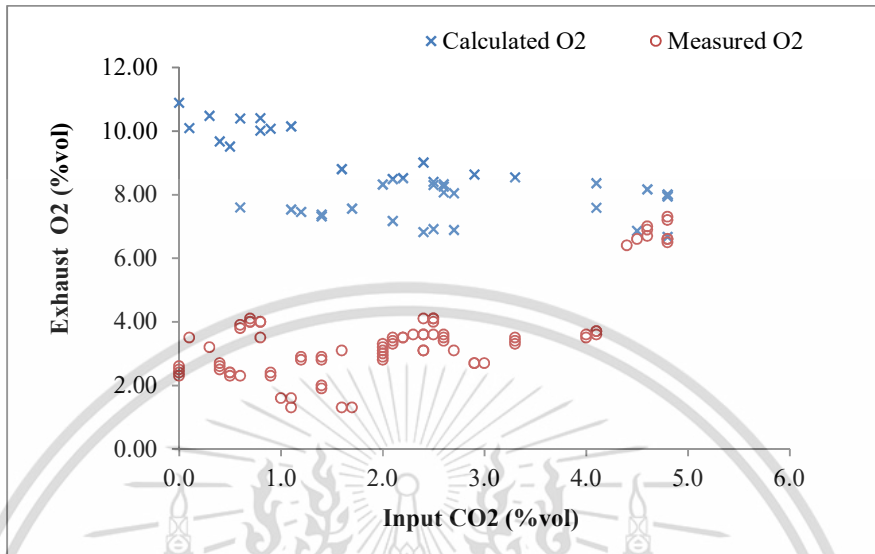
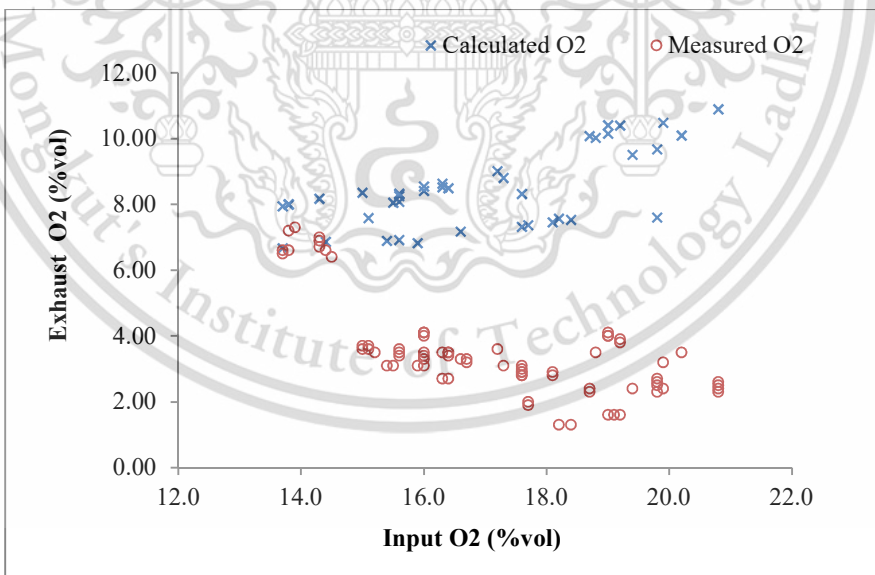


Figure 4.9-(b) Exhaust O<sub>2</sub> corresponding to Input CO

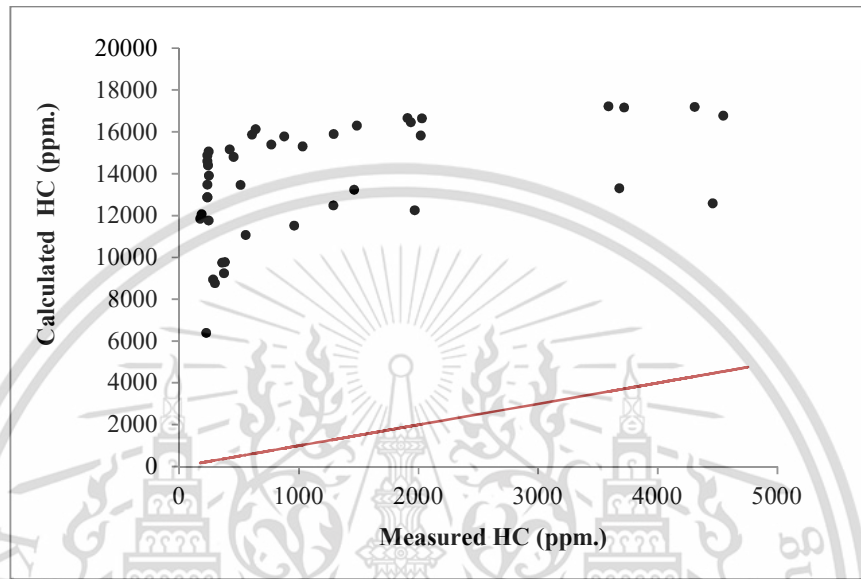


**Figure 4.9-(c)** Exhaust O<sub>2</sub> corresponding to Input CO<sub>2</sub>

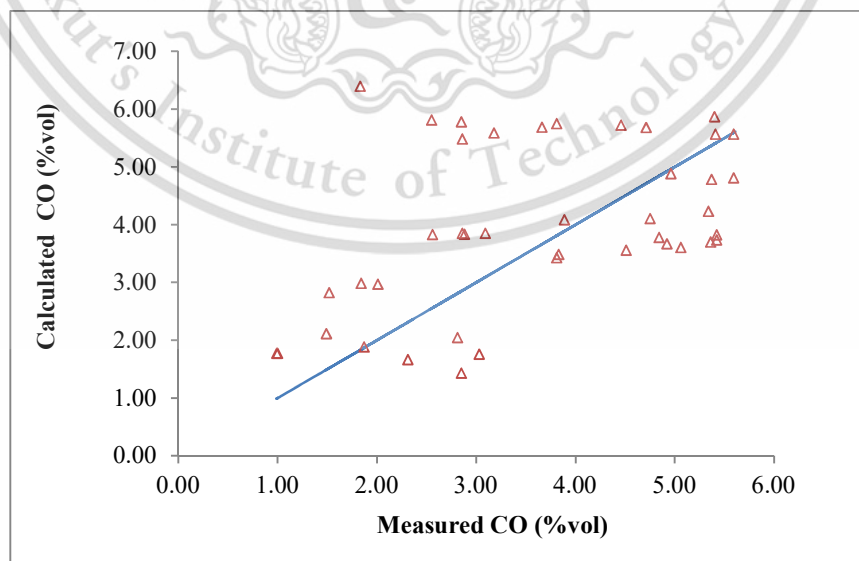


**Figure 4.9-(d)** Exhaust O<sub>2</sub> corresponding to Input O<sub>2</sub>

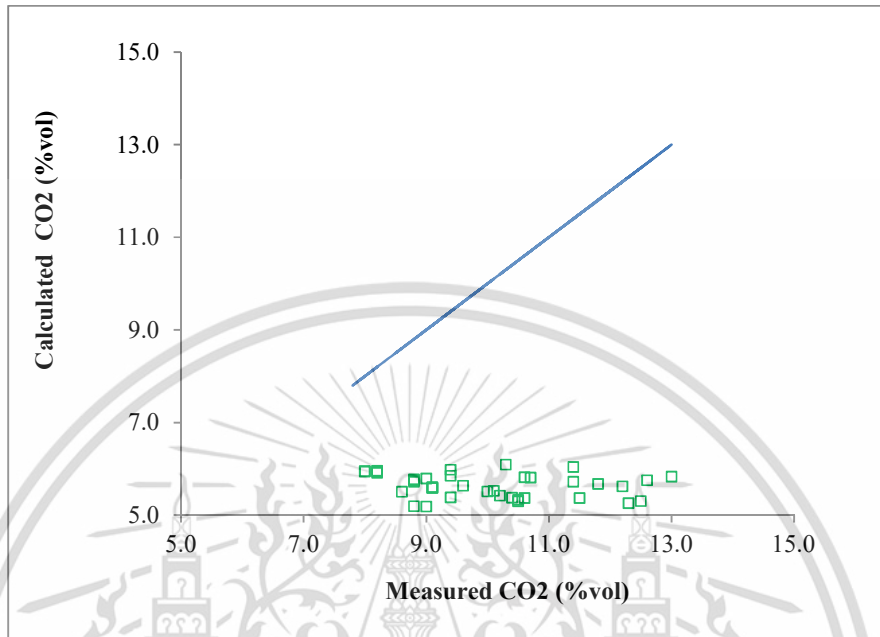
Figure 4.10-(a) to Figure 4.10-(d) demonstrates comparison between the estimations and the actual values of each exhaust gases. Only case of the exhaust HC and the exhaust CO which have consistent results while case of the exhaust CO<sub>2</sub> and the exhaust O<sub>2</sub> show poor validation of the model.



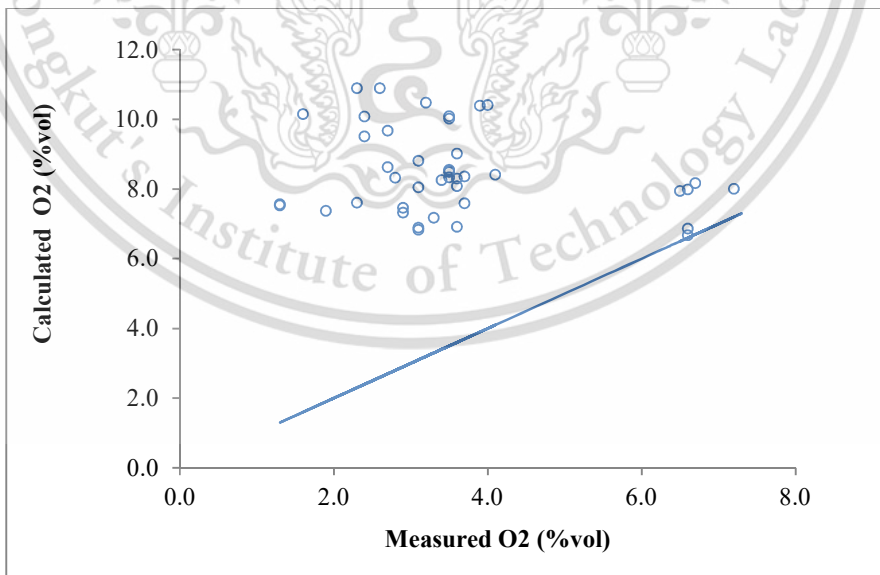
**Figure 4.10-(a)** Validation of Exhaust HC between Measurements and Calculations



**Figure 4.10-(b)** Validation of Exhaust CO between Measurements and Calculations



**Figure 4.10-(c)** Validation of Exhaust CO<sub>2</sub> between Measurements and Calculations



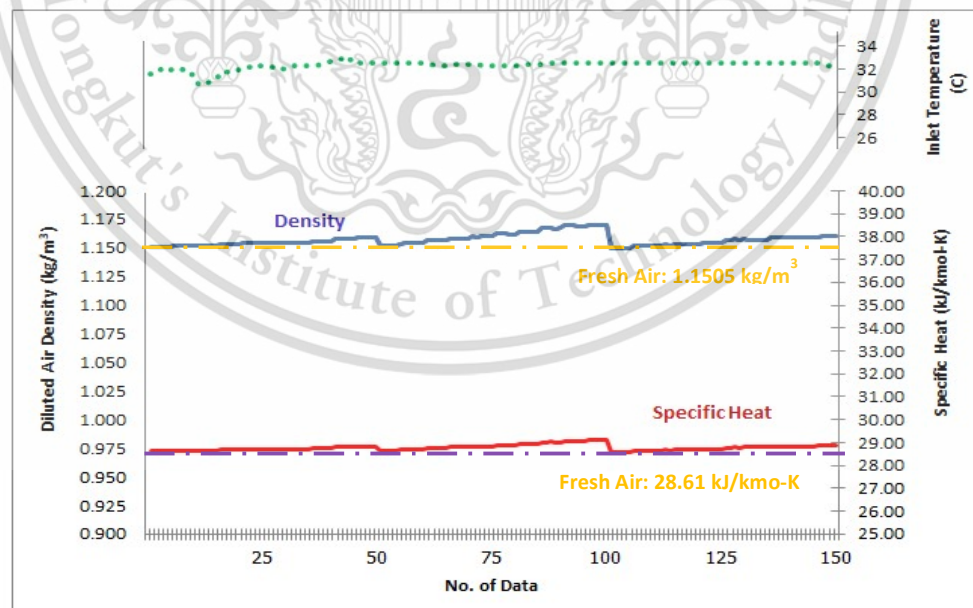
**Figure 4.10-(d)** Validation of Exhaust O<sub>2</sub> between Measurements and Calculations

Substantial different of the exhaust HC demonstrated in Figure 4.10-(a) should be pointed out that the calculated HC values from the model are referred as total hydrocarbons (THC) while the measured HC values in this experimentation obtained by Non-Dispersive Infrared (NDIR) detector are referred as n-hexane ( $n\text{-C}_6\text{H}_{14}$ ) equivalent values. HC in the engine emission consists of many hydrocarbon compounds in both gaseous and particle forms such as methane ( $\text{CH}_4$ ) and non-methane hydrocarbons (NMHC), not only n-hexane, while the amount of THC could be measured by Flame Ionization Detection (FID). HC concentrations by NDIR detector do not exactly agree with THC concentrations obtained by the FID detector because the readings of such two detectors are based on completely different principles and definitions (Adachi, 2014), resulting to substantial different as shown in the figure.

Due to the established model only relates the elemental composition of the reactant and product species and the addition 3 equations, e.g., equation (3.9)-(3.11), employed the correlation of the exhaust gas; it does not indicate the process predicts by which combustion proceeds which is much more complex. So, the mathematical model for estimation of the exhaust gas constituent should be improved by considering of the chemical effect, not only considering of the principle of mass conservation.

#### 4.1.3 Calculations of the Density and the Specific Heat of the Air Intake

Figure 4.11 shown the measured data of the inlet air temperature and the variation of the mean density and the specific heat of the diluted air at the intake port.

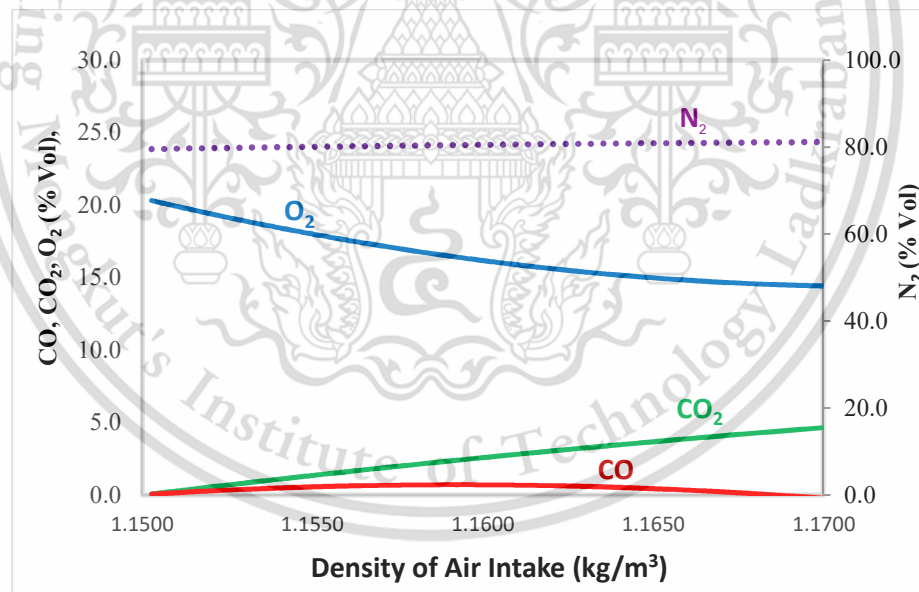


**Figure 4.11** Inlet Air Temperature, Mean Density, and Specific Heat of the Diluted Air Intake

Variation of the mean density of the diluted air intake is between 1.1502 to 1.1704 kg/m<sup>3</sup>, while the maximum difference to the normal fresh air is about 1.72%. Variation range of the specific heat of the diluted air intake is between 28.6 to 29.09 kJ/kmol-K, while the maximum difference to the normal fresh air is only 1.68%.

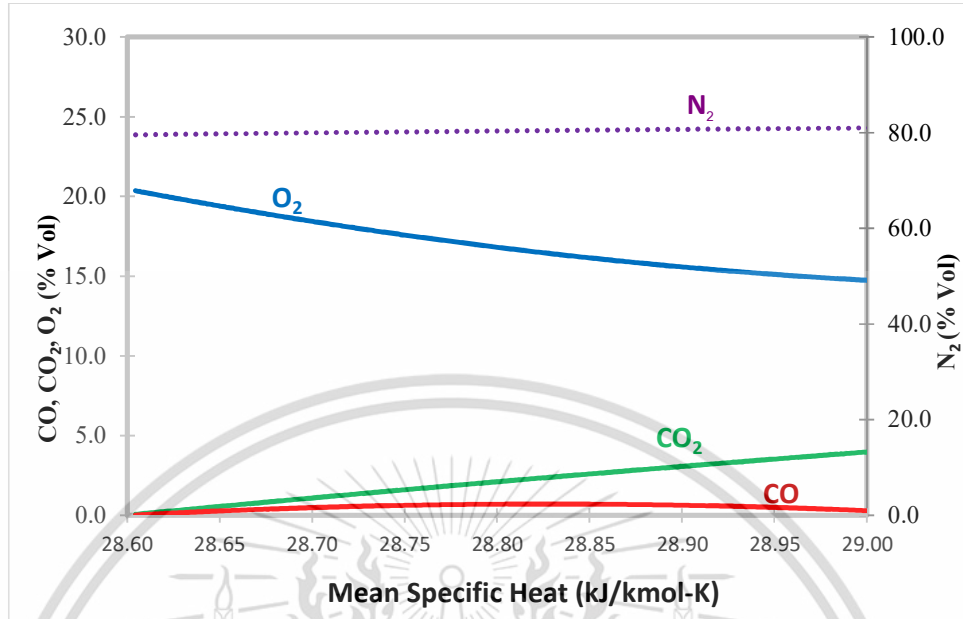
Figure 4.12 shown the effects of the concentration of the intake gas's composite, CO, CO<sub>2</sub>, O<sub>2</sub>, and N<sub>2</sub>, to the variation of the mean density of the diluted air at the intake port. It could be found that only the concentration of the O<sub>2</sub> and CO<sub>2</sub> elements which had correlation to the mean density of the diluted air intake. The variation of the diluted air composition mostly derived from the change in the proportional of these two gases, while the N<sub>2</sub> concentrations were rather constant. Increase of the CO<sub>2</sub> concentration to the air intake results in increase of the mean density of the air intake due to higher density value of the CO<sub>2</sub> gas comparing to other gases.

The variation of the density of the diluted air intake is different less than 1.72% to the density of normal air of 1.505 kg/m<sup>3</sup> at the average inlet temperature. So, the density of the diluted air intake does not alter significantly with the variation of composition of the diluted air intake, hence, the engine volumetric efficiency ( $\eta_v$ ) is not affected by the variation of the diluted air intake's composition.



**Figure 4.12** Inlet Air Temperature and Effects of Pollution Gas to Density of Air Intake

Figure 4.13 shown the effects of the concentration of the intake gas's composite, CO, CO<sub>2</sub>, O<sub>2</sub>, and N<sub>2</sub>, to the variation of the specific heat at the intake port. It could be found that only the concentration of the O<sub>2</sub> and CO<sub>2</sub> elements which affect to the total mean specific heat of the diluted air intake.

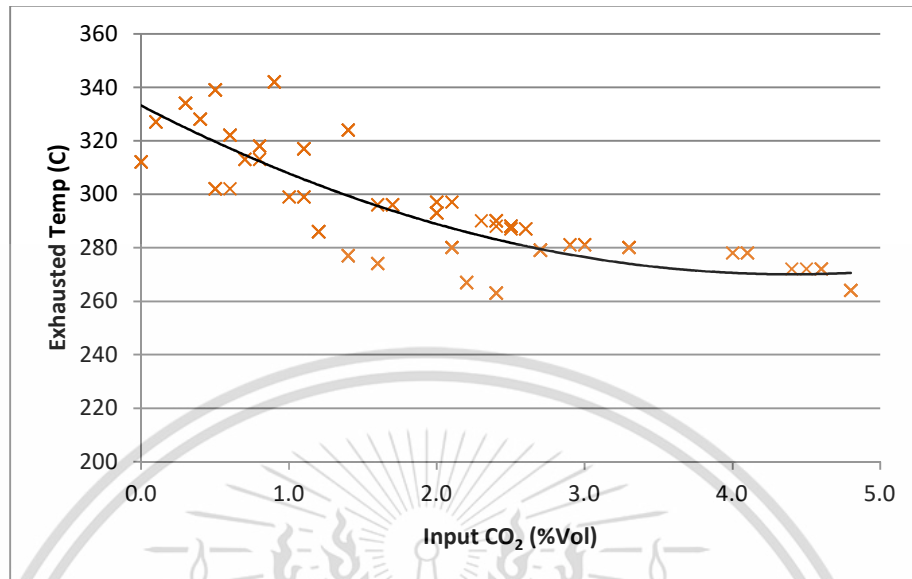


**Figure 4.13** Effects of Pollution Gas to Mean Specific Heat of the Diluted Air Intake

It can be described that variation of the diluted air composition derived from the change in the concentration of these two gases, while the N<sub>2</sub> concentrations are relatively constant. Higher the concentration of the CO<sub>2</sub> element causes lowering the total mean specific heat due to lower value of the specific heat at the ambient temperature (about 305 K) of the CO<sub>2</sub> gas. Such conditions are occurred when the concentration of the O<sub>2</sub> element decrease.

Although the specific heat of CO<sub>2</sub> at the ambient temperature has lower value than the value of O<sub>2</sub>, but the value of CO<sub>2</sub> which is the triatomic gas has more strongly increase with the combustion temperature than O<sub>2</sub> which is the diatomic gas. Since the CO<sub>2</sub> acts as an inert gas which does not participate in the combustion reaction, but it will absorb the combustion energy like the heat sink, resulting to decrease of the combustion temperature (Pundir, 2013) (Adachi, 2014). Such phenomenon could be observed from the measurement of the exhaust gas temperature which are shown in Figure 4.14.

The temperature of the exhaust gas depends upon several factors, the important factors of which are the nature of the fuel, the mixture strength, valve timing, and compression ratio. The design of the combustion chamber, and of the exhaust valve and its port, and the silencing arrangement, also have an appreciate influence. Many experimental results shown that the maximum exhaust gas temperature is achieved when the air-fuel mixture ratio is corresponding to the stoichiometric value and are decreasing when the mixture is weak or rich (Judge, 1943).



**Figure 4.14** Effects of CO<sub>2</sub> Concentration to the Exhausted Temperature

In Figure 4.14, increase the amount of CO<sub>2</sub> into the fresh air intake up to about 5.0 %vol caused the decrease of the exhaust gas temperature about 15.4%. This can be attributed to the oxygen deficient condition which results in lower combustion temperature and furthermore the specific heat of the intake CO<sub>2</sub> is higher than that of the fresh air intake which contributes to the lower combustion temperatures (Meshack, 2015).

#### 4.1.4 Effects of Intake Pollution to the Exhaust Hydrocarbon

Unburned hydrocarbon (HC) emissions arise as part of the fuel inducted into the engine is not burned and escapes the combustion process in the engines, while portions of the fuel present in the exhaust gases as C<sub>1</sub> to C<sub>8</sub> hydrocarbons usually is used as an indication of the combustion efficiency of the engine (Martin, 2014). Several processes contribute to the HC emissions in the SI engines, depending upon the engine design and operating conditions. Main sources of HC emission in the typical SI engine are (Pundir, 2013):

- (i) Flame quenching on the cylinder walls
- (ii) Flame quenching in the crevices
- (iii) Absorption and desorption in oil film on the cylinder walls
- (iv) Carbon deposits in the chamber
- (v) Misfired combustion or bulk gas quenching
- (vi) Liquid fuel in the cylinder
- (vii) Exhaust valve leakage
- (viii) Crankcase blow by gases

Figure 4.15-(a) to 4.15-(d) show correlation of the diluted air intake and the measured data of the exhaust HC concentration derived from analysis of the dry exhaust gas composition according to each intake elements. Typically, a four-stroke SI engine without exhaust aftertreatment emits 300 to 6,000 (C<sub>1</sub>) ppm HC while under idle and transient conditions the values may be higher (Pundir, 2013).

Amount of the exhaust HC are unburned fuel components which occur in the exhaust emission after incomplete combustion. Figure 4.15-(a) to 4.15-(d) demonstrate the amount of the exhaust HC to the variation of the concentration of the intake HC, CO, CO<sub>2</sub>, and O<sub>2</sub> respectively. Variation of the exhaust HC to the intake HC and the intake CO do not show evident correlations but decrease of the intake O<sub>2</sub> or increase of the intake CO<sub>2</sub> causes rise of the exhaust HC obviously until to certain values 14 %vol of O<sub>2</sub> and 5 %vol of CO<sub>2</sub>, the tested engine is instability due to misfiring.

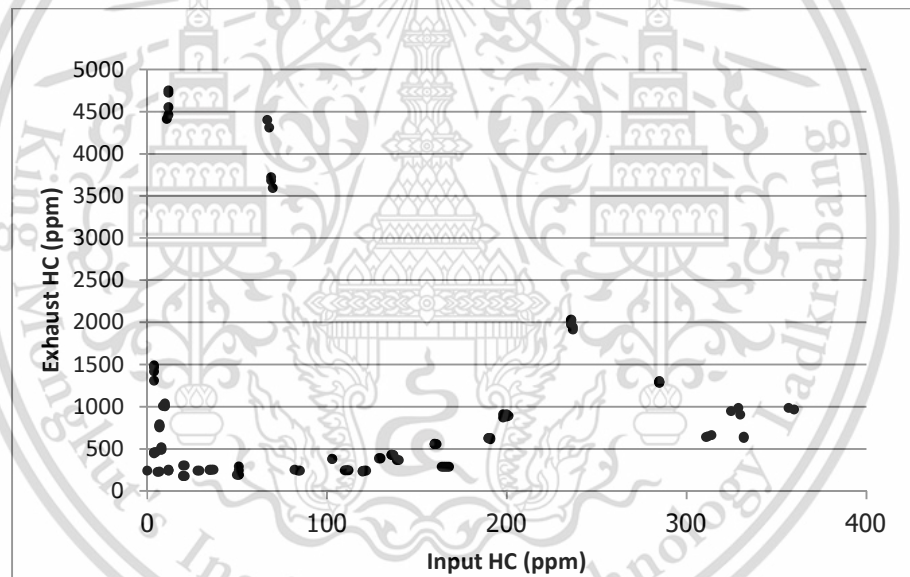


Figure 4.15-(a) Variation of Exhaust HC corresponding to Input HC

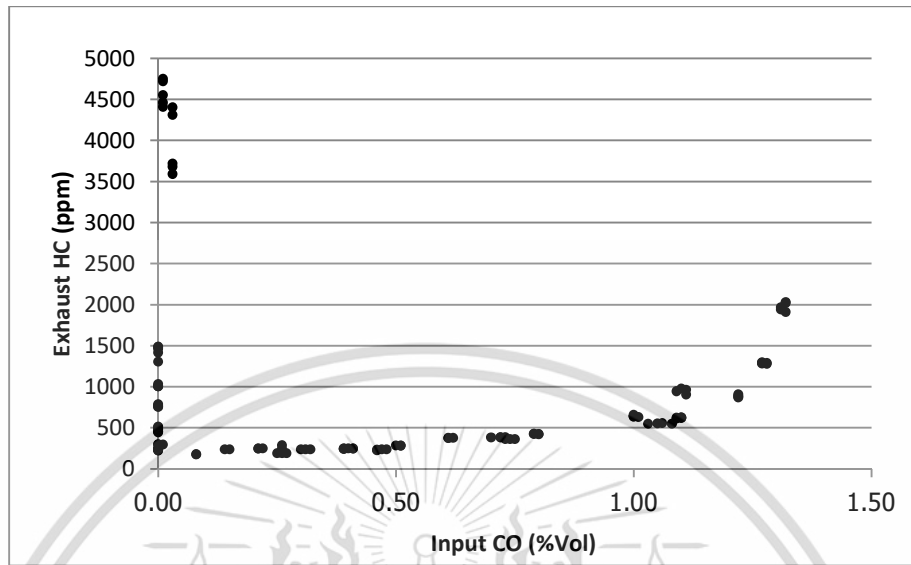
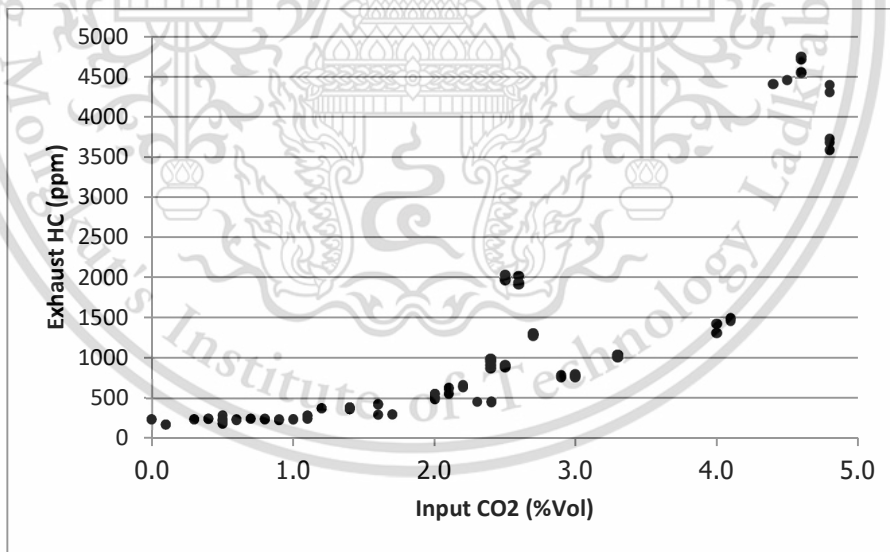
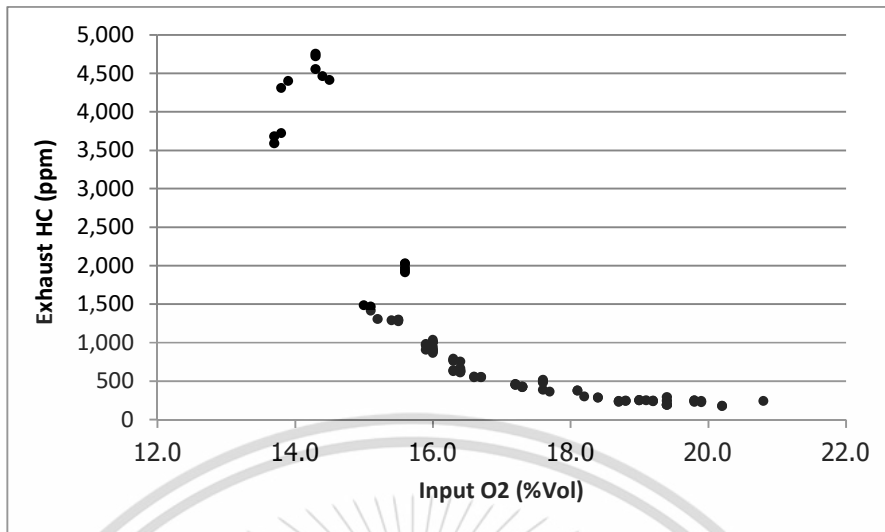


Figure 4.15-(b) Variation of Exhaust HC corresponding to Input CO





**Figure 4.15-(d)** Variation of Exhaust HC corresponding to Input O<sub>2</sub>

Such results can be described that increase of CO<sub>2</sub> portion into the air intake results to decrease of available O<sub>2</sub> intake for combustion as mentioned in section 4.1.2. As the intake CO<sub>2</sub> increases over a certain value, slow combustion speeds lead to partial burning or misfire that produces unburned HC (Jinyoung, 2001). High dilution of CO<sub>2</sub> intake causes the combustion process to be retarded resulting in burning of more fuel during expansion stroke. While the cylinder pressure falls during expansion stroke, the temperature of the unburned mixture ahead of the flame decreases resulting in extinction of the flame before it burns the entire charge and consequently in partially misfired combustion leading to high HC emissions.

#### 4.1.5 Effects of Intake Pollution to the Exhaust Carbon Monoxide

The CO emissions result from incomplete combustion and largely dependent on the air-fuel ratio. The air-fuel ratio is the most important engine parameter affecting the CO emissions. The exhaust CO increases substantially as the air-fuel ratio decreases below the stoichiometric value, the mixture become richer than stoichiometric. In an abundance of O<sub>2</sub>, the CO element would react to the O<sub>2</sub> to form CO<sub>2</sub>. Hence when there is an excess of O<sub>2</sub>, there is relatively little CO emissions. As there is a deficient O<sub>2</sub> so that there is less O<sub>2</sub> available, the amount of CO emissions increase. The same result could happen with the poor mixing of the induced charge. Mixture distribution between cylinders is also factor influences the CO emission, but not directly, poor distribution of fuel-air mixture causes cylinder-to-cylinder variation in the air-fuel ratio which conducts to significant increase in average CO emissions. Partial oxidation of hydrocarbons from the crevices that are entrained into burned gases during expansion is another factor influences to the CO emissions, but this last factor has significant

contribution to the CO emissions at lean mixture conditions. So that the exhaust CO concentration for lean mixture is usually less than 0.1 %vol.

Figure 4.16-(a) to 4.16-(d) show correlation of the diluted air intake and the measured data of the exhaust CO concentration derived from analysis of the dry exhaust gas composition according to each intake elements. Typically, a four-stroke SI engine without exhaust aftertreatment emits 0.2 to 5.0%vol CO while under idle condition the values may be increased due to fuel-rich operation (Pundir, 2013).

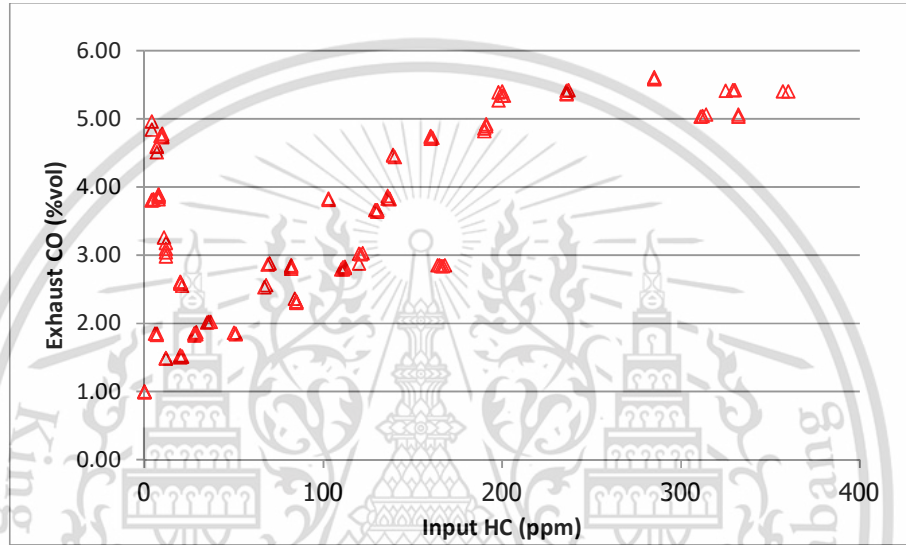


Figure 4.16-(a) Variation of Exhaust CO corresponding to Input HC

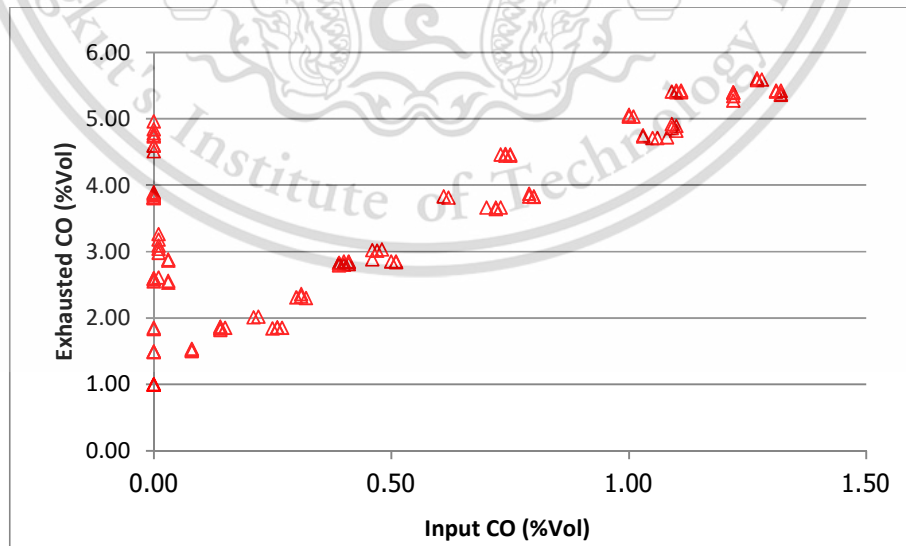
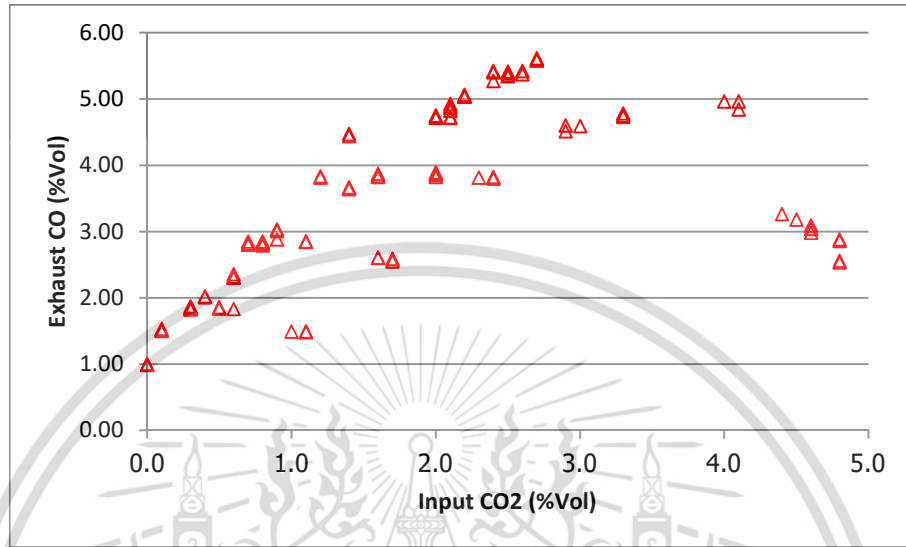
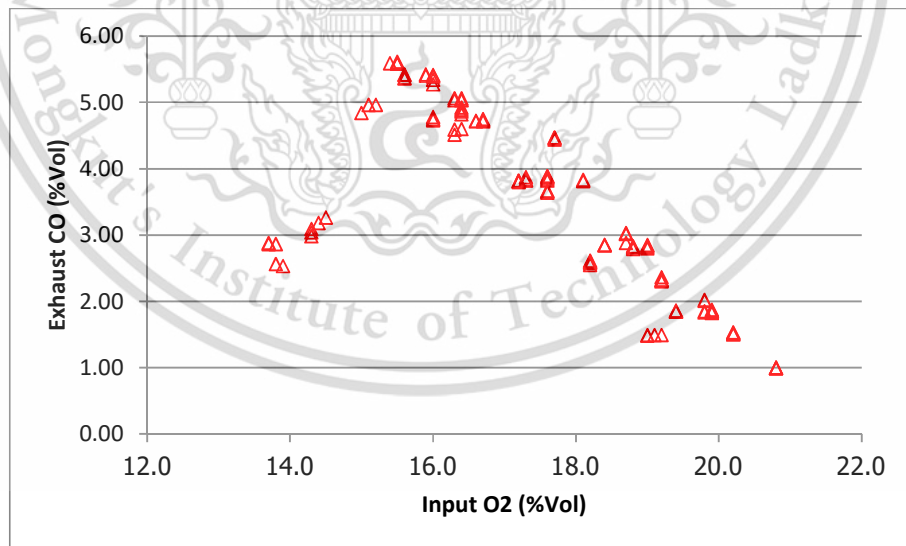


Figure 4.16-(b) Variation of Exhaust CO corresponding to Input CO



**Figure 4.16-(c)** Variation of Exhaust CO corresponding to Input CO<sub>2</sub>



**Figure 4.16-(d)** Variation of Exhaust CO corresponding to Input O<sub>2</sub>

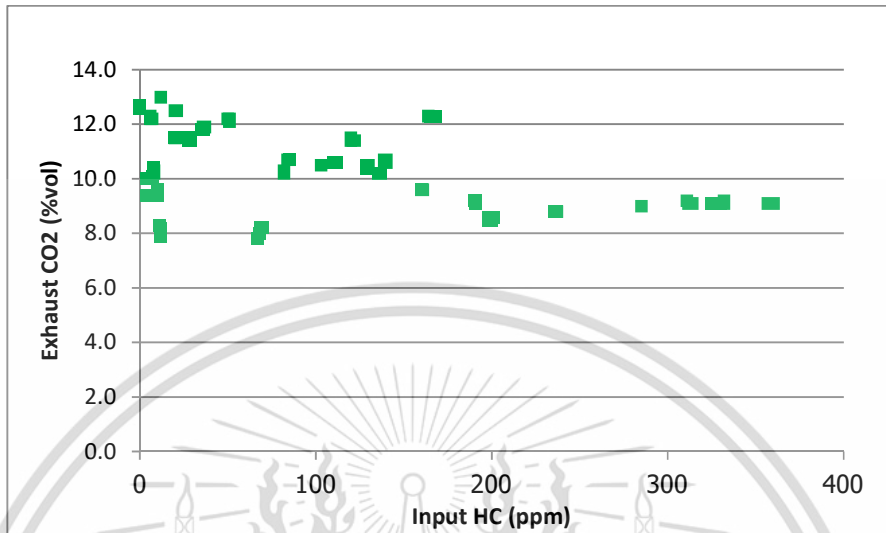
From figure 4.16-(a) to 4.16-(d), increase of all the intake pollutants CO<sub>2</sub>, CO and HC result to rise of the exhaust CO proportionally which means incomplete combustion of the engine. The maximum value of the exhaust CO is about 5.56 %vol which is higher than the exhaust CO from the combustion of fresh air, as mentioned in Table 4.1, about 5.6 times.

Such results are consistent to decrease of the intake O<sub>2</sub> which causes the richer air-fuel mixture, hence, there is not enough oxygen to oxidize all the carbon molecules sufficiently into CO<sub>2</sub>. The instability of engine is also shown in Figure 4.16-(a) and 4.16-(b) as the concentrations of the exhaust CO are dropped while the concentrations of the intake O<sub>2</sub> and CO<sub>2</sub> are lower to 14 %vol and higher than 5 %vol, respectively.

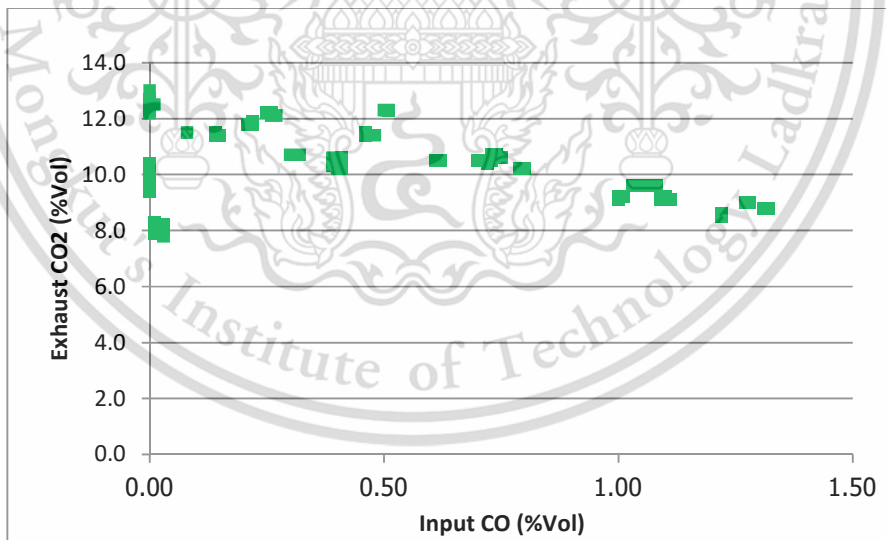
#### **4.1.6 Effects of Intake Pollution to the Exhaust Carbon Dioxide**

As the rise of the exhaust HC and CO means incomplete combustion, amount of the exhaust CO<sub>2</sub> implies to completeness of the combustion. A two-step processes may be employed to describe complete combustion of hydrocarbon fuels to form CO<sub>2</sub> as final product. Initial process, which in actual consists of several intermediate oxidation reactions, is conversion of hydrocarbon to CO. The second process is conversion of CO to CO<sub>2</sub> if sufficient O<sub>2</sub> is available. Under ideal conditions, combustion process continue at best until chemical equilibrium. At all air-fuel ratios, the combustion process can be incomplete due to imperfect mixing or because certain reactions proceed so slowly that chemical equilibrium is not reached. Besides to the previous mentioned reasons, reduction of the exhaust CO<sub>2</sub> is only to be achieved through a reduction in fuel consumption or through an altered fuel having a small amount of carbon with reference to its heating value (Merker, 2009).

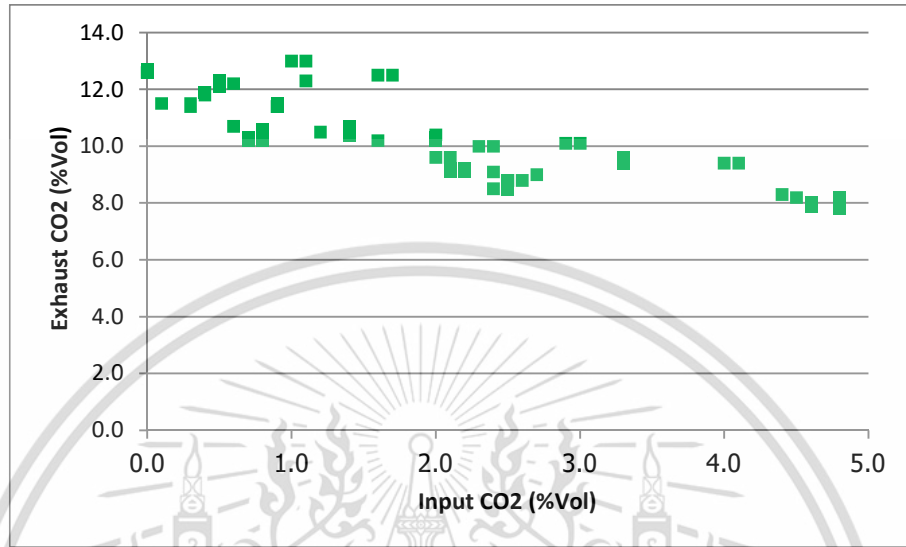
Figure 4.17-(a) to 4.17-(d) show correlation of the diluted air intake and the measured data of the exhaust CO<sub>2</sub> concentration derived from analysis of the dry exhaust gas composition according to each intake elements. For SI engine which has efficient combustion, the CO<sub>2</sub> emissions without exhaust aftertreatment would be near to 15.5 %vol approximately (<http://www.technicar.com/exhaust-analyzer/>).



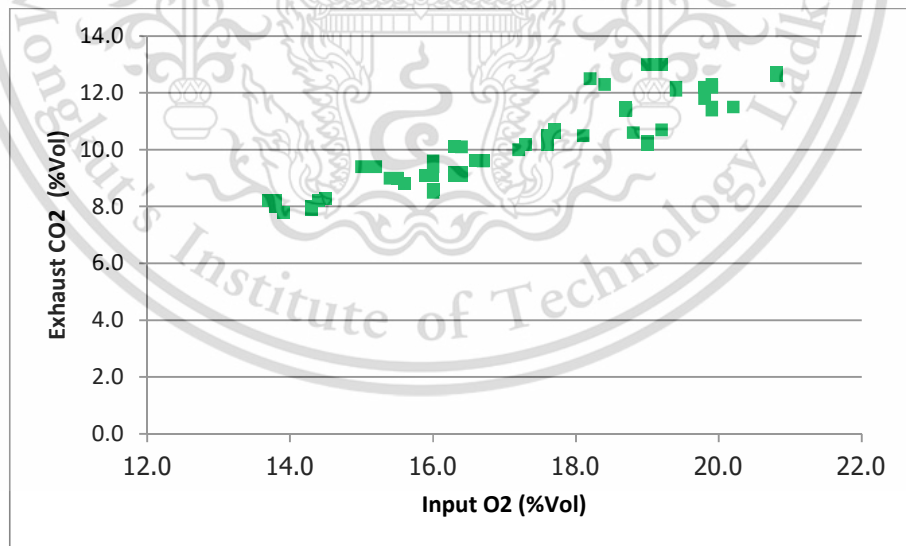
**Figure 4.17-(a)** Variation of Exhaust CO<sub>2</sub> corresponding to Input HC



**Figure 4.17-(b)** Variation of Exhaust CO<sub>2</sub> corresponding to Input CO



**Figure 4.17-(c)** Variation of Exhaust CO<sub>2</sub> corresponding to Input CO<sub>2</sub>



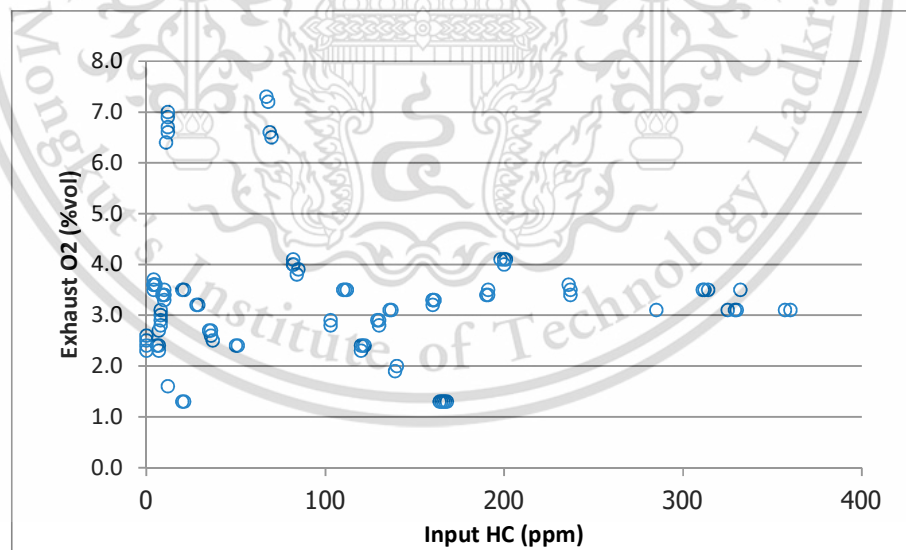
**Figure 4.17-(d)** Variation of Exhaust CO<sub>2</sub> corresponding to Input O<sub>2</sub>

According to Figure 4.17-(a) to 4.17-(d), the concentration of the exhaust CO<sub>2</sub> is lower than the value from the combustion of the fresh air intake, 12.6-12.7 %vol as mentioned in Table 4.1, and tends to decrease continuously as the concentrations of the intake CO<sub>2</sub>, CO, and HC are increased, while such variation is inversely proportion to the increase of the intake O<sub>2</sub>. The minimum concentration could be decreased to 7.8 %vol which is only half amount of the maximum value available from the stoichiometric combustion. Such results are due to incompleteness of combustion directly, while other reasons affecting to the reduction of the exhaust CO<sub>2</sub> are not relevant.

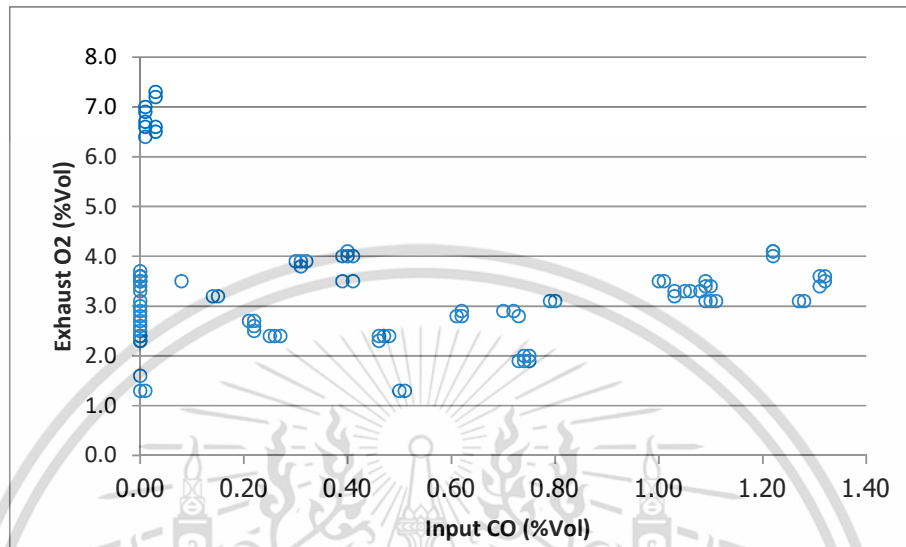
#### 4.1.7 Effects of Intake Pollution to the Exhaust Oxygen

Practically, mixing of the fuel and the air intake is not perfectly homogeneous. Although the fuel is burnt in an excess of air, some of the oxygen would be present as free oxygen in the exhaust. So that the exhaust composition usually is found to having both CO and O<sub>2</sub> in the combustion product.

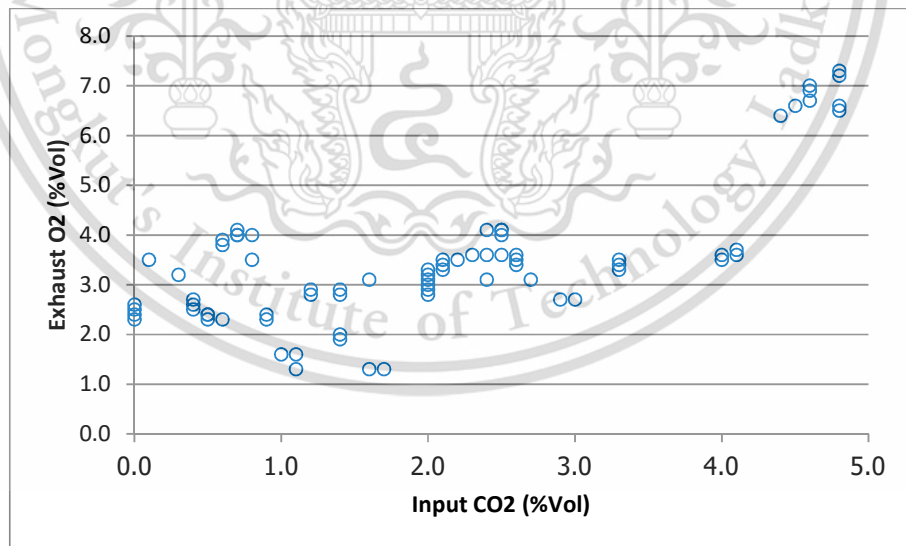
Figure 4.18-(a) to 4.18-(d) show correlation of the diluted air intake and the measured data of the exhaust O<sub>2</sub> concentration derived from analysis of the dry exhaust gas composition according to each intake elements.



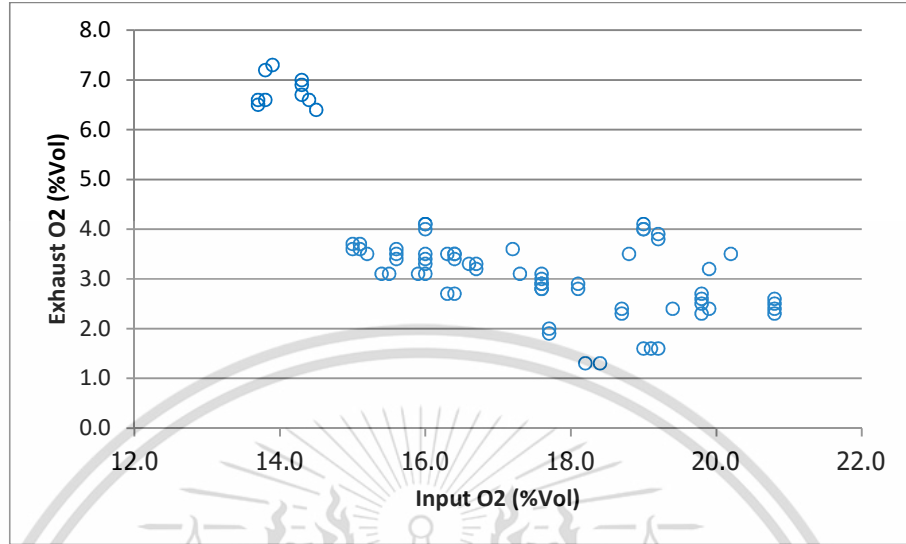
**Figure 4.18-(a)** Variation of Exhaust O<sub>2</sub> corresponding to Input HC



**Figure 4.18-(b)** Variation of Exhaust O<sub>2</sub> corresponding to Input CO



**Figure 4.18-(c)** Variation of Exhaust O<sub>2</sub> corresponding to Input CO<sub>2</sub>



**Figure 4.18-(d)** Variation of Exhaust O<sub>2</sub> corresponding to Input O<sub>2</sub>

Although there were no exact correlations between the exhaust and O<sub>2</sub> and the proportional composition of the diluted air intake, but the tested results demonstrate that amount of the exhaust O<sub>2</sub> mostly seem to be higher than the combustion of the fresh air. In combustion of the fresh air intake as mentioned in Table 4.1, range of the exhaust free oxygen was about 2.3-2.6 %vol, comparing to the combustion of the diluted air intake, average amount of the exhaust free oxygen was about 3.2 %vol while the maximum value is about 4.1 %vol.

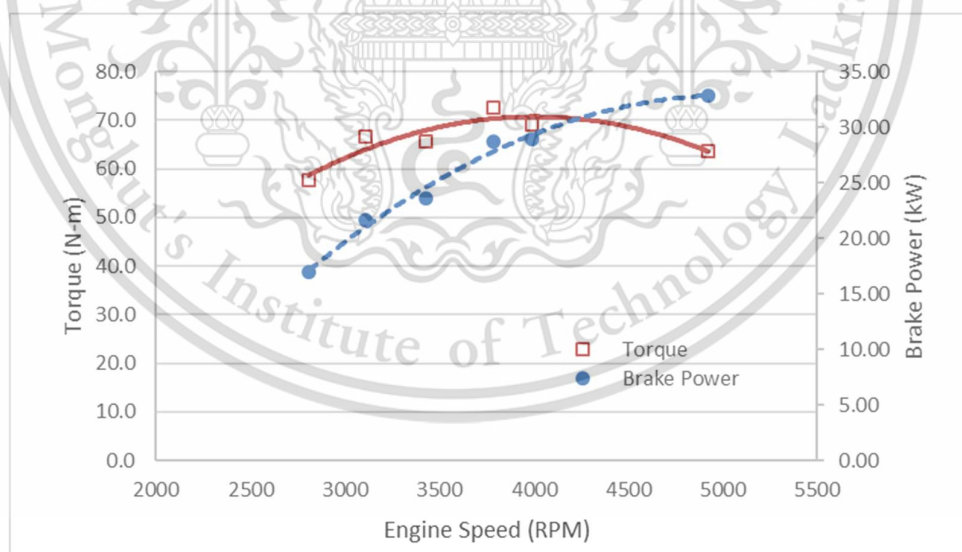
Results of the experimentations show that poor correlations are found between the excess O<sub>2</sub> and the varying concentrations of the intake elements. Non-homogeneous mixing of the fuel and the air is the main factor affecting to amount of the excess O<sub>2</sub> of which values are independent to chemical characteristics of the intake air. Also, Figure 4.18-(c) and 4.18-(d) show that dilution of CO<sub>2</sub> higher than 4.0 %vol causes occurrence of the misfire as more free oxygen are found in the exhaust component.

According to the experimental results, it was found that dilution of the CO<sub>2</sub> to the air intake has evident effect to the exhaust gas constituent. The dilution of CO<sub>2</sub> causes 3 major effect to the combustion process; 1) dilution effect due to displacement of CO<sub>2</sub> to O<sub>2</sub> in the fresh air intake resulted in the reduction in the O<sub>2</sub> supplied to the engine, 2) thermal effect due to increase of the thermal capacity of the inlet charge resulted in the reduction of the exhaust temperature, and 3) chemical effect resulted in incompleteness of the combustion process. The mathematical model for estimation of the exhaust gas constituent should be improved by considering of the chemical effect, not only considering of the principle of mass conservation.

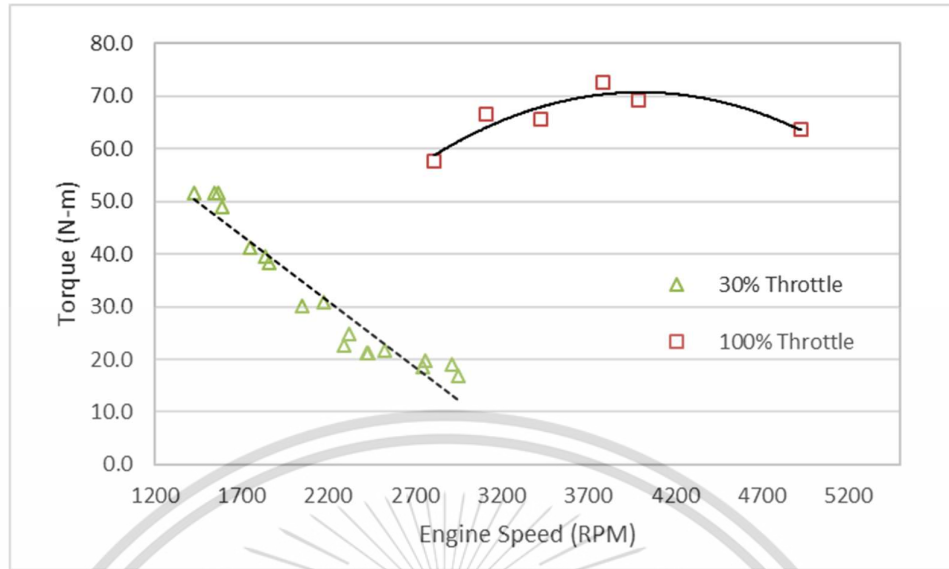
## 4.2 Results of Testing at the Partial Load Condition

The effects of the diluted air intake to the engine performance at partial load conditions had been studied experimentally by testing the engine operating at moderate operating conditions (1,200 to 3,000 rpm) with a fixed throttle setting position at 30%. From the previous experimentation, the effects of the diluted air intake were studied through varying the intake CO<sub>2</sub> concentration in the diluted air intake, and then the tested results of the fuel consumption, the output torque, the brake power, and the brake specific fuel consumption were measured with the engine speed and compared to the normal fresh air combustion test.

Figure 4.19 depicts characteristics of torque and brake power at full opened throttle position of the tested engine against the engine speed. Figure 4.20 illustrates effect of throttling the air intake to the engine torque by reducing the throttle position comparing between 100% opened throttle position and 30% opened position. Since the engine torque and the brake power have strong relationship to the volumetric efficiency. There should be little restriction to air flow in the inlet port and it is important to induce as great a mass flow rate of the air intake as possible into the engine cylinder to attain a high output torque and power. Figure 4.20 also shows satisfied results to prior studies (Greene and Lucas, 1969) that reducing the throttle position and so reducing the induction air flow rate into the engine, not only is the output torque reduced, but the maximum torque speed decrease too.

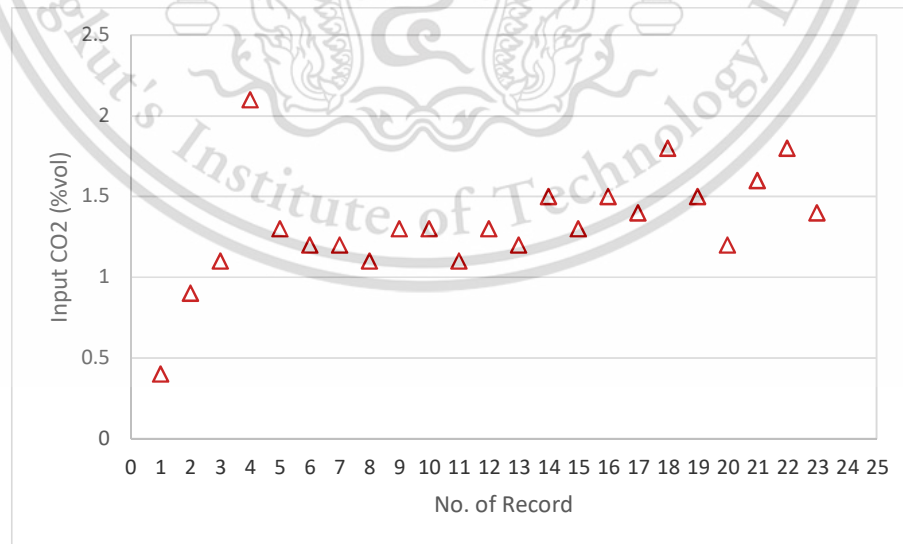


**Figure 4.19** Engine Torque and Brake Power at Fully-opened Throttle Position



**Figure 4.20** Engine Torque at 30% and 100% Throttle Position

Figure 4.21 illustrates attainable variation of the intake CO<sub>2</sub> concentration from the testing. Although the testing speed of the tested engine increase, but the maximum of the intake CO<sub>2</sub> concentration could be obtained was about 1.8 %vol. The reason why the maximum value of the intake CO<sub>2</sub> concentration could not be raised to higher value may be due to the intake CO<sub>2</sub> for testing derived from the exhaust pipe of other vehicle which was running at idle speed so that the amount of the exhaust CO<sub>2</sub> for supplying the tests was limited.

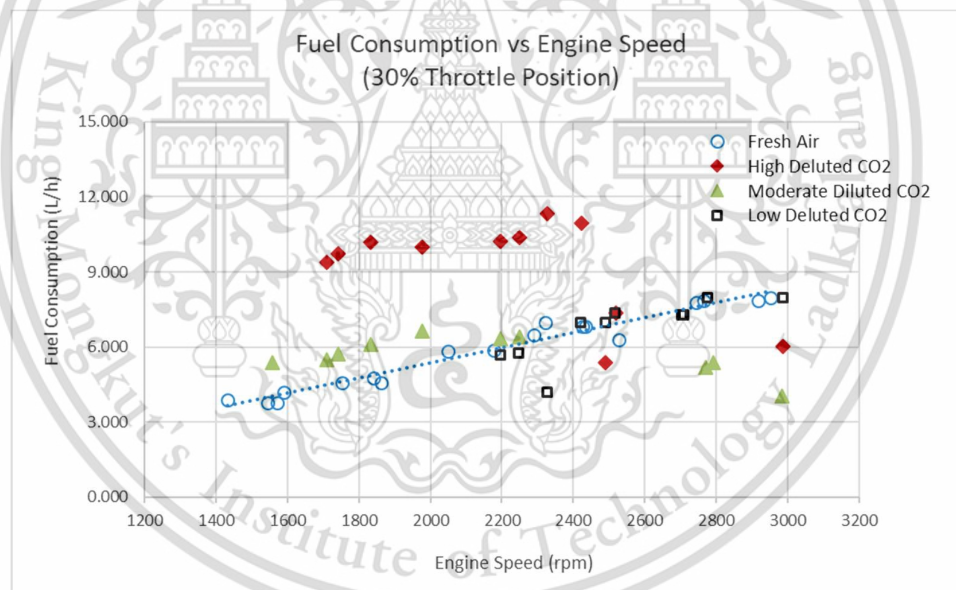


**Figure 4.21** Variation of Input CO<sub>2</sub> Concentration

Due to the testing results which will be discussed in the next section, the experimentations were divided into two intervals, e.g. low-to-moderate CO<sub>2</sub> concentration for the intake CO<sub>2</sub> lower than 1.5 %vol, and high CO<sub>2</sub> concentration for the intake CO<sub>2</sub> higher than 1.5 %vol. Both intervals were analyzed and discussed separately in the tests.

#### 4.2.1 Comparison of the fuel consumption

Figure 4.22 shows experimentation results of corresponding fuel consumption for 30% opened throttle position plotted against the engine speed. The results depict that amount of fuel consumption for combusting the air diluted by low-to-moderate CO<sub>2</sub> concentration is comparable to the combustion of the fresh air, while increase higher intake CO<sub>2</sub> causes higher fuel consumption evidently. Average fuel consumption of combustion with the fresh air intake, low-to-moderate CO<sub>2</sub> intake, and high CO<sub>2</sub> intake are 5.9 L/h, 6.2 L/h, and 9.2 L/h, respectively.



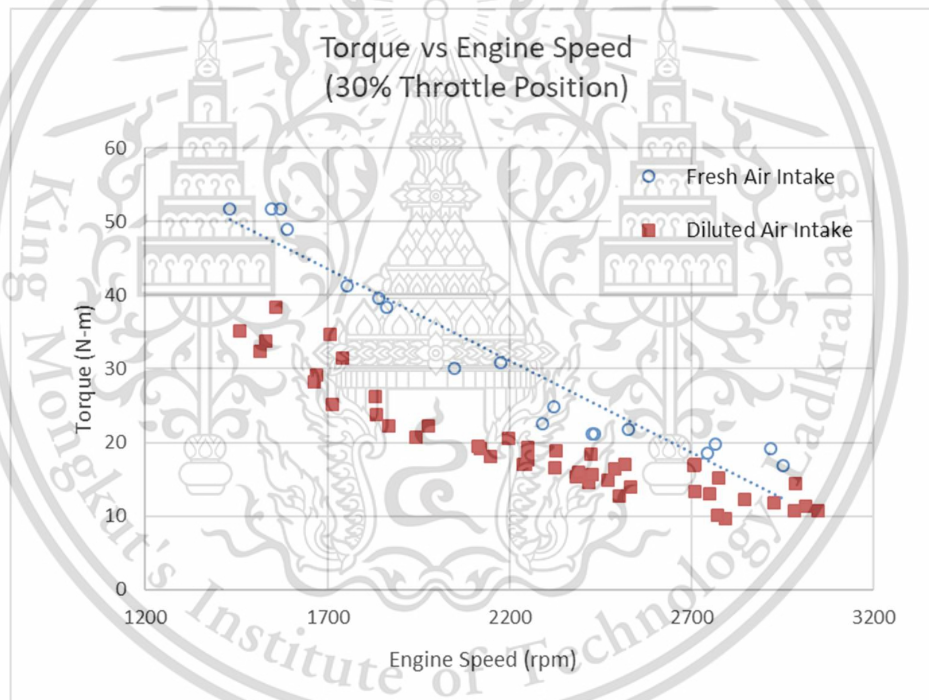
**Figure 4.22** Comparison of Fuel Consumption

As described in section 4.1, high dilution of CO<sub>2</sub> intake causes the combustion reactions to be retarded or slow down, resulting in burning of more fuel during the expansion stroke. Consequently, the combustion was ended before it burns the entire charge and the partially misfired combustion leading to high HC and CO emissions as well as the excess O<sub>2</sub>. Higher value of the excess O<sub>2</sub> may cause the engine controller unit (ECU) implies the combustion as lean (more oxygen) burning and response by

sending command to fuel injectors to inject more fuel to regulate the fuel-air mixture resulting to higher fuel consumption. Such reason satisfies to the experimental results demonstrated in Figure 4.15. Increase of dilution conducts to higher amount of the exhaust free oxygen for about 3.2 %vol averagely, while the maximum value is about 4.1 %vol.

#### 4.2.2 Comparison of the output torque

Figure 4.23 shown that curves of the engine torque decrease linearly according to increment of the engine speed. Dilution of CO<sub>2</sub> to the air intake causes a decrease in the engine torque entire the speed range. Maximum torque decreases about 25.92% comparing to the maximum torque of the fresh air intake.



**Figure 4.23** Comparison of Engine Torque

Dilution the fresh air intake displaced volume of the air required for combustion, hence reduces available oxygen in the combustion chambers and decelerates the reaction rates of the combustion process. When the combustion process is retarded, combustion is later and slow, causing burning to take place throughout the power stroke and continuing into the ensuing exhaust stroke. The result conducts drop of maximum gas pressure in the cylinders which means decrease of available work from the combustion gas.

### 4.2.3 Comparison of the brake power

Figure 4.24 shown that curves of the brake power progressively decrease with the engine speed. Due to linear decrease of the torque curve, similarity between Figure 4.23 and Figure 4.24 could be observed. Maximum power decreases about 26.46% comparing to the maximum power of the fresh air intake.

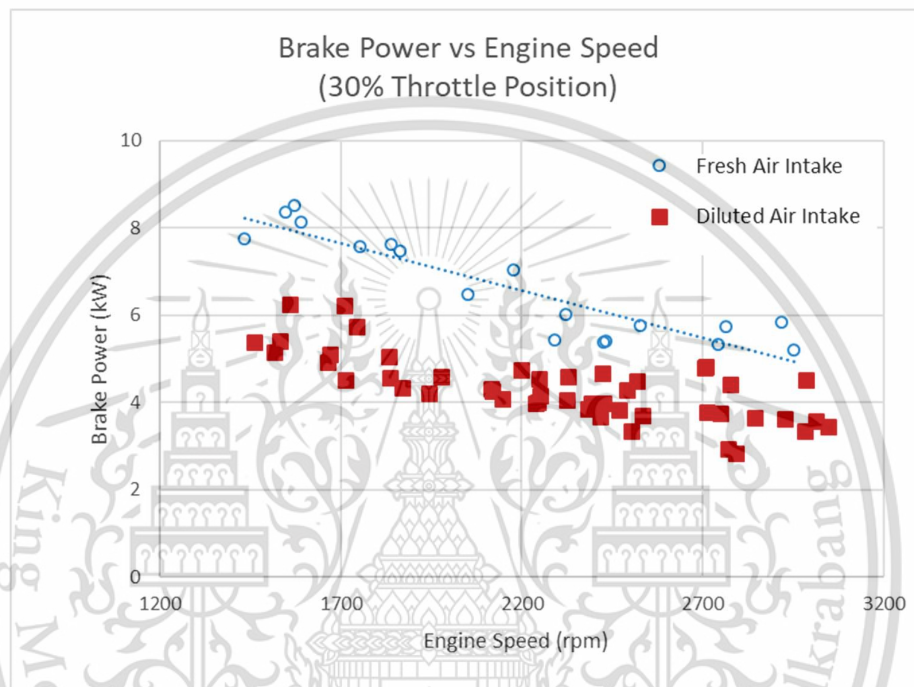
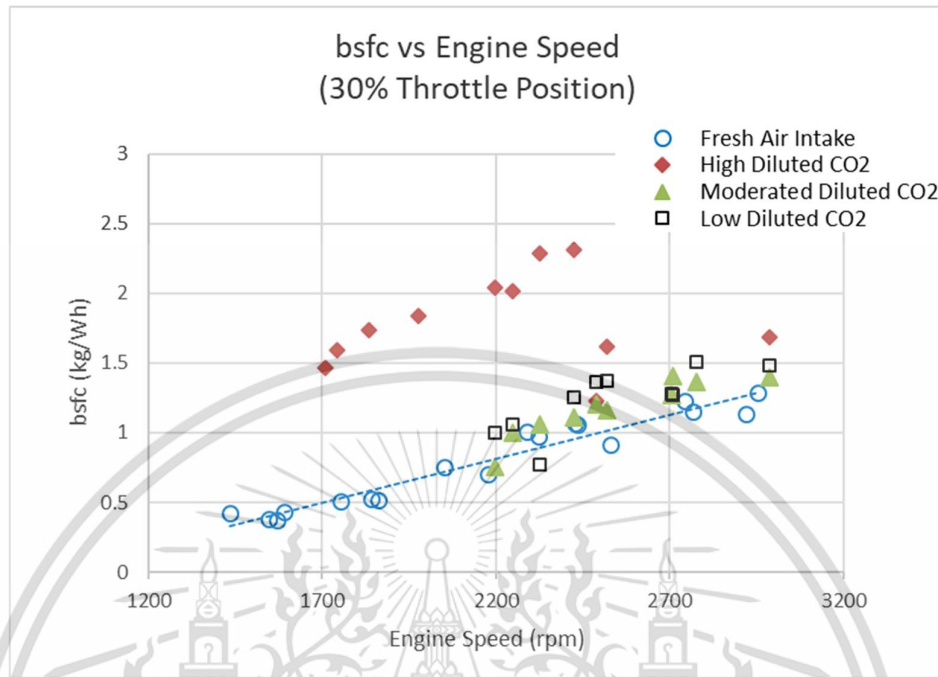


Figure 4.24 Comparison of Brake Power

As well as the engine torque, the brake power developed in the engine is directly proportional to the air mass flow rate into the engine. Dilution the fresh air intake reduces available oxygen in the combustion chambers and causes decrease of the engine torque and the brake power, which is in agreement with the prior studies (Meshack, 2015) (Domenico, 2017) (Tairu, 2017).

### 4.2.4 Comparison of the brake specific fuel consumption

The brake specific fuel consumption (bsfc) is the fuel flow rate required to produce unit brake power from a given engine. Such parameter is inversely proportional to thermal efficiency ( $\eta_{th}$ ). Figure 4.25 shows comparison of brake specific fuel consumption of the against the engine speed. Due to linear variation of both the fuel consumption and the brake power, bsfc curves are similar to the curves in Figure 4.21.



**Figure 4.25** Comparison of Brake Specific Fuel Consumption

## CHAPTER 5

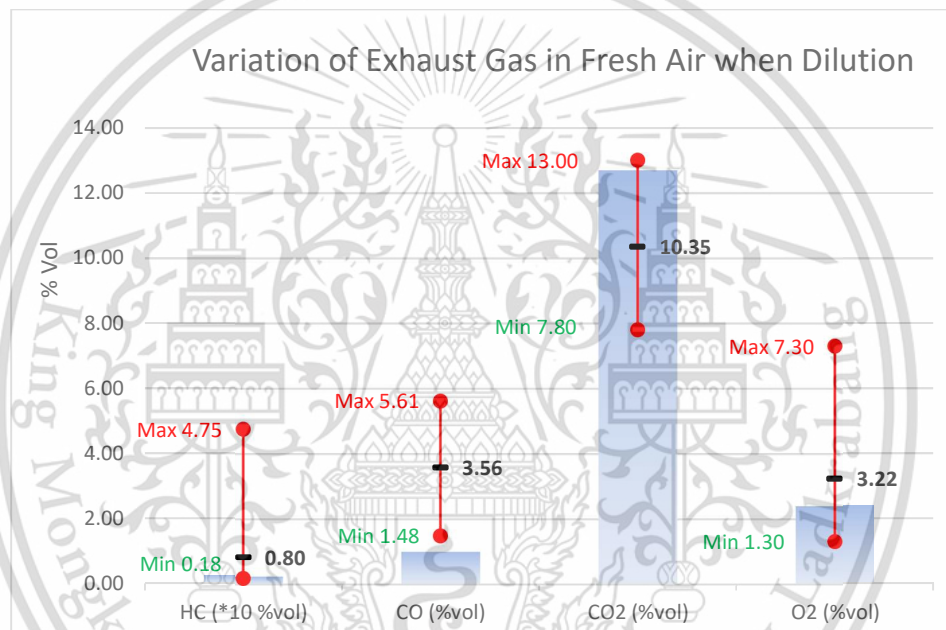
### CONCLUSIONS AND RECOMMENDATIONS

This study of the exhaust gas composition from the combustion in a gasoline engine with various proportion of the polluted air intake was diluted upstream to the engine at the idle speed of 1,100 rpm. Pollution gases composing of CO in range of 0-1.40 %vol., CO<sub>2</sub> in range of 0-5.0 %vol., and HC in range of 0-400 ppm of which proportions were varying was diluted to intake air upstream to the tested engine. The experimental results are demonstrated in Figure 5.1 and summarized as follows:

- Dilution into the intake air resulted the increase of the exhaust HC and CO concentrations. Highest values of the exhaust HC and CO were about 18.9 times and 4.6 times of the normal fresh air intake, respectively.
- Dilution into the intake air resulted the decrease of the exhaust CO<sub>2</sub> concentration. Highest decrease of the exhaust CO<sub>2</sub> was about 38.3%.
- Variation of the excess O<sub>2</sub> in the exhaust gas did not reveal any relation to the varying concentrations of the diluted pollutants.
- The exhaust gas temperature decreased as the fresh air intake was diluted by the pollutants due to incomplete combustion. Highest decrease of the exhaust temperature was about 15.73%.
- The established mathematical model was able to be used only as a predictor to the trend of the exhaust emission. The model could provide consistent results only for the exhaust HC and CO, but the estimated HC was substantial higher than the measured values due to limitation of HC detector. The estimated CO<sub>2</sub> were lower and the estimated O<sub>2</sub> were higher than the actual values. Due to the employed assumptions and difference of variables that were implemented to the mathematical model, there were significant error between the measured results and the calculated results. Therefore, the mathematical model for estimation of the exhaust gas constituent should be improved by considering of the chemical effect, not only considering of the principle of mass conservation which was generally proposed in typical ICE textbook. Separately measured of the exhaust gas composition should be conducted to ensure the variability of the emissions.
- Effect of dilution to the density of the intake air was not significant due to the intake temperature was low.
- According to the experimental results, it was found that dilution of the CO<sub>2</sub> to the air intake has evident effect to the exhaust gas constituent.

After analyzing the experimental results, significant component of the diluted air intake which affects to the exhaust emission was further considered by testing under the normal operating speed range 1,200-3,000 rpm at fixed throttle position of 30% opening. Comparisons of the engine performance parameter to the normal air intake were carried out and discussed subsequently as follows:

- Increase of the fuel consumption about 23.05% averagely
- Decrease of the maximum engine torque about 25.92% averagely
- Decrease of the maximum brake power 26.48% averagely
- Increase of the brake specific fuel consumption 9.74% averagely



**Figure 5.1** Variation of Exhaust Gas in Fresh Air when Dilution

### Improvement and Recommendation

- The Exhaust Gas Analyzer should have higher accuracy for measuring the exhaust NO<sub>x</sub> data.
- Using NDIR detector for measuring the exhaust HC are generally reported as n-hexane equivalent values, therefore FID detector should be employed to measure the total hydrocarbon (THC).
- Varying of the intake CO<sub>2</sub> concentration is limited due to limitation of supplement because the testing car running at idle speed.

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## APPENDIX A

### Specifications of Tested Engine

Engine Model	5A FE (Fuel Injection)
Fuel Type	Gasoline
Car Model	COROLLA, SPRINTER
Number of Cylinders	4
Engine Valves	DOHC 16 Valves
Displacement	1,498 cc.
Bore	78.7 mm.
Stroke	77 mm.
Output Power	100 HP @5,600 rpm
Torque	14 kg-m @4,400 rpm

## APPENDIX B

### Technical Data of Engine Test Bed

Model	MT 505H, Water Absorber
Manufacturer	ESSOM COMPANY LIMITED, Thailand
Dynamometer Type	Water Brake Absorber
Maximum Torque	: At 2,000 rpm over 750 N-m or over 150 kW : At 4,000 rpm over 900 N-m or over 375 kW : At 6,000 rpm over 750 N-m or over 450 kW
Maximum Speed	7,000 rpm
Engine	Gasoline / Diesel Engine
Cooling System	Cooling Tower with circulating pumps
Measuring Instruments	
- Torque	A load cell and an indicator
- Speed	A speed sensor and an indicator
- Fuel Flow Rate	A graduated cylinder and a fuel flow sensor
- Air Flow Rate	An air flow sensor and an indicator
- Temperature	Sensors and an indicator with a selector switch for engine water inlet and outlet, absorber water outlet, exhaust gas, and ambient air
Other Accessories	A battery, an alternator, a starter, and a fuel tank
Power Supply	220 Volt, single phrase, 50 Hz

APPENDIX C

# EMS

EMISSIONS SYSTEMS, INC.

## Portable Exhaust Gas Analyzer Operators Manual

Model # 5002 (4 & 5 Gas) Model # 8000 (Wireless)



## Technical Data

Power: 10 -16 VDC  
Ranges: HC: 0 - 2000 ppm (0-20,000 ppm High Range)  
CO: 0 - 10%  
CO2: 0 - 20%  
O2: 0 - 25%  
NO: 0 - 5000 ppm ( Nitric Oxide ) \*

Warm up: Less than 5 minutes  
Display resolution: HC: 1 ppm vol.  
CO: 0.01% vol.  
CO2: 0.1% vol.  
O2: 0.01% vol.  
NO: 1ppm\*

Accuracy ( Bar 97 EPA ASM ) HC: 4 ppm HC  
CO: 0.06% CO  
CO2: 0.3% CO2  
O2: 0.1% O2  
NO: 25 ppm\*

Drift: Zero and span drift are less than  $\pm 0.6\%$  of full scale for the first hour and less than  $\pm 0.4\%$  of full scale per hour thereafter.

System response time: Bench: 1.5 Sec/25 ft. hose 5 sec. to 90% of final reading

Ambient conditions: 35F (2C) to 120F (45C), rel. humidity 0-98%

Sample hose:25 feet ( 7.5 m) with QD coupling /200 F Degrees Max.

Sample probe:Stainless steel 1200 Degrees F ( replaceable flex tip )

Mass: Approx.: 10 LBS. ( 4.5 Kg )

## APPENDIX D

No.	Intake Air						Exhaust Gases				
	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	NO <sub>x</sub>	T	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	T
1	0	0.00	0.0	20.8	0.00	31.6	238	1.00	12.6	2.6	312
2	0	0.00	0.0	20.8	0.00	31.6	238	1.00	12.7	2.4	312
3	0	0.00	0.0	20.8	0.00	31.8	239	0.99	12.7	2.5	312
4	0	0.00	0.0	20.8	0.00	32.1	238	1.00	12.6	2.3	312
5	0	0.00	0.0	20.8	0.00	32	239	0.99	12.6	2.3	312
6	21	0.08	0.1	20.2	0.00	31.6	175	1.51	11.5	3.5	327
7	20	0.08	0.1	20.2	0.00	31.9	175	1.53	11.5	3.5	327
8	20	0.08	0.1	20.2	0.00	32	175	1.50	11.5	3.5	327
9	20	0.08	0.1	20.2	0.00	32.1	176	1.50	11.5	3.5	327
10	20	0.08	0.1	20.2	0.00	31.8	177	1.52	11.5	3.5	327
11	28	0.15	0.3	19.9	0.00	32.1	238	1.85	11.4	3.2	334
12	28	0.14	0.3	19.9	0.00	32.1	238	1.82	11.4	3.2	334
13	28	0.14	0.3	19.9	0.00	31.9	238	1.85	11.5	3.2	334
14	29	0.14	0.3	19.9	0.00	32	238	1.84	11.5	3.2	334
15	29	0.14	0.3	19.9	0.00	31.5	239	1.87	11.4	3.2	334
16	6.00	0.00	0.5	19.9	0.00	30.8	224	1.85	12.3	2.4	302
17	6.00	0.00	0.5	19.9	0.00	30.8	224	1.85	12.2	2.4	302
18	7.00	0.00	0.5	19.9	0.00	30.6	226	1.85	12.2	2.4	302
19	35	0.21	0.4	19.8	24.00	31.1	245	2.01	11.8	2.7	328
20	35	0.21	0.4	19.8	24.00	31.3	247	2.01	11.8	2.7	328
21	36	0.22	0.4	19.8	24.00	31.3	248	2.02	11.8	2.7	328
22	36	0.22	0.4	19.8	24.00	31.8	249	2.02	11.9	2.6	328
23	37	0.22	0.4	19.8	24.00	31.7	248	2.02	11.9	2.5	328
24	7.00	0.00	0.5	19.8	0.00	31.8	227	1.85	12.2	2.3	302
25	7.00	0.00	0.6	19.8	0.00	32	227	1.83	12.2	2.3	302
26	50	0.26	0.5	19.4	0.00	32	188	1.86	12.2	2.4	339
27	51	0.26	0.5	19.4	0.00	32.2	243	1.85	12.1	2.4	339
28	51	0.27	0.5	19.4	0.00	32.2	188	1.85	12.1	2.4	339
29	51	0.26	0.5	19.4	0.00	32.1	289	1.85	12.1	2.4	339
30	51	0.25	0.5	19.4	0.00	32.3	189	1.84	12.2	2.4	339

No.	Intake Air						Exhaust Gases				
	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	NO <sub>x</sub>	T	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	T
31	84.00	0.31	0.60	19.20	0.00	32.2	238	2.36	10.70	3.80	322
32	85.00	0.31	0.60	19.20	0.00	32.2	238	2.32	10.70	3.90	322
33	85.00	0.32	0.60	19.20	0.00	32.2	237	2.30	10.70	3.90	322
34	85.00	0.31	0.60	19.20	0.00	32	237	2.32	10.70	3.90	322
35	85.00	0.30	0.60	19.20	0.00	32.1	237	2.31	10.70	3.90	322
36	12.00	0.00	1.0	19.2	45.00	32.3	237	1.49	13.0	1.6	299
37	12.00	0.00	1.0	19.2	35.00	32.3	242	1.49	13.0	1.6	299
38	12.00	0.00	1.1	19.1	35.00	32.3	246	1.49	13.0	1.6	299
39	12.00	0.00	1.1	19.0	35.00	32.3	247	1.48	13.0	1.6	299
40	12.00	0.00	1.1	19.0	35.00	32.3	250	1.49	13.0	1.6	299
41	82	0.39	0.7	19.0	0.00	32.3	248	2.82	10.3	4.0	313
42	82	0.40	0.7	19.0	0.00	32.3	248	2.80	10.2	4.1	313
43	82	0.41	0.8	19.0	0.00	32.4	248	2.84	10.2	4.0	313
44	82	0.40	0.7	19.0	0.00	32.5	248	2.85	10.2	4.0	313
45	82	0.41	0.8	19.0	0.00	32.7	248	2.85	10.3	4.0	313
46	110.00	0.39	0.80	18.80	0.00	32.8	240	2.79	10.60	3.50	318
47	111.00	0.39	0.80	18.80	0.00	32.8	241	2.79	10.60	3.50	318
48	111.00	0.39	0.80	18.80	0.00	33	241	2.82	10.60	3.50	318
49	112.00	0.39	0.80	18.80	0.00	33.1	243	2.83	10.60	3.50	318
50	112.00	0.41	0.80	18.80	0.00	32.6	243	2.81	10.60	3.50	318
51	120	0.46	0.9	18.7	0.00	32.6	228	2.88	11.5	2.3	342
52	120	0.46	0.9	18.7	0.00	32.6	234	3.02	11.4	2.4	342
53	121	0.47	0.9	18.7	0.00	32.5	237	3.01	11.4	2.4	342
54	121	0.47	0.9	18.7	0.00	32.6	238	3.01	11.4	2.4	342
55	122	0.48	0.9	18.7	0.00	32.6	239	3.03	11.4	2.4	342
56	168.00	0.51	1.10	18.40	0.00	32.6	283	2.85	12.30	1.30	317
57	167.00	0.51	1.10	18.40	0.00	32.6	283	2.84	12.30	1.30	317
58	166.00	0.51	1.10	18.40	0.00	32.6	285	2.84	12.30	1.30	317
59	165.00	0.50	1.10	18.40	0.00	32.5	285	2.85	12.30	1.30	317
60	164.00	0.50	1.10	18.40	0.00	32.5	286	2.85	12.30	1.30	317
61	20.00	0.01	1.6	18.2	10.00	32.5	297	2.61	12.5	1.3	296
62	20.00	0.00	1.6	18.2	10.00	32.6	297	2.60	12.5	1.3	296
63	20.00	0.00	1.7	18.2	10.00	32.6	299	2.59	12.5	1.3	296
64	20.00	0.00	1.7	18.2	10.00	32.5	299	2.58	12.5	1.3	296
65	21.00	0.00	1.7	18.2	10.00	32.5	300	2.55	12.5	1.3	296

No.	Intake Air						Exhaust Gases				
	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	NO <sub>x</sub>	T	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	T
66	103.00	0.61	1.20	18.10	0.00	32.5	374	3.83	10.50	2.80	286
67	103.00	0.61	1.20	18.10	0.00	32.4	376	3.83	10.50	2.80	286
68	103.00	0.62	1.20	18.10	0.00	32.3	376	3.81	10.50	2.80	286
69	103.00	0.62	1.20	18.10	0.00	32.3	376	3.81	10.50	2.80	286
70	103.00	0.62	1.20	18.10	0.00	32.3	376	3.81	10.50	2.90	286
71	140	0.75	1.4	17.7	0.00	32.3	363	4.44	10.6	2.0	324
72	140	0.74	1.4	17.7	0.00	32.4	363	4.44	10.7	2.0	324
73	139	0.74	1.4	17.7	0.00	32.4	363	4.47	10.7	1.9	324
74	139	0.73	1.4	17.7	0.00	32.4	363	4.46	10.7	1.9	324
75	139	0.75	1.4	17.7	0.00	32.4	362	4.46	10.6	1.9	324
76	130.00	0.73	1.40	17.60	0.00	32.4	386	3.66	10.50	2.80	277
77	130.00	0.72	1.40	17.60	0.00	32.4	385	3.64	10.50	2.90	277
78	130.00	0.72	1.40	17.60	0.00	32.3	386	3.66	10.50	2.90	277
79	129.00	0.72	1.40	17.60	0.00	32.3	386	3.66	10.40	2.90	277
80	129.00	0.70	1.40	17.60	0.00	32.3	385	3.66	10.50	2.90	277
81	8.00	0.00	2.0	17.6	0.00	32.3	483	3.82	10.2	3.1	293
82	8.00	0.00	2.0	17.6	0.00	32.3	488	3.85	10.2	3.0	293
83	8.00	0.00	2.0	17.6	6.00	32.3	505	3.87	10.3	2.9	293
84	8.00	0.00	2.0	17.6	6.00	32.3	507	3.88	10.4	2.8	293
85	8.00	0.00	2.0	17.6	6.00	32.3	515	3.89	10.4	2.8	293
86	137.00	0.80	1.60	17.30	0.00	32.3	420	3.83	10.20	3.10	274
87	137.00	0.80	1.60	17.30	0.00	32.4	424	3.83	10.20	3.10	274
88	137.00	0.80	1.60	17.30	0.00	32.4	424	3.82	10.20	3.10	274
89	136.00	0.79	1.60	17.30	0.00	32.4	429	3.87	10.20	3.10	274
90	136.00	0.79	1.60	17.30	0.00	32.4	425	3.83	10.20	3.10	274
91	4.00	0.00	2.4	17.2	0.00	32.4	447	3.80	10.0	3.6	290
92	4.00	0.00	2.3	17.2	0.00	32.4	453	3.81	10.0	3.6	290
93	4.00	0.00	2.4	17.2	0.00	32.5	455	3.81	10.0	3.6	290
94	5.00	0.00	2.4	17.2	0.00	32.5	458	3.82	10.0	3.6	290
95	5.00	0.00	2.4	17.2	0.00	32.5	458	3.81	10.0	3.6	290
96	160	1.03	2.0	16.7	0.00	32.5	551	4.75	9.6	3.2	297
97	160	1.03	2.0	16.7	0.00	32.5	551	4.73	9.6	3.3	297
98	160	1.05	2.0	16.7	0.00	32.5	552	4.71	9.6	3.3	297
99	161	1.08	2.1	16.6	0.00	32.5	551	4.72	9.6	3.3	297
100	160	1.06	2.1	16.6	0.00	32.6	557	4.71	9.6	3.3	297

No.	Intake Air						Exhaust Gases				
	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	NO <sub>x</sub>	T	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	T
101	311.00	1.00	2.20	16.40	0.00	32.6	637	5.03	9.20	3.50	267
102	312.00	1.00	2.20	16.40	0.00	32.6	645	5.04	9.10	3.50	267
103	314.00	1.00	2.20	16.40	0.00	32.6	661	5.06	9.10	3.50	267
104	7.00	0.00	2.9	16.4	29.00	32.6	754	4.60	10.1	2.7	281
105	191	1.09	2.1	16.4	0.00	32.6	611	4.92	9.1	3.5	280
106	191	1.10	2.1	16.4	0.00	32.6	625	4.89	9.2	3.4	280
107	190	1.09	2.1	16.4	0.00	32.6	623	4.86	9.2	3.4	280
108	190	1.10	2.1	16.4	0.00	32.6	624	4.82	9.2	3.4	280
109	190	1.10	2.1	16.4	0.00	32.6	619	4.82	9.2	3.4	280
110	332.00	1.00	2.20	16.30	0.00	32.6	641	5.06	9.10	3.50	267
111	332.00	1.01	2.20	16.30	0.00	32.6	629	5.03	9.20	3.50	267
112	7.00	0.00	3.0	16.3	29.00	32.6	789	4.59	10.1	2.7	281
113	7.00	0.00	3.0	16.3	29.00	32.6	760	4.59	10.1	2.7	281
114	7.00	0.00	3.0	16.3	29.00	32.6	773	4.59	10.1	2.7	281
115	7.00	0.00	2.9	16.3	29.00	32.6	772	4.51	10.1	2.7	281
116	325.00	1.09	2.40	16.00	0.00	32.6	945	5.41	9.10	3.10	263
117	10.00	0.00	3.3	16.0	0.00	32.6	1,005	4.73	9.4	3.4	280
118	9.00	0.00	3.3	16.0	0.00	32.6	1,008	4.75	9.4	3.4	280
119	10.00	0.00	3.3	16.0	0.00	32.6	1,034	4.75	9.4	3.4	280
120	10.00	0.00	3.3	16.0	0.00	32.6	1,034	4.75	9.4	3.5	280
121	10.00	0.00	3.3	16.0	0.00	32.6	1,004	4.78	9.6	3.3	280
122	198	1.22	2.4	16.0	0.00	32.6	867	5.27	8.5	4.1	288
123	198	1.22	2.5	16.0	0.00	32.6	908	5.39	8.6	4.1	288
124	200	1.22	2.5	16.0	0.00	32.6	907	5.40	8.5	4.0	288
125	200	1.22	2.5	16.0	0.00	32.6	881	5.34	8.6	4.1	288
126	201	1.22	2.5	16.0	0.00	32.6	889	5.34	8.6	4.1	288
127	330	1.11	2.40	15.90	0.00	32.6	904	5.42	9.10	3.10	263
128	329	1.10	2.40	15.90	0.00	32.6	980	5.42	9.10	3.10	263
129	357	1.10	2.40	15.90	0.00	32.6	980	5.40	9.10	3.10	263
130	360	1.11	2.40	15.90	0.00	32.6	964	5.40	9.10	3.10	263
131	236	1.32	2.5	15.6	0.00	32.6	2030	5.36	8.8	3.6	287
132	236	1.31	2.5	15.6	0.00	32.6	1970	5.41	8.8	3.6	287
133	236	1.32	2.6	15.6	0.00	32.5	2020	5.37	8.8	3.6	287
134	237	1.32	2.6	15.6	0.00	32.5	1911	5.42	8.8	3.5	287
135	237	1.31	2.6	15.6	0.00	32.5	1938	5.42	8.8	3.4	287

No.	Intake Air						Exhaust Gases				
	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	NO <sub>x</sub>	T	HC	CO	CO <sub>2</sub>	O <sub>2</sub>	T
136	285	1.28	2.7	15.5	0.00	32.5	1278	5.59	9.0	3.1	279
137	285	1.27	2.7	15.5	0.00	32.5	1298	5.58	9.0	3.1	279
138	285	1.27	2.7	15.5	0.00	32.5	1292	5.59	9.0	3.1	279
139	285	1.27	2.7	15.5	0.00	32.5	1284	5.61	9.0	3.1	279
140	285	1.28	2.7	15.4	0.00	32.5	1290	5.59	9.0	3.1	279
141	4.00	0.00	4.0	15.2	0.00	32.5	1,306	4.96	9.4	3.5	278
142	4.00	0.00	4.1	15.1	0.00	32.5	1,464	4.96	9.4	3.7	278
143	4.00	0.00	4.0	15.1	0.00	32.5	1,415	4.96	9.4	3.6	278
144	4.00	0.00	4.1	15.0	0.00	32.5	1,486	4.84	9.4	3.6	278
145	4.00	0.00	4.1	15.0	0.00	32.5	1,487	4.84	9.4	3.7	278
146	11.00	0.01	4.4	14.5	28.00	32.5	4,410	3.26	8.3	6.4	272
147	12.00	0.01	4.5	14.4	28.00	32.5	4,460	3.18	8.2	6.6	272
148	12.00	0.01	4.6	14.3	43.00	32.5	4,750	2.98	7.9	7.0	272
149	12.00	0.01	4.6	14.3	36.00	32.5	4,720	3.04	8.0	6.9	272
150	12.00	0.01	4.6	14.3	28.00	32.4	4,550	3.09	8.0	6.7	272
151	67.00	0.03	4.8	13.9	0.00	32.5	4,400	2.53	7.8	7.3	264
152	69.00	0.03	4.8	13.8	0.00	32.4	3,720	2.86	8.2	6.6	264
153	68.00	0.03	4.8	13.8	0.00	32.3	4,310	2.56	8.0	7.2	264
154	70.00	0.03	4.8	13.7	0.00	32.3	3,590	2.88	8.2	6.5	264
155	69.00	0.03	4.8	13.7	0.00	32.3	3,680	2.86	8.2	6.6	264



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### Topic






- Call for papers
- Abstract submission deadline
- Notification of abstract acceptance
- Full manuscript submission deadline
- Notification of full manuscript acceptance
- Early bird registration deadline
- Camera-ready manuscript submission deadline
- Conference dates

### Dates

Jan 1, 2015  
 Extended to **May 30, 2016**  
 June 4, 2015 June 15, 2015  
 August 1, 2015 **Extended to August 7, 2016**  
 September 1, 2015 **Extended to September 15, 2015**  
 October 5 - November 10, 2015  
 October 16 - 2016 October 22, 2015  
 December 16-18, 2015

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 The TSME-ICoME 2015 organizing committee would like to take this opportunity to thank everyone for participating this conference and all event's activities. We very much appreciate and are deeply grateful for your kind assistance and co-

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	Today	6
	Yesterday	40
	This week	78
	This month	48
	Total	127302

Wednesday, 02 May 2018 03:54



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# Experimental Study of Exhaust Gas Composition from Combustion in Gasoline Engine with Polluted Air Intake

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## Abstract

This research was an experimental study of exhaust gas composition from combustion in a gasoline engine with polluted air intake. The experimentations were tested at the engine's idle speed with removal of oxygen sensor. Pollution gases composing of carbon monoxide CO in range of 0 to 1.40 % vol., carbon dioxide CO<sub>2</sub> in range of 0 to 5.0 % vol., and unburned hydrocarbon HC in range of 0 to 400 ppm of which proportions were varying was diluted to intake air upstream to the tested engine. Compositions and temperature of the exhaust gas from the exhaust pipe were sampled and analyzed through Exhaust Gas Analyzer in order to determine the concentrations of CO, CO<sub>2</sub>, HC, and excess O<sub>2</sub>. Experimentation results revealed that increase of the HC, CO, and CO<sub>2</sub> concentrations to the intake air caused increase of the exhausted HC and CO concentration while the CO<sub>2</sub> concentration in the exhaust gas decreased. The results also showed that the variation of the excess O<sub>2</sub> in the exhaust gases did not relate to the varying concentrations of the diluted pollutants. Comparison of the exhaust gas composition predicted from the established mathematical model to the experimental results showed that implementation of the established model only provided the variation tendency of the exhausted CO and CO<sub>2</sub> consistent to the experimental results. Therefore, measurements of the actual exhaust gas composition were still necessary to ensure accuracy results.

**Keywords:** Exhaust gases; Gasoline engine; Pollution; Exhaust gas analyzer; Combustion

## 1. Introduction

Exhaust gas composition depends on the relative proportions of fuel and air fed to the engine, fuel composition, and completeness of combustion process [1]. However, local chemistry of the burning process and the chamber geometry will also have a profound influence on the final composition of any exhaust gas [2]. In reality, dissociation reactions take place at elevated temperatures and pressured even under stoichiometric combustions, some free carbon monoxide (CO) and hydrogen (H<sub>2</sub>) will be created, so that there would be some carbon monoxide (CO) in the existence and minor amount of oxygen (O<sub>2</sub>) and hydrogen (H<sub>2</sub>). As the air-fuel mixture is progressively richer than stoichiometric proportion, the exhaust gas would contain greater amounts of CO and H<sub>2</sub> while would still show little traces of free oxygen O<sub>2</sub>. There are two principal dissociation reactions involved; 1)  $CO + \frac{1}{2}O_2 \rightleftharpoons CO_2$  and 2)  $H_2O + CO \rightleftharpoons H_2 + CO_2$ . Maximum temperature of combustion is limited by the first reaction when it is proceeding as energy is released. The second reaction is often called the "water gas reaction". In addition, incompletely burning of the fuel and non-uniform air-fuel mixture fed to each cylinder are also affect the composition of exhaust gas. Although the combustion equations are useful for determining the combustion products, they do not correspond closely to the actual constituents. For all these reasons, the composition of the exhaust gases cannot be calculated and predicted easily. The best way to determine the composition of the engine exhaust gases is to measure directly.

Experimental work conducted by Martin A. Elliott, et al. [3] on the motor coaches had shown that driving condition had a marked effect on the emission rate of all constituents. The highest CO concentration in the exhaust gas occurred under idle condition. The largest NO<sub>x</sub> concentration was at cruise and acceleration conditions and decreased below 60 ppm during idle and deceleration condition. The engine conditions giving rise to high HC concentration were observed at both deceleration and idle conditions; more than 34% of the supplied fuel was probably wasted during deceleration condition.

S.H. Graf, G.W. Gleeson, and W.H. Paul [4] reviewed correlations between the exhaust gas constituents of gasoline engines from various researchers and experimentations. The correlation between CO<sub>2</sub>-O<sub>2</sub> was applicable for lean mixtures but for extremely rich mixtures it was advised to measure directly. There is a distinct correlation between CO-CO<sub>2</sub> and could be of practical used as a time-saving device in exhaust gas analysis. The correlation between CO-H<sub>2</sub> described that the ratio/slope of CO to H<sub>2</sub> in percentage by volume was non-linear over the range of values which were varying from 1.96 to 2.9.

Claudio Mazzoleni, et al. [5] showed that the concentration of HC, CO, NO, and particulate matters on an individual vehicle were poorly correlated. Therefore, one or more pollutants cannot be used as predictor for other pollutants, but it should be measured separately to ensure that the variability of the emissions was properly characterized.

Effects of intake pollutant to the engine emission were mostly studied through varying of rate of EGR (Exhaust Gases Recirculation), while the effects of

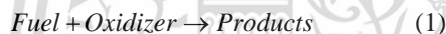
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proportion of the pollutant concentration have not been studied as much. Applications of EGR could reduce a large amount of NO<sub>x</sub> emission, but it leads to increase of CO and HC emissions [1, 6]. Increase of EGR rate results to slow combustion speeds which lead to partial burning or misfire that produces higher CO and HC [6].

This research is an experimental study of exhaust gas composition from combustion in a gasoline engine with polluted air intake and is tested at the engine's idle speed of 1,150 rpm with removal of oxygen sensor. Various proportion of pollution gas concentrations are diluted to the intake air upstream to the engine. The exhaust gas composition and temperature are sampled and analyzed through exhaust gas analyzer in order to determine the concentrations of CO, CO<sub>2</sub>, HC, and excess oxygen O<sub>2</sub>. Experimentation results are used to investigate the effects of each intake pollutant element to the exhaust gases emission. Comparisons to an established mathematical model and prior studies are also considered to further discussion.

### 2. Theory

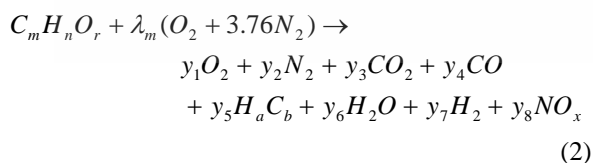
An overall combustion reaction in the gasoline engine can be written as



Generally, the fuel is hydrocarbon fuel, the oxidizer normally is air (O<sub>2</sub>+3.76N<sub>2</sub>), and the products are CO<sub>2</sub>, H<sub>2</sub>O, CO, H<sub>2</sub>, O<sub>2</sub>, NO<sub>x</sub>, N<sub>2</sub>, unburned hydrocarbon (HC), and soot particles which are mainly solid carbon and usually can be omitted from the analysis. Among of the combustion products, CO, HC, NO<sub>x</sub>, and CO<sub>2</sub> are considered as pollutants [2].

Air-fuel ratio is one of the most important parameter that affects the engine exhaust emissions. Variation in emissions with air-fuel ratio for any typical premixed gasoline engines is demonstrated in reference [7].

The actual combustion equations including the effects of dissociations for any generic hydrocarbon fuel containing alcohols C<sub>m</sub>H<sub>n</sub>O<sub>r</sub> in the internal combustion engine can be demonstrated explicitly as



where  $\lambda_m$  is the molecular air-fuel ratio, and  $y_1, y_2, y_3, y_4, y_5, y_6, y_7, y_8$  are the number of mole of the combustion products, and  $a, b$  are the number of hydrogen atom and carbon atom in the unburned hydrocarbon, respectively.

To solve for the unknowns in Eq. (2), various assumptions are determined and it is required carbon balance, hydrogen balance, oxygen balance, nitrogen

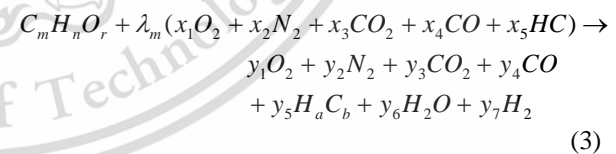
balance, and additional equations to complete the analysis.

### 3. Experimental Methodology

The experimentation using engine model: Toyota 5A-FE 1600 cc. which are tested at the engine's idle speed of 1,150 rpm. The oxygen sensor had been removed so as to run the engine at the constant fuel injection rate. The engine was turned into the safe mode operation; the manufacturer's set fuel injection rate. Two vehicles with and without catalytic converter are employed to generated pollution gases from combusting gasohol-91 fuel. The pollution gases composing of CO in range of 0-1.40 %vol., CO<sub>2</sub> in range of 0-5.0 %vol., and HC in range of 0-400 ppm of which proportion is varied and diluted to the intake fresh air in a mixing tank upstream to the tested engine. Compositions and temperature of the exhaust gases from the exhaust pipe before entering the catalytic converter are sampled and analyzed through Exhaust Gas Analyzer EMS Model 5002 in order to determine the concentrations of CO, CO<sub>2</sub>, HC, and excess oxygen O<sub>2</sub>.

#### 3.1 Mathematical Modeling

To predict the exhaust gas composition, mathematical model is established. It is well aware that using the exhausted gas as EGR is purposely to reduce NO<sub>x</sub> formation. However, the gasoline engines was typically operated in the fuel-rich condition or close to the stoichiometric condition, so that the amount of NO<sub>x</sub> in the exhaust tails is comparatively much less than the others; CO, CO<sub>2</sub> and HC. Only trace amount of NO were recorded from all tests and the combustion products, NO<sub>x</sub> was assumed to transform to N<sub>2</sub>. Hence, the equation of combustion in the gasoline engine with polluted air can be modified from Eq. (2) as follow:



where the molecular air-fuel ratio  $\lambda_m$  is derived from experimental measurement at the engine's idle speed and  $x_1, x_2, x_3, x_4, x_5$  are the mole fraction of the diluted air composing of dry pollutant gases and induced fresh air which are measured from the exhaust gas analyzer. so that,

$$x_1 + x_2 + x_3 + x_4 + x_5 = 1 \quad (4)$$

To simplify the analysis, we assume that all moles of the unburned hydrocarbon H<sub>a</sub>C<sub>b</sub> in Eq. (3) have values of  $a=b=1$ . Thus, there are 7 unknowns to be solved in Eq. (3), e.g.,  $y_1, y_2, y_3, y_4, y_5, y_6, y_7$  which are needed to be solved simultaneously through carbon balance, hydrogen balance, oxygen balance, nitrogen balance, and additional three equations as follows:

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Carbon balance:

$$m + \lambda_m(x_3 + x_4 + x_5) = y_3 + y_4 \quad (5)$$

Hydrogen balance:

$$n + \lambda_m(x_5) = 2y_6 + 2y_7 \quad (6)$$

Oxygen balance:

$$r + \lambda_m(2x_1 + x_3 + 2x_4) = 2y_1 + y_3 + 2y_4 + y_6 \quad (7)$$

Nitrogen balance:

$$\lambda_m(x_2) = y_2 \quad (8)$$

An equation concerning to the equilibrium constant for the dissociation reaction of  $H_2 + CO_2 \Leftrightarrow H_2O + CO$  can be written as

$$k_{H_2O+CO} = \frac{y_4 y_6}{y_3 y_7} \quad (9)$$

where the value of  $k_{H_2O+CO}$  is an empirical constant and is commonly recommended to be 3.5 or 3.8 [1].

Another two equations implemented to solve the solution derived from the experimental data between the exhaust gas  $CO_2$ - $O_2$  and the exhaust gas  $CO$ - $CO_2$  [4] are as below

$$[CO_2] = 14.023 - 0.674[O_2] \quad (10)$$

$$[CO] = 22.25 - 1.66[CO_2] \quad (11)$$

where  $[CO]$ ,  $[CO_2]$ , and  $[O_2]$  are the percentage by volume of  $CO$ ,  $CO_2$ , and  $O_2$ , respectively.

### 3.2 Experimental Apparatus

The photography of the experimental apparatus employed for this study is shown in Fig. 1. Details of the sampling points to be measured are demonstrated as schematic diagram in Fig. 2.



Fig.1 Photography of Experimental Apparatus

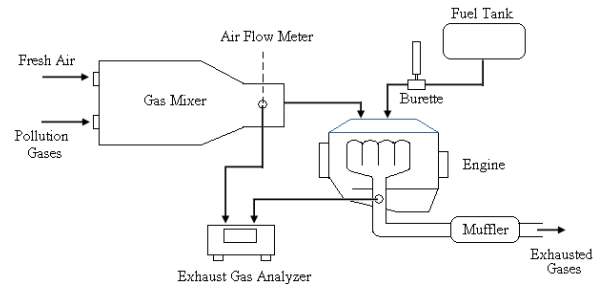


Fig. 2 Schematic diagram of the experimentations

## 4. Results and Discussion

Experimental results are used to investigate the effects of each pollution gases to the exhaust gases emission combusted from the gasoline engine. And also, comparisons to the mathematical model and prior studies are considered to further discussion.

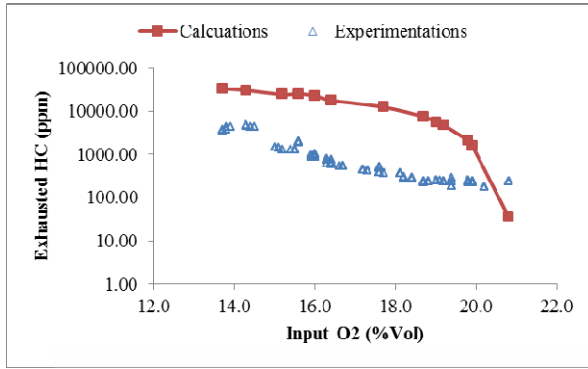
### 4.1 Effects of Intake Pollutants to the Exhaust Gas Composition

Figs. 3-6 show correlation of varying proportion of the polluted air and the experimental data derived from analysis of the dry exhaust gas composition, e.g.,  $HC$ ,  $CO$ ,  $CO_2$ , and  $O_2$  which are explained according to each intake elements.

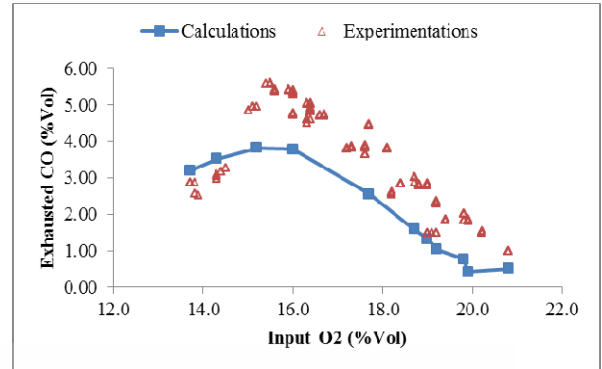
Amount of the exhausted  $HC$  are unburned fuel components which occur in the exhaust emission after incomplete combustion. Figs. 3(a)-3(d) demonstrate the variation of the exhausted  $HC$  to each intake polluted air element  $O_2$ ,  $CO_2$ ,  $CO$ , and  $HC$ , respectively. Decrease of the intake  $O_2$  or increase of the intake  $CO_2$  cause rise of the exhausted  $HC$  obviously until to certain values 14% of  $O_2$  and 5% of  $CO_2$ , the tested engine are instability due to misfiring. Variation of the exhausted  $HC$  to the intake  $CO$  and the intake  $HC$  do not show evident correlations. However, the solutions from the established mathematical model are extremely higher than the experimental results as shown on logarithm scale of the exhausted  $HC$ . These lower amounts of measured  $HC$  concentration may be due to only evaporated  $HC$  emissions were measured by the exhaust gas analyzer while the condensed  $HC$  in the sampling line between the sampling point to the measurement point could not be measured in the exhaust gas analyzer. Therefore, installation of a heated sampling device is important for sampling unburned  $HC$  emissions. It allows the exhaust gas to be ducted from the sampling point to the measuring point across an electrically heated line. This approach could eliminate system failure caused by condensation in the sampling line [8].

Figs. 4(a)-4(d) demonstrate the variation of the exhausted  $CO$  to each intake polluted air element  $O_2$ ,  $CO_2$ ,  $CO$ , and  $HC$ , respectively. Increase of all the intake pollutants  $CO_2$ ,  $CO$  and  $HC$  result to rise of the

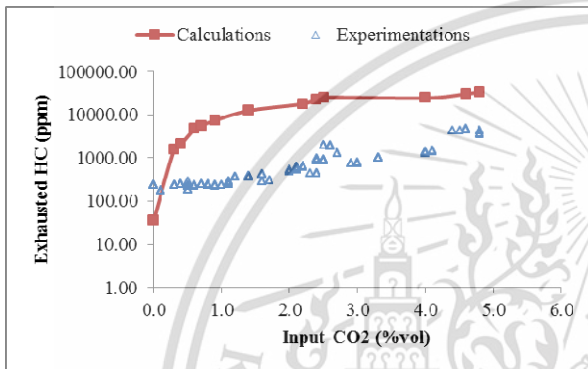
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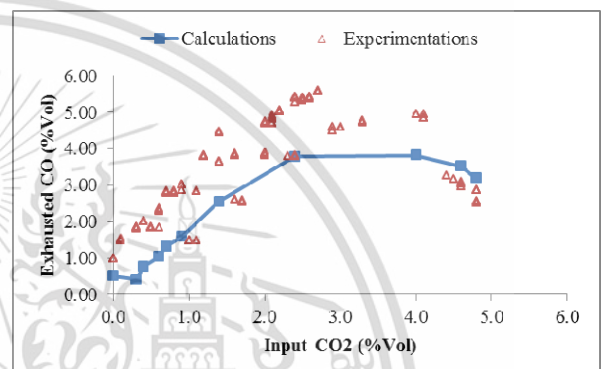
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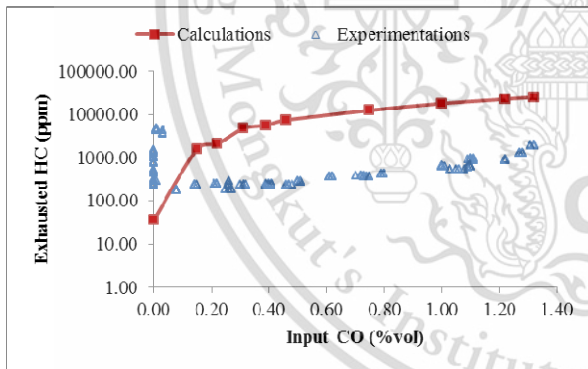
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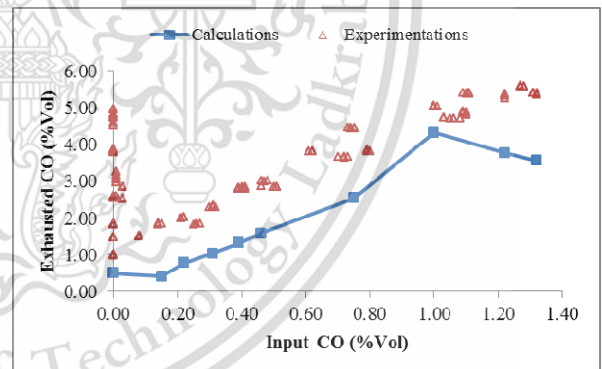
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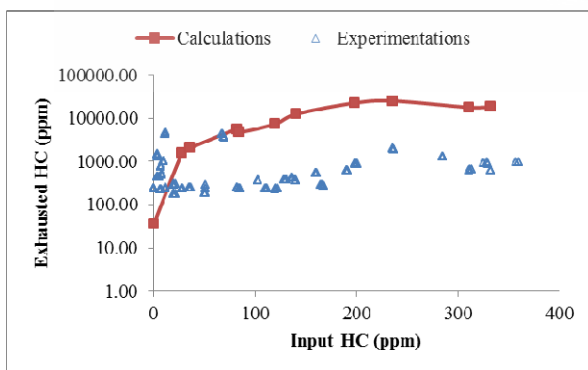
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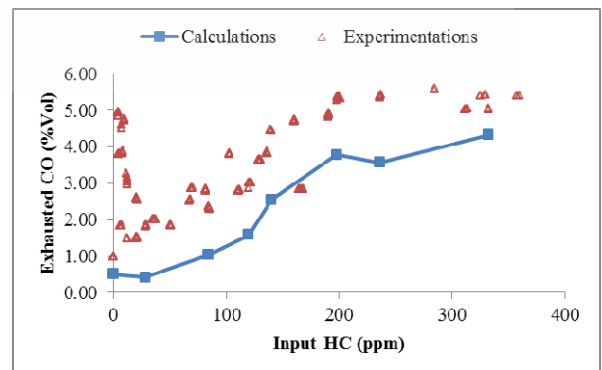
(c)



(c)



(d)

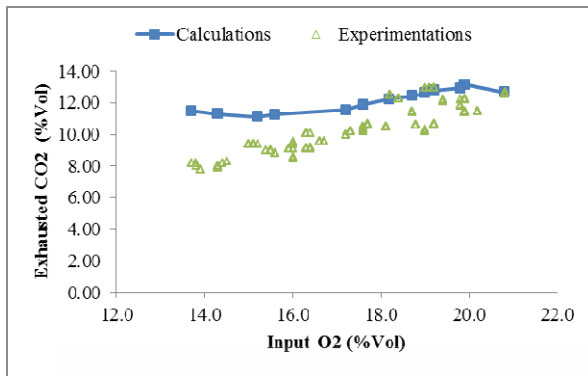


(d)

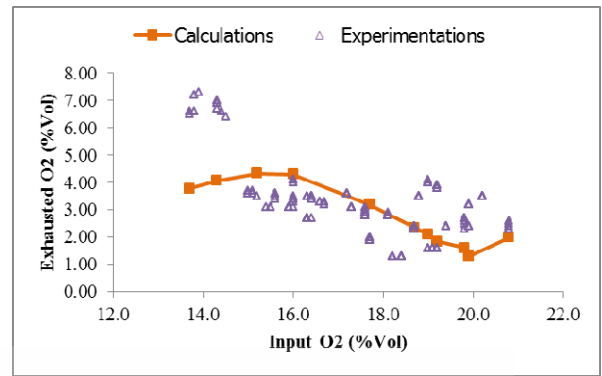
Fig. 3 Variation of Exhausted HC to Intake Elements

Fig. 4 Variation of Exhausted CO to Intake Elements

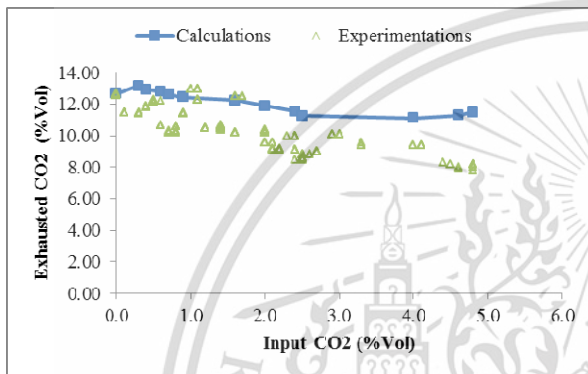
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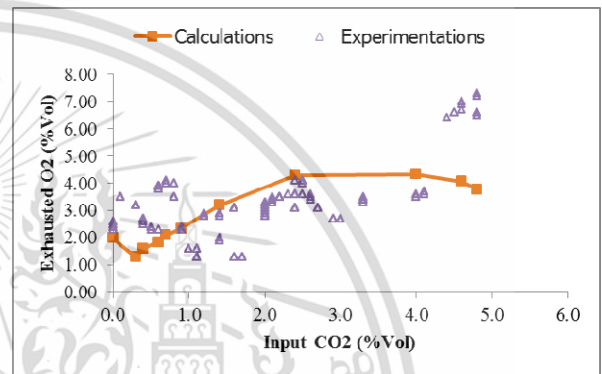
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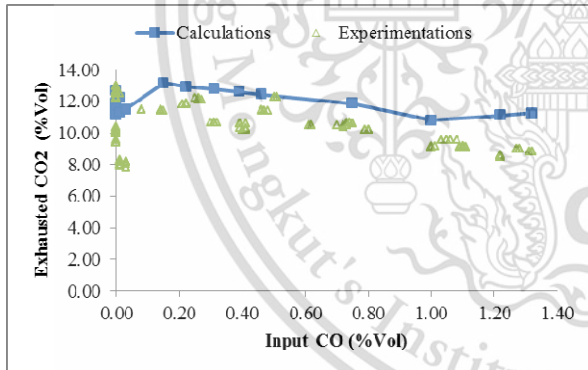
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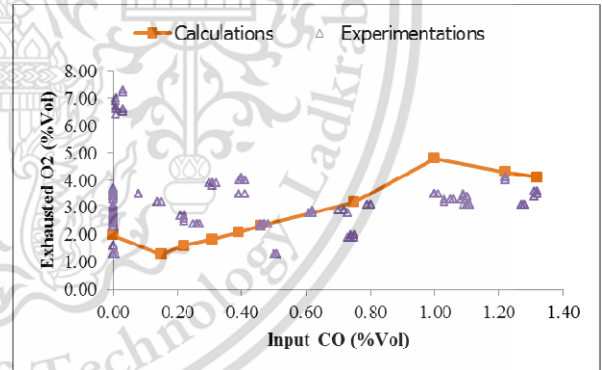
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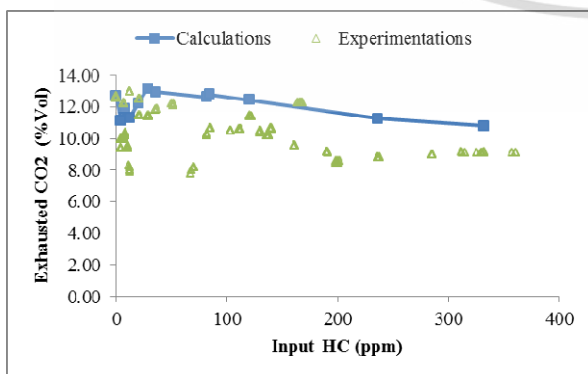
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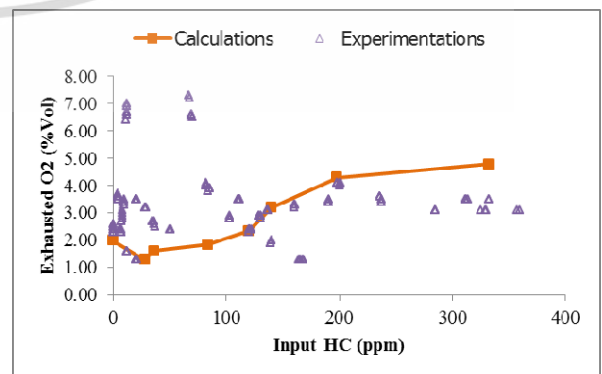
(c)



(c)



(d)



(d)

Fig. 5 Variation of Exhausted CO<sub>2</sub> to Intake Elements

Fig. 6 Variation of Exhausted O<sub>2</sub> to Intake Elements

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exhausted CO proportionally which means incomplete combustion of the engine. Such results are consistent to decrease of the intake O<sub>2</sub> which causes the richer air-fuel mixture; hence, there is not enough oxygen to oxidize all the carbon molecules sufficiently into CO<sub>2</sub>. The instability of engine are also shown in Figs 4(a)-4(b) as the concentrations of the exhausted CO are dropped while the concentrations of the intake O<sub>2</sub> and CO<sub>2</sub> are lower to 14% and higher than 5%, respectively. The calculated values seemed to provide good predictive tendency between the exhausted CO and each intake. However, in all of the tested range, the actual concentrations of the emission are all higher than the calculation values.

As the rise of the exhausted HC and CO means incomplete combustion, the concentration of the exhausted CO<sub>2</sub> imply to completeness of the combustion. Figs. 5(a)-5(d) describe the variation of the exhausted CO<sub>2</sub> to each intake polluted air element. The exhausted CO<sub>2</sub> tends to decrease continuously as the concentrations of the intake CO<sub>2</sub>, CO, and HC are increased, while such variation is inversely proportion to the increase of the intake O<sub>2</sub>. The calculation values are seemed to provide good predictive tendency between the exhausted CO<sub>2</sub> and each intake. However, in all of the tested range, the actual concentrations of CO<sub>2</sub> are all lower than the calculation values. The instability of engine are also shown in Figs 4(a)-4(b) as the concentrations of the exhausted CO are dropped while the concentrations of the intake O<sub>2</sub> and CO<sub>2</sub> are lower to 14% and higher than 5%, respectively. The calculated values seemed to provide good predictive tendency between the exhausted CO and each intake. However, in all of the tested range, the actual concentrations of the emission are all higher than the calculation values.

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Figs. 6(a)-6(d) illustrate the variation of the excess O<sub>2</sub> in the exhaust gas to each intake polluted air element O<sub>2</sub>, CO<sub>2</sub>, CO, and HC, respectively. The experimental results show that poor correlations are found between the excess O<sub>2</sub> and the varying concentrations of the intake elements as well as the available calculation values.

As the amount of the intake air decreases, the combustion temperature is also decreased. Fig. 7 shows the exhaust gas temperature drop as the concentration of the intake O<sub>2</sub> is decreasing linearly.

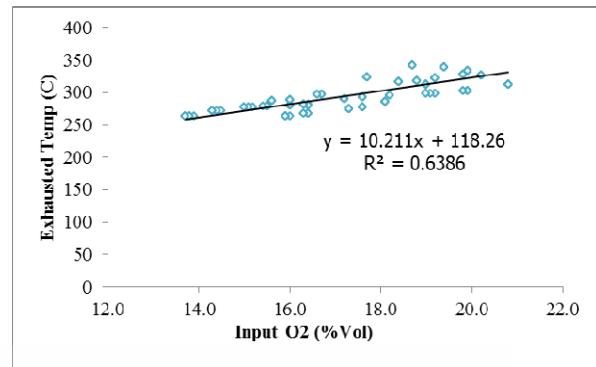


Fig. 7 Variation of Exhaust Gas Temperature to the Intake Oxygen

### 4.2 Correlation of Exhaust Gas Composition

Comparisons of the experimental results to the available correlation of the exhausted O<sub>2</sub> and CO<sub>2</sub>, and the exhausted CO and CO<sub>2</sub> from the prior study in normal fresh air intake [4] are demonstrated as shown in Figs. 8-9. The correlations can be approximated to a linear equation as the described relations in the figure.

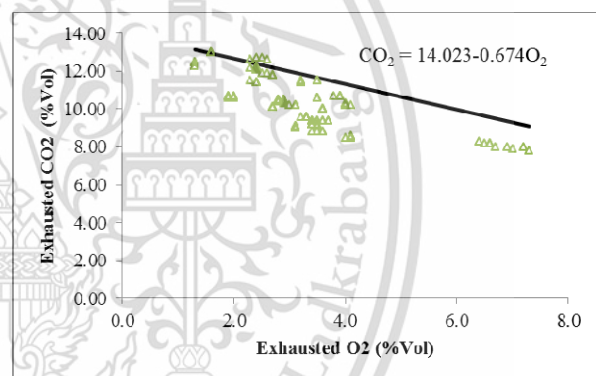


Fig. 8 Comparison of the Experimental Data to the Correlation of the Exhausted O<sub>2</sub> and the Exhausted CO<sub>2</sub>

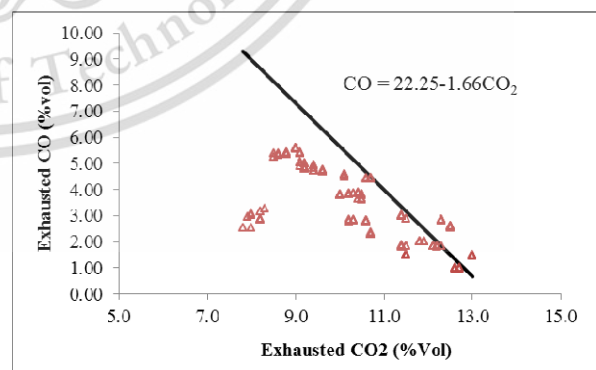


Fig. 9 Comparison of the Experimental Data to the Correlation of the Exhausted CO<sub>2</sub> and the Exhausted CO

In Fig. 8, for normal fresh air intake, the concentration of the exhausted CO<sub>2</sub> decreases linearly as the exhausted O<sub>2</sub> increases. For the polluted air intake, such relation is also consistent but the decreasing trend of the exhausted CO<sub>2</sub> is lower than

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the normal fresh air intake. This means the completeness of combustion is lower than the fresh air intake.

The amount of CO+CO<sub>2</sub> in the exhaust gas implies total carbon to be burned in the engine. In Fig. 9, for normal fresh air intake, the concentration of the exhausted CO decreases as the exhausted CO<sub>2</sub> increases. For the polluted air intake, such relation is also consistent but the decreasing trend of the exhausted CO is lower than the normal fresh air intake as evident in the experimental results that high concentration of HC is observed.

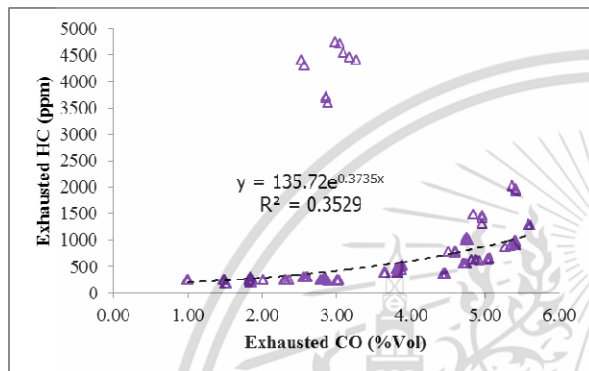


Fig. 10 Correlation of the Exhausted Carbon Monoxide and the Exhausted HC

Fig. 10 demonstrates that the concentration of the exhausted CO and the exhausted HC has no obvious relation which is consistent to the prior study.

### 5. Conclusion

The study of the exhaust gas composition from the combustion in a gasoline engine with various proportion of the polluted air intake was diluted upstream to the engine at the idle speed. The exhaust gas composition and temperature were sampled and analyzed through the exhaust gas analyzer in order to determine the concentrations of CO, CO<sub>2</sub>, HC, and excess oxygen O<sub>2</sub>. Experimentation results was used to investigate the effects of each intake pollutant element to the exhaust gases emission and compare to the established mathematical model and the prior studies, therefore, it could be summarized that

All the pollutants diluted into the intake air causes the increase of HC and CO concentrations in the exhaust gas composition while the CO<sub>2</sub> concentration decreased due to incompleteness of the combustion in the engine. Experimental results showed that the variation of the excess O<sub>2</sub> in the exhaust gas did not reveal any relation to the varying concentrations of the diluted pollutants.

The exhaust gas temperature decreased as the fresh air intake was diluted by the pollutants due to incomplete combustion as mentioned above.

The established mathematical model was able to be used only as a predictor to the trend of the concentration of CO and CO<sub>2</sub> in the exhaust gas composition, but prediction of the HC concentration in

the exhaust gas did not provide proper results due to extremely different values to the experimental results. However, the calculations of the CO concentration were usually lower than the experimental results, and the calculations of the CO<sub>2</sub> concentration were usually higher than the experimental results. Because of scattering in the experimental data of the exhausted O<sub>2</sub> concentration, the model may not provide the results consistent to the experimental results accurately.

The exhausted CO<sub>2</sub> and O<sub>2</sub>, and the exhausted CO and CO<sub>2</sub> had correlations consistent to the prior study, while the exhausted CO and HC shown poor relation. Therefore, separately measured of the exhaust gas composition should be conducted to ensure the variability of the emissions.

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