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Babylon University
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Department of Mechanical Engineering

Thermal Design of a Four-Stroke, Spark Ignition Engine

A Thesis

Submitted to the College of Engineering of the University of Babylon in Partial Fulfillment of the Requirements for the Degree of Master of Science in Mechanical Engineering

By

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بسم الله الرحمن الرحيم

يؤتي الحكمة من يشاء ومن يؤت

الحكمة فقد أوتي خيرا كثيرا وما

يذكر إلا أولو الألباب

صدق الله العظيم

سورة البقرة ٢٩٦

الإهداء

إلى...روح والدي الطاهرة احتراما لذكراه

إلى...ينبوع الحنان الدافئ...والدتي

CERTIFICATION

We certify that this thesis entitled ***"Thermal design of a four stroke spark ignition engine "*** was prepared by *"**Laith Hassan Jawad"*** under our supervision at **Babylon University**, College of Engineering in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

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(In The Name of Allah ,The Gracious and Merciful)

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Liath Hassan Jawad
٢٠٠٢

ABSTRACT

Models can be of great assistance to the engine designers if they give a good representation of the engine system. Their economic value is in the reduction in time and costs for the development of new engines and their technical value is in the identification of areas which require specific attention as the design study evolves.

The cycle analysis (engine simulation) is a useful tool for the following reasons:

١.It provides a better understating of the variables involved and their effect on engine performance.

٣.It is a useful tool in optimizing an engine design for a particular application.

The cycle analysis gives a useful relationship among the performance parameters, compression ratio, fuel-air ratio, and of fuel used.

The mathematical model was put to study engine performance. This included the study of the cylinder maximum temperature, pressure and spark timing. The mathematical model deals with ٤-stroke, spark ignition naturally aspirated engine. Numerical methods can be used to solve some equation, such as Newton-Raphson and Runge-Kutta. A computers program in Quick Basic was constructed and developed to carry out the solution.

The engine performance studied for different operating parameters, such as mean effective pressure, torque, power, specific fuel consumption, thermal efficiency and found that optimum values for this variables at fuel-air ratio equal one.

Study the effect of dissociation of component on engine performance, and this very important to the emission and pollution for study later.

Study the effect of fuel-air ratio on flame speed , and found that the maximum flame speed for fuel-air slightly rich mixture.

Variation in initial pressure , mixture strength have only modest effect on thermal efficiency , The effects of variations in these variables on man effective pressure (mep) are more substantial , however , because mep depends directly on the initial charge density .

List of contents

Latin symbols		
symbol	Description	unit
n	Carbon atoms in fuel	
m	Hydrogen atoms in fuel	
(F/A)	Fuel to air ratio	Kg_f/kg_a
a	Mole of fuel	Mole
N	Total mole of mixture or products	Mole
P	Pressure or power	N/m^2 or kW
V	Volume	M^3
R_{mol}	Universal gas constant	kJ/kg.mol.K
T	Temperature	K˙
n_i	NO. of moles of species i	Mole
X_i	Mole fraction of species i	/
U	Specific internal energy	kJ /kg
H	Specific enthalpy	kJ / kg
U	Total internal energy	kJ
H	Total enthalpy	kJ
Z_{ij}	Polynomial coefficient for species i	
Q_V	Heat of reaction at constant volume	kJ/kg
C_P	Specific heat at constant presure	kJ/kg.K
C_V	Specific heat at constant volume	kJ/kg.K
w	Work	kJ/kg
k	Ratio of Specific heat at constant pressure to Specific heat at constant volume	
K_P	Equilibrium constant	
CR	Compression ratio	
R_{bs}	Ratio of cylinder bore to piston stroke	
D	Bore	m
L	Stroke	m
R	Ratio of connecting rod length to crank radius	
r	Crank radius	m
ℓ	Connecting rod length	m
A	Area	M^2
rps	Rotational speed of the crank shaft	Revolution per second
g	Gibbs free energy	kJ/kg
S	Entropy	kJ/kg.K
Re	Reynolds no.	
pr	Prandtl no.	
ν	Specific volume	M^3
x	Burned mass fraction	
S.F.C	Specific fuel consumption	gr/kW.hr

AFR	Air to fuel ratio	

Creek symbols		
Φ	Equivalence ratio	
γ	Atoms of oxygen in fuel	
α	Atoms of nitrogen in fuel	
f	Residual mass fraction	
ℓ	Connecting rod length	m
θ	Crank angle	degree
r	Specific volume	M^3/kg
μ_g	Dynamic viscosity	Kg/sec.m
π	Constant	٣.١٤
ω	Angular velocity	Rad/sec
η	Efficiency	
η_V	Volumetric efficiency	

Subscripts	
f	Fuel
a	Air
am	Ambient
i	Species or indicated
m	Mechanical
o	At absolute zero
p	Products or piston
R	Reactants
St	Stoichiometric
T	Total
j	Loop from ١ to ٥
v	Constant volume
١	Initial step
٢	Final step
i+١	Next STEP
d	Displacement
c	Clearance
ch	Cylinder head
u	Unburned
b	Burned or brake
ad	Adiabatic
g	Gas
s	Surface

تصميم محرك يعمل بالقدح رباعي الشوط من الناحية الحرارية

أطروحة

مقدمة إلى كلية الهندسة في جامعة بابل كجزء من متطلبات نيل
درجة ماجستير علوم في الهندسة الميكانيكية

قدمت من قبل
ليث حسن جواد المياحي

تشرين الأول ٢٠٠٢

CHAPTER 1

INTRODUCTION

١.١: General

The internal combustion engine is a heat engine that converts chemical energy into mechanical energy, which is available on a rotating output shaft. Chemical energy of the fuel is first converted to thermal energy by means of combustion or oxidation with air inside the engine. This thermal energy raises the temperature and pressure of the gases within the engine, and the high-pressure gas then expands against the mechanical mechanisms of the engine. This expansion is converted by the mechanical linkages of the engine to a rotating crankshaft, which is the output of the engine.the crankshaft, in turn's connected to a transmission and/or power train to transmit the mechanical energy to the desired final use There is always a need for higher power engines, and obtained the optimum engine design, therefore, the mathematical models is very signification to represent the engine and development this engine during short time.

١.٢: Theory of Combustion

The combustion process in a gaseous fuel-air mixture ignited by a spark is characterized by more or less rapid development of a flame which starts from the ignition point and spreads in a continuous manner outward from the ignition point. When this flame travels or spreads continuously to the end of the chamber without any sudden change in speed or shape, the combustion is called normal. But when the mixture appears to ignite and burn ahead of the flame, the phenomenon is called autoignition which may

lead to sudden increase in reaction rate and measurable pressure waves leading to what is called detonation.

In the normal combustion, there are two zone of combustion and the forward boundary of the reacting zone is called the flame front.

When a spark plug fires, the high voltage, discharge high energy ignites the air-fuel mixture between and near the electrodes. This creates a spherical flame front that propagates outward into the combustion chamber. In the initial stage, the flame front moves very slowly because of its small originated size. This is called the first stage of combustion,and the second stage is the propagation of flame front throughout the combustion chamber.

١.٣:Real Engine Cycle Events

All processes of the engine cycle are illustrated in a simple as shown Fig.(١.١).

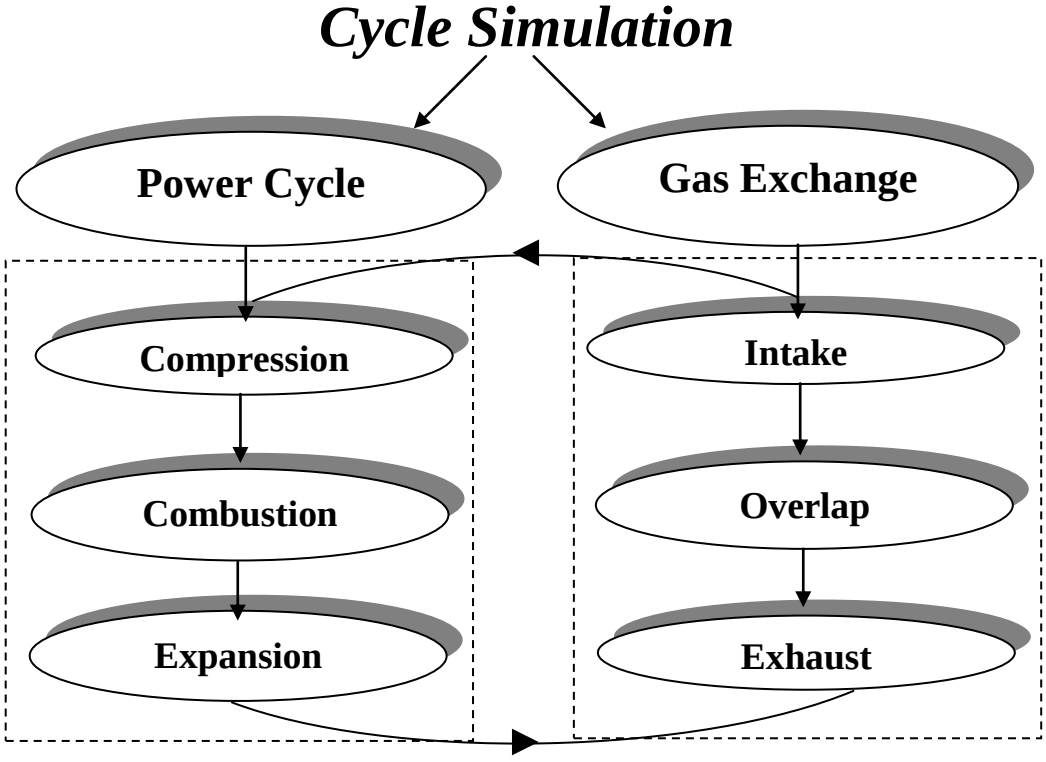


Fig.(١.١): represents the events of real cycle, ٤-stroke, S.I. engine

١.٤:Objective of the Present Work

The objective of this work is the modeling of a four stroke spark ignition engine. The two mathematical models can be derived and all

equations programmed using (Quick Basic) language.Two computer programs is to be constructed and developed to study the engine performance. The first program is related to the ideal engine design and the second program is related to the real time engine design.

The effects of many operating parameters on engine performance are included in the mathematical models. The models will be used to predict the following parameters: engine performance such as indicated mean effective pressure, indicated thermal efficiency, brake mean effective pressure, brake thermal efficiency, indicated and brake power output, indicated and brake specific fuel consumption, temperature and pressure distribution in cylinder and torque.

All computation to be done by using step by step evaluation based on numerical method, like Newton-Raphson and Runge-Kutta methods.

The aim of the present work lies in obtaining results in a short time.for preliminary work, it might not immediately predict the optimum engine design, but it will be study the performance and obtained the optimum engine parameters which is very necessary to study and design.

١.٥:Layout of the Thesis

This thesis falls into six chapters. Chapter one is an introduction and chapter two is concerned, with a brief literature review. Chapter three is devoted to the theoretical analysis that leads to the estimation of the engine performance.

Chapter four deals with the computational procedures where two computer programs are developed to perform all the calculations based on the theoretical analysis. Chapter five gives the presentation and discussion of the results. Chapter six represents the conclusions and recommendations from this work and introduces suggestions to develop this work in the future.

CHAPTER 2

LITERATURE REVIEW

٢.١: Introduction

The interest in this work is directed towards S.I. naturally aspirated engine. Hence the literature survey is directed toward S.I. naturally aspirated engine, and the effect of operating parameters on performance and engine design.

٢.٢: Review of the Engine Study

J. Mayo [١٩٧٥] studied the effect of engine design parameters on combustion rate in spark ignition engines, by calculation, shown that fast combustion rates tend to produce low NO_X levels. Their analysis has been partially confirmed by actual engine data on a highly turbulent fast burn engine. Thus , an experimental program was conducted to develop quantitative data on the effect of some of the engine design features that are known to have a significant influence on combustion rate. The effect of five engine design factors on combustion rate was determined. Burn time measured on ١٧ different engine configurations, each configuration containing different design levels of the five design factors. Burn time measurement was also made on a carbureted highly turbulent bowl-in piston fast burn engine with a centrally located spark plug. Significant reduction in burn time from a production combustion chamber was achieved by application of the design factors. The effect of emissions was also examined.

M.K. Gajendra Babu [١٩٧٥], studied a simulation and evaluation of a four stroke single cylinder spark ignition engine, of a mathematical model. This work has two parts. The first part describes the development of the mathematical model. The instruments that were developed for the evaluation of the model are included in the second part. The simulation results are found to agree well with the experimental data. In the expansion process, the equilibrium gas composition was found. The open period, comprising the exhaust and inlet processes, was solved by computing the mass flow rate across the valves, the properties at the mesh points of the intake and exhaust systems, and the temperature and pressure of the gas in the cylinder.

At the end of cycle, resistance networks for heat transfer were set up to estimate the metal temperature. The mechanical efficiency was found by estimating friction. In the evaluation of the model, the pressure time diagram for the gases in the cylinder, in the intake system, and in the exhaust system were measured in an instrumented engine.

R, S. Benson [١٩٧٥], studied a simulation model including intake and exhaust systems for a single cylinder four-stroke cycle spark ignition engine. The power cycle simulation requires only one empirical factor to correct for turbulent speed of the flame front in order to complete the cycle calculation including NO. Calculations are presented which compare well with previous work.

The model combines a full power cycle simulation with the prediction of NO emission with a comprehensive gas dynamic model for the cylinder and ducts allowing for chemical reactions in the exhaust pipe. Calculations are presented comparing the predicted NO with test results from a single cylinder engine. It is shown that good agreement is obtained between the predicted and measured NO over an equivalence range (٠.٨-

١.١) when the flame speed is corrected at the equivalence (٠.٩) corresponding to the peak NO.

S.D. Hires [١٩٧٨] studied the prediction of ignition delay and combustion intervals for a homogenous charge, spark ignition engine, corrections for the ignition delay and combustion energy release intervals in a homogeneous charge, spark ignited engine are developed. After incorporation within a simplified engine cycle simulation, predicted values of these two combustion's are based on four fundamental quantified, the turbulent integral scale, the turbulent micro scale; the turbulent intensity and the laminar flame speed.

Two empirical constants scale the correlation to a given engine. Predicted values for the ignition delay and burn intervals show good agreement with experimental results for wide variation in engine operating and design conditions (e.g.; engine speed and load, spark timing, EGR, air fuel ratio, and compression ratio) in addition, the shapes of the predicted mass fraction burned curves agree well with published data.

M. Metghalchi [١٩٨٠] studied the laminar burning velocity of propane air mixtures, and has been measured in the pressure range ٠.٤ to ٤٠)- atm and temperature range (٢٩٨-٧٥٠K) for equivalence ratios (٠.٨-١.٥). The measurements were made in a constant volume spherical combustion bomb, which could be heated, to ٥٠٠K. A thermodynamic analysis was used to calculate the laminar burning velocity from a pressure time history of the combustion process the measured values were correlated using both power law and experimental expressions.

A.C. Alkidas [١٩٨٠] studied the heat transfer characteristics of a spark ignition engine, transient heat flux measurements were obtained at

four position on the cylinder head of a four stroke single cylinder spark ignition engine. Tests were performed for both fired and motored operation of the engine. The primary engine operational variable was engine speed. The results showed that the heat flux varies considerably with position of measurement. At fired conditions, the initial high rate of increase of heat flux at each position of measurement correlated with calculated time arrival of the flame at that position.

Finally, as expected, the peak heat flux was found to increase with increased engine speed.

M. Metghalchi [١٩٨٢], studied the burning velocities of mixtures of air with methanol, isooctane, and indolene at high pressure and temperature, have been measured using the constant volume bomb method for fuel air equivalence ratios (٠.٨-١.٥) over the pressure and temperature ranges (٠.٤-٥٠) atm and (٢٩٨-٧٠٠). The effect of adding simulated combustion products to stoichiometric isooctane air mixture we also studied for diluent mass fractions (٠-٠.٢). Over the range studied, the results can be fit within ±١٠٪ by the functional form $S_U=S_{UO}(T_U/T_O)^{\alpha}(P/P_O)^{\varphi}$ (١-٢.١β),where S_{UO} depends on fuel type and equivalence ratio and α and φ depend only on equivalence ratio. In overlapping ranges, the results agree well with those previously reported.

G.C. Davis [١٩٨٢] studied the effect of in-cylinder flow processes (swirl, squish and turbulence intensity) on engine efficiency. The engine simulation model is a thermodynamic model coupled to sub models for the various physical processes of in cylinder swirl, squish and turbulence velocities that transfer and flame propagation.

The swirl and turbulence models are based on an integral formation of the angular momentum equation. This model account for the effects of

changes in geometry of the intake system and the chamber design on in cylinder flow processes. The combustion model is an entertainment burn up model applicable to the mixing controlled region of turbulence flame propagation. The flame is assumed to propagate spherically from one or two spark plug. A heat transfer model that is dependent upon the turbulence level is used to compute the heat loss from the unburned and burned gases. These sub models are calibrated from experimental data. A parameter study is conducted to determine the effects of in cylinder geometry on burn duration, heat transfer and fuel consumption. These results indicate that swirl; squish and turbulence intensity levels can be varied to produce a minimum in fuel consumption for the conditions examined.

G.G. Lucas [١٩٨٢] studied the effect of combustion chamber shape on the rate of combustion, flame propagation, the cylinder pressure for twenty on spark ignition engine combustion chamber design in spark ignition engine. These show that, for given compression ratio, the flame speed is not significantly affected by chamber design. However, the rate of combustion, the pressure history and cyclic dispersion are greatly affected. The burning velocity may be affected by combustion chamber design in two ways. First, increasing the charge to the turbulence by producing squish should also increase the burning velocity. Squish may be defined as the inward radial motion of the cylinder charge created by piston movement near TDC of the compression stroke.

F.T. Connolly and A.E.Yagle [١٩٩٣] investigated a new model relating cylinder combustion pressure to crank shaft angular velocity in an interval of combustion engine, primarily the fluctuations in velocity were near the cylinder firing frequency. Experimental results that confirm a

combustion pressure to angular velocity model of an IC.SI engine have been presented. This model was presented in Connolly and yagle (١٩٩٢,١٩٩٣) and developed and tested (via computer simulation and experimental measurements) in Connolly (١٩٩٢). In particular, the model relates a random sequence parametrizing cyclic combustion pressure variability to fluctuations in angular velocity.the model takes the form of alinear, first order differential equation relating continuous combustion pressure to continuous angular velocity wave forms, as well as relating a cyclic combustion pressure variability squence to samples of angular velocity at each combustion.

Paulina. S. Kuo [١٩٩٦] studied a cylinder pressure in a spark-ignition engine : a computational model. The project described in this article attempts to accurately predict the gas pressure changes within the cylinder of a spark-ignition engine using thermodynamic principles. The model takes into account the compression, combustion, and expansion that occur in the cylinder. Comparisons with actual pressure data show the model to have a high degree of accuracy. The model is further evaluated on its ability to predict the angle of spark firing and burn duration.

A.C. Alkidas [١٩٩٧] studies experimentally indicators of fuel maldistribution, several exhaust emissions were found to be good indicators for fuel maldistribution (both cylinder and cylinder to cylinder) in spark ignition engines.the quality of combustion greatly affected these indicators, thus possibly limiting their applicability. Corrections were developed to desensitize these indicators to the quality of combustion.

Y.M. Yacoub [١٩٩٨] studied the development and validation of a thermodynamic model for an S.I single cylinder engine, a multi zone quasi dimensional model that illustrates the intake, compression, combustion, expansion, and exhaust processes has been developed for a single cylinder four stroke spark ignition engine. The model takes into consideration mass and energy conservation in the engine cylinder, intake and exhaust plenums and crankcase plenum. The model calculates in stantaneous variations in gas thermodynamic states, gas properties, heat release rates .in cylinder turbulence, piston ring motion blow by, nitric oxide, and carbon monoxide formation. The cycle simulation accounts for induced gas velocities due to flame propagation in the turbulence model, which is applied separately to each gas one. This allows for the natural evolution of the averaged mean and turbulence velocities in burned and unburned gas regions. The present model prediction of thermal efficiency, indicated mean effective pressure, peak values of gas pressure, ignition delay, concentrations of nitric oxide, carbon monoxide and carbon dioxide are proven to be in agreement with experimental data.

Ivan. Arsie [١٩٩٨] studied the models for prediction of performance and emissions in a spark ignition engine a sequentially structured approach, a thermodynamic model for the simulation of performance and emissions in a spark ignition engine is presented. The model is part of an integrated system of models with a hierarchical structure developed for the study and the optimal design of engine control strategies.

The main thermodynamic model is based on the classical two-zone approach. A multi zone model is then derived from the two zone calculation, for proper evaluation of temperature gradients in the burned gas region . The emissions of HC, CO, and NO_X are the predicted by three sub models.

The results of the thermodynamic cycle model validation, performed over more than ٣٠٠ engine operating conditions, show a satisfactory level of agreement between measured and predicted data cycles.

Morea [٢٠٠٠] studied a multi one thermodynamic and quasi-one dimensional hybrid models applied to four-stroke spark ignition engine modeling. Model is capable to predict torque, power, specific fuel consumption, volumetric efficiency, etc; as well as instantaneous pressure, temperature, etc; within cylinder and engine manifolds. Inside the cylinder, flow is described by means of the conservation of mass principle and the first law of thermodynamics. A first order ordinary differential equation system is obtained and solved using a numerical methods. Flow in manifolds is characterized by means of the fundamental conservation equations: mass, momentum and energy. Two type mixtures are considered: fuel air mixtures and combustion products. Special attention is dedicated to real gas behavior due to specific heats are function of temperature.

Summary : -

Most of the literature published presented modeling of I.C.E. Some depend on experimental data.

The areas of interest are the study of the influence of operating parameters on engine performance.

A complete and comprehensive study is still needed to take effect of optimum operating condition on engine performance.

CHAPTER 3

THEORETICAL ANALYSIS

٣.١: General

The spark ignition engine works on the Otto cycle .In an ideal cycle a mixture of fuel and air is compressed, ignited and the combustion takes place at constant volume .The products of combustion expand and at the end of expansion stroke the exhaust is at constant volume. In practice this ideal cycle is modified due to **(a)** heat losses in the compression, combustion and expansion periods, and **(b)** the finite time for the combustion processes. In the first part of this chapter the ideal cycle will be examined and in the second part the real cycle .The analysis will start directly with the chemical reactions and the corresponding thermodynamic properties of the substances.

The majority of reciprocating engines operate on what is known as the four-stroke cycle. Each cylinder requires four strokes of its piston two revolutions of the crankshaft to complete the sequence of events, which produces one power stroke.

The sequence of events of fuel-air cycle as shown in Fig (٣.١) is as follows:

١- Intake stroke; which starts with the piston at TDC and ends with the piston at BDC, in which it withdraws fresh mixture into the cylinder. To increase the mass induced; the inlet valve opens shortly before TDC and closes after BDC.

٢-**C**ompression stroke; when both valves are closed and the mixture inside the cylinder is compressed by the piston moving from BDC to TDC to small fraction of its initial volume. Towards the end of the compression stroke, combustion is initiated and the cylinder pressure rises rapidly.

٣-**P**ower stroke; or expansion stroke, which starts with the piston at TDC and ends at BDC as the high-temperature, high-pressure, gases push the piston down and force the crank to rotate. During the power stroke as the piston had to do during compression. As the piston approaches BDC the exhaust valve opens to initiate the exhaust process and lowers the cylinder pressure close to exhaust pressure.

٤-**E**xhaust stroke; the remaining burned gases exit the cylinder: as the cylinder pressure is substantially higher than the exhaust pressure; they are swept out by the piston moving towards TDC. As the piston approaches TDC the inlet valve opens just after the exhaust valve closes, and the cycle starts again. The four-stroke cycle require, for each engine cylinder, two-crankshaft revolution for each power stroke.

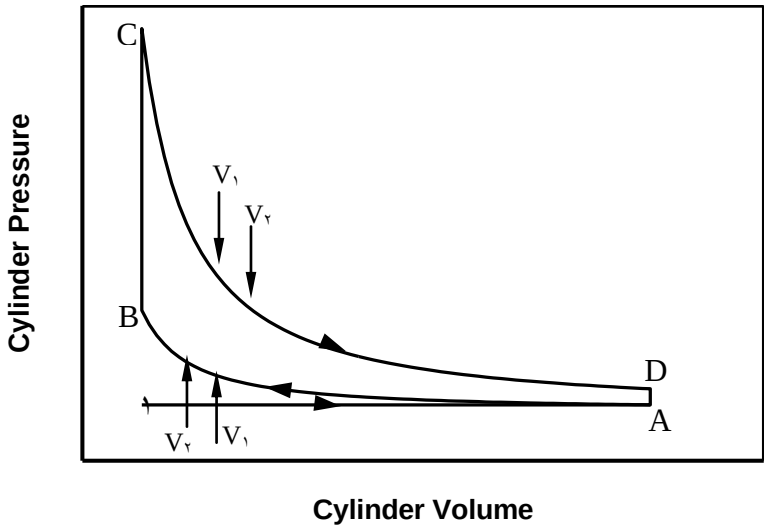


Fig. (٣.١): Ideal (Otto Cycle).

٣.٢: Thermo Chemistry of Fuel –Air Mixture

When assessing the performance of any engine, the most important processes involved are those taking place in the engine cylinder. That is, the compression, combustion and expansion processes.

The fuel is assumed to be formed of carbon and hydrogen only. To facilitate calculation an equivalent hydrocarbon CnHm will be set to represent the fuel. Where **n** is the carbon atoms in the fuel, and if **m** is the hydrogen atoms in the fuel.

The basic stoichiometric chemical representation for a hydrocarbon-oxygen reaction is,

$$C_nH_m+(n+٠.٢٥m)\ O_٢=nCO_٢+٠.٥mH_٢O \qquad (٣.١)$$

The ratio of number of moles of $O_٢$ to the number of moles of $N_٢$ in air is given by;

$$O_٢+٧٩/٢١N_٢=O_٢+٣.٧٦N_٢ \qquad (٣.٢)$$

Hence the basic stoichiometric equation for hydrocarbon-air reaction is;

$$C_nH_m+(n+٠.٢٥m)O_٢+٣.٧٦(n+٠.٢٥m)N_٢=nCO_٢+٠.٥mH_٢O+٣.٧٦(n+٠.٢٥m)N_٢ \qquad (٣.٣)$$

The stoichiometric equation defines the correct (ideal) mixture only. To allow for mixtures different from the correct mixture the equivalence ratio ϕ is introduced this is defined as the ratio of the actual fuel to air ratio (F/A) to the stoichiometric fuel to air ratio $(F/A)_{St}$. Hence for a mixture of equivalence ratio ϕ, we have:-

$$C_nH_m+(n+٠.٢٥m)\ \frac{1}{\phi}\ O_٢+٣.٧٦(n+٠.٢٥m)\ \frac{1}{\phi}\ N_٢ \qquad (٣.٤)$$

If the equivalence ratio ϕ is greater than unity, the mixture is said to be rich. If it is less than unity the mixture is said to be weak or lean. Spark ignition engines normally run with rich or weak mixtures; while compression ignition engines normally run with weak mixture only.

The hydrocarbon-air mixture at point A in the cycle Fig.(٣.١)can be defined as follows: -

For a moles of CnHm the mixture will be

$$a(C_nH_m+(n+٠.٢٥m)O_٢\frac{1}{\phi}+٣.٧٦(n+٠.٢٥m)N_٢\frac{1}{\phi}$$

(٣.٥)

And the total number of moles N_A of the mixture is

$$N_A=a\ (١+٤.٧٦(n+٠.٢٥m)\ \frac{1}{\phi}\ ) \qquad (٣.٦)$$

The magnitude of a is obtain from the ideal gas equation for the mixture;

$$a=\frac{P_AV_A}{N_AR_{mol}T_A} \qquad (٣.٧)$$

The cycle calculation is simply carried out by dividing the cylinder volume to a number of steps. For each step the first law of thermodynamics is solved .In the state between ١ and ٢ for the combustion and expansion stroke a second set of equations is required to represent thermal equilibrium for the chemical reaction; these are the dissociation equations.

The numerical solutions are based on the Newton-Raphson method for each step.

The chemical reaction equation with dissociation is as follows: -

$$a(C_nH_m+(n+٠.٢٥m)\frac{1}{\phi}O_٢+٣.٧٦(n+٠.٢٥m)\frac{1}{\phi}N_٢)=n_١CO_٢+n_٢CO+n_٣H_٢O+$$

$$n_٤H_٢+n_٥O_٢+n_٦N_٢ \qquad (٣.٨)$$

If x is the mole fraction, then

$$X_1=\frac{n_1}{N_T},X_2=\frac{n_2}{N_T},\ X_3=\frac{n_3}{N_T},\ X_4=\frac{n_4}{N_T},\ X_5=\frac{n_5}{N_T},\ X_6=\frac{n_6}{N_T}, \qquad (٣.٩)$$

The total number of moles N_T is

$$N_T=n_١+\ n_٢+\ n_٣+\ n_٤+\ n_٥+\ n_٦ \qquad (٣.١٠)$$

The specific energy of the mixture “u” is given by

$u=x_1u_1+ x_2u_2+ x_3u_3+ x_4u_4+ x_5u_5+ x_6u_6$ (٣.١١)

Or

$$u = \sum x_i u_i \text{ , Where } i=1 \text{ to } 6 \quad (٣.١٢)$$

And the specific enthalpy of the mixture "**h**" is given by

$$h = \sum x_i h_i \quad (٣.١٣)$$

And for system at rest,

$$h - u = \sum x_i (h_i - u_i) \quad (٣.١٤)$$

Or

$$h - u = \sum x_i (R_{mol} T) = R_{mol} T \quad (٣.١٥)$$

Or

$h=u+R_{mol}T$ for the mixture (٣.١٦)

The total internal energy U is: -

$$U = N_T u = N_T \sum x_i u_i = \sum n_i u_i \quad (٣.١٧)$$

And,

$H=U+R_{mol}T$ (٣.١٨)

Where $\mathbf{n_i}$ equals the number of moles of gas **i** in the mixture.

$u=u_o+u(T)$ (٣.١٩)

And the specific enthalpy as

$h=h_o+h(T)$ (٣.٢٠)

Where $\mathbf{u_o}$ and $\mathbf{h_o}$ are the specific internal energy and specific enthalpy at the absolute zero, respectively.

Hence,

$h_o=u_o$ (٣.٢١)

The specific enthalpy **h(T)** of a gas of species i is defined as $\mathbf{h_i(T)}$ and the corresponding internal energy as $\mathbf{u_i(T)}$. The numerical value of $\mathbf{h_i(T)}$ is given by

$h_i(T)=R_{mol}(Z_{i1}T+ Z_{i2}T^2+ Z_{i3}T^3+ Z_{i4}T^4+ Z_{i5}T^5)$

(٣.٢٢)

Where Z_{i1}, Z_{i2}, Z_{i3}, Z_{i4}, and Z_{i5} are the polynomial of coefficients for the species i, or in general it can be written as Z_{ij}(j=١ to ٥),the numerical value of $h_i(T)$ is then

$$h_i(T) = R_{mol}\sum_{i=1}^{5} Z_{ij}T^j \qquad (٣.٢٣)$$

The corresponding value of $\mathbf{u_i(T)}$ will be

$u_i(T)= h_i(T)-R_{mol}T$ (٣.٢٤)

And

$$u_i(T) = R_{mol}\left[\sum_{i=1}^{5} Z_{ij}T^j - T\right] \qquad (٣.٢٥)$$

The polynomial coefficients $\mathbf{Z_{ij}}$ **(j=١ to ٥),** are given in appendix A for a number of gases.

The absolute specific enthalpy h_i is given by: -

$h_i=h_i(T)+h_{oi}$

(٣.٢٦)

Where h_{oi} is the enthalpy at absolute zero of species; and it's coefficient are given in appendix A as **(Z_{i6}).**

The absolute specific internal energy $\mathbf{u_i}$ is given by: -

$u_i(T)+u_{oi}$ (٣.٢٧)

Where $\mathbf{u_{oi}}$ is the internal energy at absolute zero. This is equal to $\mathbf{h_{oi}}$.

For a mixture of gases

$U=U_o+U(T)$ (٣.٢٨)

$H=H_o+H(T)$ (٣.٢٩)

Where: -

$$H_o = U_o = N_T\sum x_i u_{oi} = \sum n_i u_{oi} \qquad (٣.٣٠)$$

And,

$$U(T) = N_T\sum x_i u_i(T) = \sum n_i u_i(T) \qquad (٣.٣١)$$

$$H(T) = N_T \sum x_i h_i(T) = \sum n_i h_i(T) \qquad (٣.٣٢)$$

٣.٣: Heat of Reaction or Calorific Values.

This is calculated as follows: -

In a constant volume calorimeter, the combustion process of fuel air mixture at constant volume and the reaction takes place in a container surrounded by a water jacket. The initial temperature of the reaction is T_S. After combustion the temperatures of products are cooled to the same temperature as the reactions. The heat lost by the gases in the system equals the heat of reaction of the fuel plus air at constant volume.

For this case from the first law

$$Q=U_P-U_R \qquad (٣.٣٣)$$

Q is equal to $\mathbf{Q_V}$ which is the heat of reaction at constant volume

(Where $\mathbf{Q_V}$ is negative for an exothermic reaction).

$$Q_V=U_P-U_R \qquad (٣.٣٤)$$

So,

$$U_P=N_P(u_p(T)+u_{op}) \qquad (٣.٣٥)$$

And,

$$U_R=N_R(u_R(T)+u_{oR}) \qquad (٣.٣٦)$$

Where,

$$u_P(T) = \sum (x_i u_i(T))_P \quad , \qquad (٣.٣٧)$$

$$\mathbf{u_{oP} = \sum (x_i u_{oi})_P} \quad , \qquad (٣.٣٨)$$

$$u_R(T) = \sum (x_i u_i(T))_R \quad \text{,and} \qquad (٣.٣٩)$$

$$\mathbf{u_{oR} = \sum (x_i u_{oi})_R} \qquad (٣.٤٠)$$

And the heat of reaction, $\mathbf{Q_V}$, is: -

$$Q_V=N_P u_p(T)-N_R u_R(T)+ \Delta U_o \qquad (٣.٤١)$$

$\Delta U_o = N_P u_{op} - N_R u_{oR}$ (٣.٤٢)

٣.٤: Cycle Processes

In cycle processes the cylinder volume can be divided into a number of steps and the calculation is done step by step.

٣.٤.١: Compression Stroke

This corresponds to the period from **A** to **B** on the pressure volume diagram Fig.(٣.١).

Consider a small change in volume from $\mathbf{V_١}$ to $\mathbf{V_٢}$ the first law of thermodynamic for the change

$dQ - dW = dU$ (٣.٤٣)

And since the process is adiabatic, **dQ=٠**, we have

$dU + dW = ٠$ (٣.٤٤)

For a charge in pressure from $\mathbf{P_١}$ to $\mathbf{P_٢}$ the work **dW** is Approximately: -

$dW = PdV = ٠.٥(P_١ + P_٢)(V_٢ - V_١)$ (٣.٤٥)

The internal energy change is dU, which is given by

$dU = U_٢ - U_١$ (٣.٤٦)

The internal energy of the mixture is a function of the composition of cylinder contents and temperature. The internal energy is therefore the internal energy of the reactants in the hydrocarbon-air combustion process. The composition is considered to be constant during the compression stroke.

If $\mathbf{U_R}$ is the internal energy of the reactants, then from equation (٣.٢٨)

$U_R = U_{oR} + U_R(T)$ (٣.٤٧)

If $\mathbf{(u_o)_{CnHm}}$, $\mathbf{(u_o)_{O٢}}$, $\mathbf{(u_o)_{N٢}}$ are the specific internal energies at absolute zero for $\mathbf{C_nH_m}$, $\mathbf{O_٢}$ and $\mathbf{N_٢}$, then the internal energy for the reactants at absolute zero U_{oR} is

$$U_{oR}=a((u_o)_{CnHm}+(n+٠.٢٥m)\ \frac{1}{\phi}\ (u_o)_{O٢}+٣.٧٦(n+٠.٢٥m)\ \frac{1}{\phi}\ (u_o)_{N٢})$$

(٣.٤٨)

From equation (٣.٢١) $\mathbf{h_o=u_o}$ and $\mathbf{U_{OR}}$ can be calculated from the coefficient $\mathbf{Z_{i٧}}$ as in appendix A.

If the specific internal energies, **u(T)** are $\mathbf{u(T)_{CnHn}}$, $\mathbf{u(T)_{O٢}}$ and $\mathbf{u(T)_{N٢}}$, for $\mathbf{C_nH_m}$,$\mathbf{O_٢}$ and $\mathbf{N_٢}$,the $\mathbf{U_R(T)}$ for the mixture is given by

$$U_R(T)=a(u(T)_{CnHm}+(n+٠.٢٥m)\ \frac{1}{\phi}\ u(T)_{O٢}+٣.٧٦(n+٠.٢٥m)\ \frac{1}{\phi}\ u(T)_{N٢})$$

(٣.٤٩)

The internal energy $\mathbf{U_R(T)}$ can be calculated from the polynomial coefficient given in table (٣.١). The internal energy change $\mathbf{U_٢-U_١}$ is then: -

$U_٢-U_١=(U_R)_٢-(U_R)_١=(U_{oR})_٢-(U_{oR})_١+(U_R(T_٢)-(U_R(T_١)$

Or

$$U_٢-U_١= (U_R(T_٢)-(U_R(T_١) \qquad (٣.٥٠)$$

Since for a mixture of constant composition: -

$(U_{oR})_٢=(U_{oR})_١$

Since in each step a change in state from $\mathbf{V_١}$ to $\mathbf{V_٢}$ then the first law is: -

$$U_٢-U_١+dW=٠ \qquad (٣.٥١)$$

Substituting for $\mathbf{U_٢-U_١}$ and **dW** into equation?

$$(U_R(T_٢)-(U_R(T_١)+ ٠.٥(P_١+P_٢)(V_٢-V_١)=٠ \qquad (٣.٥٢)$$

The internal energy term $U_R(T_٢)$ depends on $T_٢$, therefore in equation (٣.٥٢) there are two unknowns $P_٢$ and $T_٢$; $P_١$, $T_١$ and $U_R(T_١)$ are known at beginning of the initial step. These are related by the equation of State.

$$P_٢=P_١(V_١/V_٢)(T_٢/T_١) \qquad (٣.٥٣)$$

To solve for $P_٢$ and $T_٢$ we use the Newton – Raphson method as follows:

$$f(T_٢)= (U_R(T_٢)-(U_R(T_١)+ ٠.٥(P_١+P_٢)(V_٢-V_١)=٠ \qquad (٣.٥٤)$$

And

$$\frac{df(T_2)}{dT_2} = f'(T_2) = \frac{dU_R(T_2)}{dT_2} = N_R C_V(T_2) \qquad (٣.٥٥)$$

Since from $C_V = (\frac{\partial u}{\partial T})_V$ then

$$\frac{dU_R(T_2)}{dT_2} = N_R C_V(T_2) \qquad (٣.٥٦)$$

And $$\frac{dU_R(T_1)}{dT_2} = 0$$

In the expression (٣.٥٥) it was assumed that the rate of change of work dW with temperature over the volume change of small step is negligible and;

$$(T_2)_{i+1} = (T_2)_i - \frac{f(T_2)_i}{f'(T_2)_i}$$

Or

$$(T_2)_{i+1} = (T_2)_i - \frac{f(T_2)_i}{N_R\, C_V(T_2)_i} \qquad (٣.٥٧)$$

The first estimate of $\mathbf{T_2}$ can be obtained from the expression as the initial condition,

$$T_2 = T_1(\frac{V_1}{V_2})^{k-1} = T_1(\frac{V_1}{V_2})^{\frac{R_{mol}}{C_V(T_1)}} \qquad (٣.٥٨)$$

Where $k = \frac{C_P(T_1)}{C_V(T_1)}$

And the first estimated value of $\mathbf{P_2}$ is obtain from equation (٣.٥٣), where subscripts ١ and ٢ refers to initial and final states of a small step.

٣.٤.٢: *The Combustion Process*

This period corresponds to BC in the pressure volume diagram of Fig.(٣.١). In this analysis the products of combustion are assumed to be carbon dioxide **(CO_2)**, carbon monoxide **(CO)**, water vapor **(H_2O)**,

hydrogen **(H_2)**, oxygen **(O_2)** and nitrogen **(N_2)**. A more comprehensive list of products will be examined later in section ().

For the combustion of one mole of $\mathbf{C_nH_m}$ in air:

$$n_1CO_2+n_2CO+n_3H_2O+n_4H_2+n_5O_2+n_6N_2 , \qquad (٣.٥٩)$$

Where $\mathbf{n_1}$, $\mathbf{n_2}$, $\mathbf{n_3}$, $\mathbf{n_4}$, $\mathbf{n_5}$ and $\mathbf{n_6}$ are the number of moles of $\mathbf{CO_2}$, **CO**, $\mathbf{H_2O}$, $\mathbf{H_2}$, $\mathbf{O_2}$ and $\mathbf{N_2}$ per mole of $\mathbf{C_nH_m}$ respectively.

The general chemical reaction for the combustion process from **B** to **C** will then be of the form

$$a(C_nH_mO_\gamma N_\alpha \ +(n+0.25m-0.5\gamma)\frac{1}{\phi}O_2+3.76(n+0.25m-0.5\gamma)\frac{1}{\phi}N_2)=a$$

$$(n_1CO_2+n_2CO+n_3H_2O+n_4H_2+n_5O_2+n_6N_2) \qquad (٣.٦٠)$$

To determine the number of mole n_1 to n_6 and considering dissociation reactions, in the present case there are two such reactions:

$$CO + H_2O \Leftrightarrow H_2 + CO_2$$

And

$$CO + \frac{1}{2}O_2 \Leftrightarrow CO_2$$

The first reaction is the water-gas reaction: this states that in the equilibrium state the rate of formation of hydrogen (H_2) and carbon dioxide (CO_2) is equal to the rate of formation of carbon monoxide (CO) and water vapor (H_2O). The concentrations of CO, H_2O,H_2 and CO_2 in the equilibrium mixture can be determined from their partial pressures through the equilibrium equation.

$$\frac{P_{CO2}\ P_{H2}}{P_{CO}\ P_{H2O}} = K_{P_1} \qquad (٣.٦١)$$

The partial pressures are the normalized partial pressures and K_{P_1} is the equilibrium constant.

The second reaction is the carbon monoxide **(CO)** reaction: this states that the rate of formation of carbon dioxide **(CO_2)** is equal to the rate of

formation of **(CO)** and oxygen **(O_2)** at equilibrium. The concentrations of **CO_2, CO, O_2** can be obtain from the equilibrium equation

$$\frac{(P_{CO_2})}{(P_{CO})(P_{O_2})^{0.5}} = K_{P_2} \qquad (٣.٦٢)$$

The normalized partial pressure P_i for substance i is

$$P_i = \frac{a\ n_i}{a\ N_T} p_T = \frac{n_i}{N_T} P_T$$

Where $\mathbf{P_T}$ is the normalized total pressure for the mixture at point **C** Fig. (٣.١), $\mathbf{N_T}$ is the total number of moles of products.

Substitute the partial pressure P_i in the equilibrium equation (٣.٦١) and (٣.٦٢) to get: -

$$\frac{n_1 n_4}{n_2 n_3} = K_{P_1} \qquad (٣.٦٣)$$

And,

$$(\frac{n_1}{n_2})^2 \frac{1}{n_5} = \frac{P_T}{N_T}(K_{P_2})^2 \qquad (٣.٦٤)$$

To obtain the number of mole $\mathbf{n_1}$ to $\mathbf{n_5}$ chemical mass balance for carbon, hydrogen and oxygen have to be performed these are:

Assume $\alpha = \gamma = 0$, then for: -

Carbon:

$n=n_1+ n_2$ (٣.٦٥)

Hydrogen:

$0.5m= n_3+ n_4$ (٣.٦٦)

Oxygen:

$$2(n+0.5m) \quad \frac{1}{\phi} \quad = \quad 2n_1+ \quad n_2+ \quad n_3+ \quad 2n_5$$

(٣.٦٧)

The state equation at **C** (fig ٣.١) is

$P_C V_C = a\ N_T R_{mol} T_C$

Where

$$\frac{P_C}{N_T} = \frac{aR_{mol}T_C}{V_C} \qquad (٣.٦٨)$$

We can combine equation (٣.٦٣), and equation (٣.٦٨) in terms of $n_١$ to give

$$f(n_1) = (n_1 - (\frac{2}{\varphi}(n + 0.25m) - (n + 0.25m))) -$$
$$0.5m(\frac{(n - n_1)}{(n + n_1(\frac{1}{K_{P_1}} - 1))}) + (\frac{2N_T}{P_C K_{P_1}^2})(\frac{n_1}{(n - n_1)})^2 = 0 \qquad (٣.٦٩)$$

By using Newton-Raphson to solve for $\mathbf{n}_١$, then

$$\frac{df(n_1)}{dn_1} = (1 + \frac{m}{2}(\frac{(n + n_1(\frac{1}{K_{P_1}} - 1)) + (n - n_1)(\frac{1}{K_{P_1}} - 1)}{(n + n_1(\frac{1}{K_{P_1}} - 1))^2}) +$$
$$\frac{2N_T}{P_C (K_{P_2})^2}(\frac{2n_1(n - n_1)^2 + 2(n_1)^2(n - n_1)}{(n - n_1)^4}) \qquad (٣.٧٠)$$

If n_i is the estimated value of n_i (initial condition)'then n_{i+1} is

$$n_{i+1} = n_i - \frac{f(n_1)}{\frac{df(n_1)}{dn_1}}$$

Or

$$n_{i+1} = n_i - \frac{f(n_1)}{f'(n_1)} \qquad (٣.٧١)$$

Where $f'(n_1) = \frac{df(n_1)}{dn_1}$

Expression (٣.٦٩) gives the number of moles of carbon dioxide $n_١$ at any temperature. We can then find the number of moles of carbon monoxide $n_٢$ from (٣.٦٥): -

$$n_٢ = n - n_١ \qquad (٣.٧٢)$$

And the number of moles of oxygen ($n_٥$) from (٣.٦٤)

$$n_5 = \left(\frac{n_1}{n_2}\right)^2 \frac{N_T}{P_C} \frac{1}{(K_{P_2})^2} \qquad (٣.٧٣)$$

Combining equations (٣.٦٥), (٣.٦٦) and (٣.٦٧) the number of moles of hydrogen ($n_٤$) can be obtain from: -

$$n_٤ = n_١ + ٢\, n_٥ - \left(\frac{2}{\phi}\,(n+٠.٢٥m)-(n+٠.٢٥m)\right) \qquad (٣.٧٤)$$

And the number of moles of water vapour ($n_٣$) from equation (٣.٦٦),

$$n_٣ = ٠.٥m - n_٤ \qquad (٣.٧٥)$$

Finally, the nitrogen composition ($n_٦$) is obtain directly from equation (٣.٦٠)

$$n_٦ = \frac{3.76}{\phi}\,(n+٠.٢٥m) \qquad (٣.٧٦)$$

Thus if we known the maximum temperature $\mathbf{T_C}$ we can determine the composition of the products of combustion. Temperature $\mathbf{T_C}$ is obtain from the first law of thermodynamic:

For adiabatic combustion **dQ=٠** and for constant volume combustion **dW=٠**, then

$$dU = U_P - U_R \qquad (٣.٧٧)$$

As before, the internal energy U is defined as

$$U = U_O + U(T)$$

At the state B the internal energy is

$$U_R = U_{oR} + U_R(T_R) \qquad (٣.٧٨)$$

At the state C the internal energy is

$$U_p = U_{op} + U_p(T_p) \qquad (٣.٧٩)$$

The internal energies ca be expressed in terms of the composition of the mixtures at B and C and the specific internal energies, then: -

$$U_{oR} = a\left((u_o)_{CnHm} + (n+٠.٢٥m)\frac{1}{\phi}(u_o)_{O٢} + \frac{3.76}{\phi}(n+٠.٢٥m)\,(u_o)_{N٢}\right)$$

(٣.٨٠)

$$U_R(T_R)=a(u(T_R)_{CnHm}+\frac{1}{\phi}(n+٠.٢٥m)u(T_R)_{O٢}+\frac{3.76}{\phi}(n+٠.٢٥m)\,u(T_R)_{N٢})$$

(٣.٨١)

$$U_{OP}=a(n_١(u_o)_{CO٢}+n_٢(u_o)_{CO}+n_٣(u_o)_{H٢O}+n_٤(uo)_{H٢}+n_٥(u_o)_{O٢}+n_٦(u_o)_{N٢})$$

(٣.٨٢)

$$U_P(T_P)=a(n_١u(T_P)_{CO٢}+n_٢u(T_P)_{CO}+n_٣u(T_P)_{H٢O}+n_٤u(T_P)_{H٢}+n_٥u(T_P)_{O٢}+n_٦u(T_P)_{N٢})$$

(٣.٨٣)

If $\mathbf{U_P}$ and $\mathbf{U_R}$ are substituted into equation (٣.٧٧) then,

$$f(T_P)=(U_{oP}+U_P(T_P))-(U_{oR}+U_R(T_R)=٠ \qquad (٣.٨٤)$$

Now the internal energies U_{OR}, $U_R(T_R)$, from equation (٣.٨٠) and (٣.٨١) are fixed since they refer to the reactants at temperature $\mathbf{T_R}$.

The internal energies U_{OP} will depend on the composition of the products and this depend on $\mathbf{T_P}$; the internal energy $U_P(T_P)$ will depend on the composition of the products and the temperature. If : -

$$\frac{dU_{OP}}{dT_P}=0$$

Then,

$$\frac{df(T_P)}{dT_P}=\frac{dU_P(T_P)}{dT_P}=U'(T_P) \qquad (٣.٨٥)$$

And from $C_V=(\frac{\partial u}{\partial T})_V$, then

$$U'(T_P)=N_TC_V(T_P)=aN_TC_V(T_P) \qquad (٣.٨٦)$$

Where

$N_T=n_١+n_٢+n_٣+n_٤+n_٥+n_٦$

Equation (٣.٨٤) can then be solved numerically by Newton-Raphson method:

$$(T_P)_{i+1} = (T_P)_i - \frac{f(T_P)_i}{N_T C_V (T_P)_i} \qquad (٣.٨٧)$$

The first estimate for $\mathbf{T_P}$ can be obtain by the approximate expressions [٢٥]

$$T_P = T_R + ٢٥٠٠\phi f \qquad \text{for } \phi \leq 1$$

$$T_P = T_R + ٢٥٠٠\phi f - 700(\phi - 1) f \qquad \text{for } \phi > 1$$

Where,

ϕ Is the equivalence ratio and f is the residual mass fraction.

The maximum temperature in the cycle corresponds to $\mathbf{T_P}$. This is dependent on the air to fuel ratio, the temperature and pressure at the commencement of combustion and the fuel.

٣.٤.٢: *Expansion Stroke*

This corresponds to the period from **C** to **D** in the pressure volume diagram Fig.(٣.١).

Although during combustion the maximum temperature is reached, the chemical reactions continue to take place during the expansion stroke; this is because the chemical species are in thermodynamic equilibrium.

The composition of the gas mixture will therefore vary with pressure and temperature. To satisfy thermodynamic equilibrium the chemical reactions for water gas and carbon dioxide, continuously take place. The analysis for the determination of the composition given for the combustion process (section ٣.٤.٢) is therefore the same for the expansion process. If a small change from $\mathbf{V_١}$ to $\mathbf{V_٢}$ takes place, then pressure and temperature will change from $\mathbf{P_١}$, $\mathbf{T_١}$, to $\mathbf{P_٢}$, $\mathbf{T_٢}$.

These will be related by

$P_١V_١ = N_١R_{mol}T_١$ and $P_٢V_٢ = N_٢R_{mol}T_٢$,

And the number of moles at initial **(١)** and final **(٢)** states will be

$N_١ = (n_١ + n_٢ + n_٣ + n_٤ + n_٥ + n_٦)_{T١}$,

$N_٢ = (n_١ + n_٢ + n_٣ + n_٤ + n_٥ + n_٦)_{T٢}$,

To obtain T_2 at the end, the cylinder volume is divided into small change in volume.

Applying the first law of thermodynamic,

And for an adiabatic process **dQ=0** with a small step V_1 to V_2 , the work is approximately: -

$$dW=PdV=0.5(P_1+P_2)(V_2-V_1) \quad (3.88)$$

The change in internal energy **dU** is

$$dU=U_2-U_1 \quad (3.89)$$

For temperature T_1 the internal energies are;

$$U_{OP_1} = a(n_1(u_o)_{CO_2} + n_2(u_o)_{CO} + n_3(u_o)_{H_2O} + n_4(u_o)_{H_2} + n_5(u_o)_{O_2} + n_6(u_o)_{N_2}) \quad (3.90)$$

$$U_P(T_1) = a(n_1u(T_1)_{CO_2} + n_2u(T_1)_{CO} + n_3u(T_1)_{H_2O} + n_4u(T_1)_{H_2} + n_5u(T_1)_{O_2} + n_6u(T_1)_{N_2}) \quad (3.91)$$

And at T_2;

$$U_{OP_2} = a(n_1(u_o)_{CO_2} + n_2(u_o)_{CO} + n_3(u_o)_{H_2O} + n_4(u_o)_{H_2} + n_5(u_o)_{O_2} + n_6(u_o)_{N_2}) \quad (3.92)$$

$$U_P(T_2) = a(n_1u(T_2)_{CO_2} + n_2u(T_2)_{CO} + n_3u(T_2)_{H_2O} + n_4u(T_2)_{H_2} + n_5u(T_2)_{O_2} + n_6u(T_2)_{N_2}) \quad (3.93)$$

The number of moles n_1 to n_6 being different at each temperature. At the initial temperature T_1 the internal energies U_{OP} and $U_p(T_1)$ are constant; however , the first law of thermodynamic then becomes: -

$$f(T_2) = (U_{OP2} + U_P(T_2)) - (U_{OP1} + U_P(T_1)) + (\frac{P_1 + P_2}{2})(V_1 - V_2) = 0 \quad (3.94)$$

Where: - $\frac{df(T_2)}{dT_2} = 0$ and $\frac{dU_{OP}}{dT_2} = 0$,

Then,

$$\frac{df(T_2)}{dT_2} = \frac{dU_P(T_2)}{dT_2} = f'(T_2) = N_2C_V(T_2) \quad (3.95)$$

Equation (٣.٩٤) can be solved numerically using Newton-Raphson method to give : -

$$(T_2)_{i+1} = (T_2)_i - \frac{f(T_2)_i}{f'(T)_i} \qquad (٣.٩٦)$$

The initial condition at the first estimate of $\mathbf{T_2}$,

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{\frac{R_{mol}}{C_V(T_1)}} \qquad (٣.٩٧)$$

And the pressure ratio is

$$\frac{P_2}{P_1} = \frac{N_2}{N_1}\frac{V_1}{V_2}\frac{T_2}{T_1} \qquad (٣.٩٨)$$

٣.٥: Geometrical Properties of Reciprocating Engines

The following parameters define the basic geometry of a reciprocating engine Fig.(٣.٢)

Compression ratio CR

$$CR = \frac{\text{max imum cylinder volume}}{\text{min imum cylinder volume}} = \frac{V_d + V_c}{V_c} \qquad (٣.٩٩)$$

Where V_d is the swept volume and V_c is the clearance volume.

Ratio of cylinder bore to piston stroke;

$$R_{bs} = \frac{D}{L} \qquad (٣.١٠٠)$$

Ratio of connecting rod length to crank radius;

$$R = \frac{\ell}{r} \qquad (٣.١٠١)$$

The stroke and crank radius are related by

$$L = ٢r \qquad (٣.١٠٢)$$

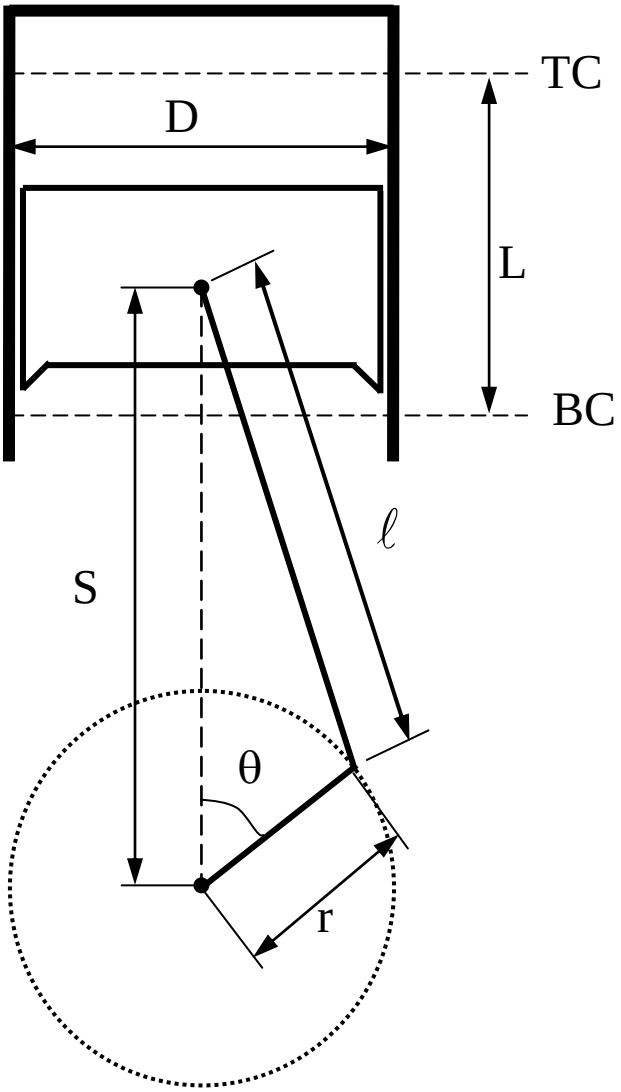


Fig. (٣.٢): geometry of cylinder, piston, connecting rod, and crankshaft.

The cylinder volume V at any crank position θ is

$$V = V_C + \frac{\pi}{4} D^2 (\ell + r - s) \qquad (٣.١٠٣)$$

Where **S** is the distance between crank axis and the piston pin axis and is given by

$$S = r\cos\theta + (\ell^2 - r^2\sin^2\theta)^{\frac{1}{2}} \qquad (٣.١٠٤)$$

Where: - Θ ..is the crank angle. Equation (٣.١٠٣) with the above definition can be rearranged to give: -

$$\frac{V}{V_C} = 1 + \frac{1}{2}(CR - 1)(R + 1 + \cos\theta - (R^2 - \sin^2\theta)^{\frac{1}{2}} \qquad (٣.١٠٥)$$

The combustion chamber surface area **A** at any crank position θ is given by: -

$$A = A_{ch} + A_P + \pi D(\ell + r - s) \qquad (M)$$

Where A_{ch} is the cylinder head surface area and $\mathbf{A_P}$ is the piston crown surface area. For flat-topped pistons, $A_P = \frac{\pi}{4}D^2$. Using equation. (٣.١٠٤)and equation (M) can be rearranged:

$$A = A_{ch} + A_P + \frac{\pi DL}{2}(R + 1 - \cos\theta - (R^2 - \sin^2\theta)^{\frac{1}{2}}) \qquad (٣.١٠٦)$$

An important characteristic speed is the mean piston speed $\overline{S}_P$ defined as: -

$$\overline{S}_P = 2L\,rps \qquad (٣.١٠٧)$$

Where **rps** is the rotational speed of the crankshaft per second.

٣.٦: Gibbs Frees Energy

With knowledge of two properties of a substance all other thermodynamic properties can be calculated through the relations. the Gibbs free energy function is: -

$g=h-TS$ (٣.١٠٨)

Where

g: Gibbs free energy function

T: temperature (K)

S: entropy (kJ/kg. K), which is defined as: -

$S=S(T)-R_{mol}\ln P+S_O$ (٣.١٠٩)

Substituting equation. (٣.١٠٩) into equation. (٣.١٠٨) gives: -

$g=h-T(S=S(T)-R_{mol}\ln P+S_o)$ (٣.١١٠)

Substituting equation. (٣.٢٠) into equation. (٣.١١٠), gives: -

$g=(h_o-TS_o)+(h(T)-TS(T))+R_{mol}T\ln P$ (٣.١١١)

But

$g_o=h_o-TS_o$ (٣.١١٢)

$g(T)=h(T)-TS(T)$ (٣.١١٣)

Where $\mathbf{g_o}$ is the Gibbs free energy at absolute zero then: -

$g=g(T)+R_{mol}T\ln P+g_o$ (٣.١١٤)

And,

$g^o=g(T)+g_o$ (٣.١١٥)

Substitute equation. (٣.١١٥) in equation. (٣.١١٤), gives: -

$g=g^o+R_{mol}T\ln P$ (٣.١١٦)

٣.١: Calculation of Equilibrium Constant.

To calculate the equilibrium constant, with stoichiometric reaction: -

$$\nu_a A+\nu_b B \Leftrightarrow \nu_c C+\nu_d D \quad (٣.١١٧)$$

Application of equation (٣.١١٦), gives,

$$\nu_a g_a^o+\nu_b g_b^o-\nu_c g_c^o-\nu_d g_d^o = R_{mol}T(\nu_C \ln P_c+\nu_d \ln P_d-\nu_a \ln P_a-\nu_b \ln P_b) \quad (٣.١١٨)$$

And the standard state Gibbs- function change is determined by: -

$$-\Delta G_T^o=\nu_a g_a^o+\nu_b g_b^o-\nu_c g_c^o-\nu_d g_d^o \quad (٣.١١٩)$$

In terms of partial pressure equation (٣.١١٨) can be written as follows: -

$$-\Delta G_T^o=R_{mol}T\ln\frac{P_c^{\nu_c}P_d^{\nu_d}}{P_a^{\nu_a}P_b^{\nu_b}} \quad (٣.١٢٠)$$

Equation (٣.١٢٠) becomes as follows:

$$\ln\frac{P_c^{\nu_c}P_d^{\nu_d}}{P_a^{\nu_a}P_b^{\nu_b}}=\frac{-\Delta G_T^o}{R_{mol}T}=\ln K_P \quad (٣.١٢١)$$

Or

$$K_P = \ln \frac{P_c^{\nu_c} P_d^{\nu_d}}{P_a^{\nu_a} P_b^{\nu_b}} \qquad (٣.١٢٢)$$

Where $\mathbf{K_P}$ is the equilibrium constant.

From equation (٣.١٢١)

$$\ln K_P = \frac{-\Delta G_T^o}{R_{mol} T} \qquad (٣.١٢٣)$$

Since ΔG_T^o is function of temperature only and the equilibrium constants are function of temperature only; therefore, the equilibrium constant can be expressed in terms of the mole-fraction and the total pressure as follows: -

$$K_P = \left(\frac{X_c^{\nu_c} X_d^{\nu_d}}{X_a^{\nu_a} X_b^{\nu_b}}\right) P^{\nu_c + \nu_d - \nu_a - \nu_b} \qquad (٣.١٢٤)$$

Where $\mathbf{X_a}$, $\mathbf{X_b}$, $\mathbf{X_c}$, and $\mathbf{X_d}$ are the mole fraction of reactions and product.

From equation (٣.١١٥) $g^o = g(T) + g_o$

If the reference temperature is equal to absolute zero the entropy will be zero for a pure substance, equation (٣.١١٢) then becomes

$$G_o = h_o \qquad (٣.١٢٥)$$

And

$$g(T) = g^o - h_o \qquad (٣.١٢٦)$$

Equation (٣.١٢٣) can be applied to the chemical reaction, which contain a number of reactants and products, then: -

$$-\Delta G_T^o = \sum(\nu\ g^o)_R - \sum(\nu\ g^o)_P \qquad (٣.١٢٧)$$

And,

$$\ln K_P = \frac{1}{R_{mol}} \left[\sum(\nu\ g^o)_R - \sum(\nu\ g^o)_P\right] \qquad (٣.١٢٨)$$

Substituting equation (٣.١٢٦) into equation (٣.١٢٨) gives: -

$$\ln K_P = \sum\left[\frac{\nu(g(T)+h_o)}{R_{mol}T}\right]_R - \sum\left[\frac{\nu(g(T)+h_o)}{R_{mol}T}\right]_P \qquad (٣.١٢٩)$$

$$\ln K_P = \left\{\sum\left[\frac{\nu g(T)}{R_{mol}T}\right]_R - \sum\left[\frac{\nu g(T)}{R_{mol}T}\right]_P\right\} + \left\{\sum\left[\frac{\nu h_o}{R_{mol}T}\right]_R - \sum\left[\frac{\nu g h_o}{R_{mol}T}\right]_P\right\} \qquad (٣.١٣٠)$$

And

$$\frac{-\Delta H_o}{R_{mol}T} = \sum\left[\frac{\nu h_o}{R_{mol}T}\right]_R - \sum\left[\frac{\nu g h_o}{R_{mol}T}\right]_P \qquad (٣.١٣١)$$

Substituting equation (٣.١٣١), into equation (٣.١٣٠), gives: -

$$\ln K_P = \left\{\sum\left[\frac{\nu g(T)}{R_{mol}T}\right]_R - \sum\left[\frac{\nu g(T)}{R_{mol}T}\right]_P\right\} - \frac{\Delta H_o}{R_{mol}T} \qquad (٣.١٣٢)$$

The function g(T) can be evaluated using the polynomial coefficient data in the appendix A,

The enthalpy **h**, the stander Gibbs function g^o and the temperature **T** are related by [٢٦],

$$-\frac{h}{T^2} = \frac{d(\frac{g^o}{T})}{dT} \qquad (٣.١٣٣)$$

Where

$g^o = g(T) + h_o$

$h = h(T) + h.$

The specific enthalpy **h** and **h(T)** for a substance are given in equation (٣.٢٠) and (٣.٢٢) respectively,

$$\frac{h(T)}{R_{mol}T} = \frac{h - h_o}{R_{mol}T} = Z_{i1} + Z_{i2}T + Z_{i3}T^2 + Z_{i4}T^3 + Z_{i5}T^4 \qquad (٣.١٣٤)$$

Dividing by T we obtain

$$\frac{h(T)}{R_{mol}T^2} = \frac{Z_{i1}}{T} + Z_{i2} + Z_{i3}T + Z_{i4}T^2 + Z_{i5}T^3$$

Substituting for $\frac{h(T)}{R_{mol}T^2}$ and $\frac{g^0}{T}$ in equation. (٣.١٣٣), the following simple differential equation is formed:

$$-\left[\frac{h_0}{R_{mol}T^2}+\frac{Z_{i1}}{T}+Z_{i2}+Z_{i3}T\ +Z_{i4}T^2+Z_{i5}T^3\right]dT=\frac{1}{R_{mol}}d\left(\frac{g(T)+h_0}{T}\right) \quad (٣.١٣٥)$$

Integration equation. (٣.١٣٧) gives

$$\frac{g(T)}{R_{mol}T}=Z_{i1}(1-\ln T)-\sum_{i=2}^{i=5}\frac{Z_i}{i-1}T^{i-1}-Z_{i6} \quad (٣.١٣٦)$$

٣.٨: Adiabatic Flame Temperature

The Adiabatic flame temperature is the ideal theoretical maximum temperature that can be obtained from burning of a given fuel and air mixture.

An estimation of the maximum temperature reached in an IC engine can be obtained by calculating the (adiabatic flame temperature) of the input air- fuel mixture, this is done as follows:

$$\sum_{prod}N_ih_i=\sum_{react}N_ih_i \quad (٣.١٣٧)$$

And

$$f(T_{ad})=\sum_{prod}N_ih_i-\sum_{react}N_ih_i \quad (٣.١٣٨)$$

The total enthalpy of products $H_{prod}=\sum_{prod}N_ih_i$, and the total enthalpy of reactants $H_{react}=\sum_{react}N_ih_i$, and substituting these expressions in equation (٣.١٣٨), gives: -

$$f(T_{ad})=H_{prod}-H_{react} \quad (٣.١٣٩)$$

And

$$\frac{df(T_{ad})}{dT_{ad}} = \frac{dH_{prod}}{dT_{ad}} \qquad (٣.١٤٠)$$

Since from $C_P = (\frac{\partial h}{\partial T})_P$, then equation. (٣.١٤٠), becomes

$$\frac{df(T_{ad})}{dT_{ad}} = (N_i C_P)_{prod} = f'(T_{ad}) \qquad (٣.١٤١)$$

By using Newton-Raphson method to solve the equation above to get the adiabatic flame temperature.

$$(T_{ad})_{i+1} = (T_{ad})_i - \frac{f(T_{ad})_i}{f'(T_{ad})_i} \qquad (٣.١٤٢)$$

The first estimate for T_{ad} can be obtained by the approximate expressions from [٢٥]

$T_{ad}=T_R+٢٥٠٠\phi f$ for $\phi \leq 1$

$T_{ad}=T_R+٢٥٠٠\phi f -700(\phi-1)f$ for $\phi > 1$

This adiabatic flame temperature can be assumed to be the temperature of the burnt gases at the beginning of combustion [٢٧]

٣.٩:Combustion and Thermodynamic Equilibrium

The combustion equation for fuel of the form $C_nH_mO_\gamma N_\alpha$ can be written as:

$$a(C_nH_mO_\gamma N_\alpha + \frac{a}{\phi}(n+0.25m-0.5\gamma)\left[O_2 + \frac{78}{21}N_2 + \frac{1}{12}AR\right] \rightarrow X_1H_2O + X_2H_2$$
$$+ X_3OH + X_4H + X_5N_2 + X_6NO + X_7N + X_8CO_2 + X_9CO + X_{10}O_2 + X_{11}O + X_{12}AR \qquad (٣.١٤٣)$$

At low temperatures, typically below ١٧٠٠K ,$X_٤$, $X_٦$, $X_٧$, and $X_{١١}$ are zero[١].For lean and stoichiometric mixtures ($\phi \leq 1$) CO and $H_٢$ can be neglected and atom balance equations are sufficient to determine the products composition. For rich mixtures where ($\phi > 1$) O_2 can be neglected and an equilibrium reactions are introduced to determine the molar fractions.

The following species are considered to be present in the products inside the cylinder and in the exhaust gases. They are referred to by the number in brackets appearing against them.

(١) H_2O (٢) H_2 (٣) OH (٤) H (٥) N_2 (٦) NO

(٧) N (٨) CO_2 (٩) CO (١٠) O_2 (١١) O (١٢) AR.

The composition is calculated in terms of molar fractions of these species denoted by X_i. The fuel is written as $C_nH_mO_\gamma N_\alpha$, and one mole of the total products arises from a mole of fuel plus $(\frac{1}{\phi})$ times the stoichiometric quantity of air (ϕ = equivalence ratio).

The equilibrium distribution of theses species can be fully described by the following reactions [٢٥]: -

$$(1)\ \frac{1}{2}H_2 \Leftrightarrow H \qquad (٣.١٤٤)$$

$$(2)\ \frac{1}{2}O_2 \Leftrightarrow O \qquad (٣.١٤٥)$$

$$(3)\ \frac{1}{2}N_2 \Leftrightarrow N \qquad (٣.١٤٦)$$

$$(4)\ 2H_2O \Leftrightarrow 2H_2 + O_2 \qquad (٣.١٤٧)$$

$$(5)\ H_2O \Leftrightarrow OH + \frac{1}{2}H_2 \qquad (٣.١٤٨)$$

$$(6)\ CO_2 + H_2 \Leftrightarrow H_2O + CO$$

(٣.١٤٩)

Using the expressions of equilibrium constant , equation (٣.١٢٤) can be rewritten as,

$$(1)\ K_{P_1} = \frac{\sqrt{P}X_4}{\sqrt{X_2}} \qquad (٣.١٥٠)$$

And K_{P_1} can be calculated from equation (٣.١٣٢), after substituting equation (٣.١٣٦) into equation (٣.١٣٢), then

$$\ln K_P = \left\{ \sum \left[\frac{\nu g(T)}{R_{mol} T} \right]_R - \sum \left[\frac{\nu g(T)}{R_{mol} T} \right]_P \right\} - \frac{\Delta H_o}{R_{mol} T}$$

And

$$\frac{g(T)}{R_{mol} T} = Z_{i1}(1 - \ln T) - (Z_{i2} T + Z_{i3} \frac{T^2}{2} + Z_{i4} \frac{T^3}{3} + Z_{i5} \frac{T^4}{4}) - Z_{i6}$$

Let

$$F(i) = (Z_{i2} T + Z_{i3} \frac{T^2}{2} + Z_{i4} \frac{T^3}{3} + Z_{i5} \frac{T^4}{4})$$

Where:

i… any species

$$\Delta H_O = (N_i h_o)_P - (N_i h_o) = 216110.1 \ \ 10^3 \qquad \frac{J}{kg.mole}$$

And,

$$\ln K_{P_1} = (0.5 Z_{21} - Z_{41})(1 - \ln T) - (0.5F(2) - F(4))$$
$$- (0.5 Z_{26} - Z_{46}) - \frac{\Delta H_O}{R_{mol} T} \qquad (٣.١٥١)$$

$$(2) \ \ K_{P_2} = \frac{\sqrt{P} X_{11}}{\sqrt{X_{10}}} \qquad (٣.١٥٢)$$

$$\Delta H_O = 246923.1 \ \ 10^3 \qquad \frac{J}{kg.mole}$$

$$\ln K_{P_2} = (0.5 Z_{101} - Z_{111})(1 - \ln T) - (0.5F(10) - F(11))$$
$$- (0.5 Z_{106} - Z_{116}) - \frac{\Delta H_O}{R_{mol} T} \qquad (٣.١٥٣)$$

$$(3) \ \ K_{P_3} = \frac{\sqrt{P} X_7}{\sqrt{X_5}} \qquad (٣.١٥٤)$$

$$\Delta H_O = 471368.6 \ \ 10^3 \qquad \frac{J}{kg.mole}$$

$$\ln K_{P_3} = (0.5Z_{51} - Z_{71})(1 - \ln T) - (0.5F(5) - F(7))$$
$$- (0.5Z_{56} - Z_{76}) - \frac{\Delta H_O}{R_{mol} T} \qquad (٣.١٥٥)$$

$$(4)\quad K_{P4} = \frac{PX_{10}}{b^2} \qquad (٣.١٥٦)$$

$$\Delta H_O = 4.78464 \quad 10^8 \qquad \frac{J}{kg.mole}$$

$$\ln K_{P_4} = ((2Z_{11} - 2Z_{21} - Z_{101}))(1 - \ln T) - (2F(1) - 2F(7) - F(10))$$
$$- (2Z_{16} - 2Z_{26} - Z_{106}) - \frac{\Delta H_O}{R_{mol} T} \qquad (٣.١٥٧)$$

$$(5)\quad K_{P5} = \frac{\sqrt{P}X_3}{b\sqrt{X_2}} \qquad (٣.١٥٨)$$

$$\Delta H_O = 2.78668 \quad 10^8 \qquad \frac{J}{kg.mole}$$

$$\ln K_{P_4} = ((Z_{11} - Z_{31} - Z_{21}))(1 - \ln T) - (F(1) - F(3) - F(2))$$
$$- (Z_{16} - Z_{36} - Z_{26}) - \frac{\Delta H_O}{R_{mol} T} \qquad (٣.١٥٩)$$

$$(6)\quad K_{P6} = \frac{bX_9}{X_8} \qquad (٣.١٦٠)$$

$$\Delta H_O = 0.4052 \quad 10^8 \qquad \frac{J}{kg.mole}$$

$$\ln K_{P_6} = ((Z_{81} + Z_{21} - Z_{11} - Z_{91}))(1 - \ln T) - (F(8) + F(2) - F(1) - F(9))$$
$$- (Z_{86} + Z_{26} - Z_{16} - Z_{96}) - \frac{\Delta H_O}{R_{mol} T} \qquad (٣.١٦١)$$

$$(7)\quad K_{P7} = \frac{\sqrt{P}X_6}{b\sqrt{X_5}} \qquad (٣.١٦٢)$$

$$\Delta H_O = 3.289977 \quad 10^8 \qquad \frac{J}{kg.mole}$$

$$\ln K_{P_7} = ((Z_{11} + 0.5Z_{51} - Z_{21} - Z_{61}))(1 - \ln T) - (F(1) + F(5) - F(2) - F(6))$$
$$-(Z_{16} + 0.5Z_{56} - Z_{26} - Z_{66}) - \frac{\Delta H_O}{R_{mol} T} \qquad (٣.١٦٣)$$

Where: $b = \frac{X_1}{X_2}$

Z_{ij}… polynomial coefficient in appendix (A)

i=١ to ١٢

J=١ to ٦

$X_١$, $X_٢$, $X_٣$, $X_٤$, $X_٥$, $X_٦$, $X_٧$, $X_٨$, $X_٩$, $X_{١٠}$, $X_{١١}$, and $X_{١٢}$ are the mole fraction of $H_٢O$, $H_٢$, OH, H, $N_٢$, NO, N, $CO_٢$, CO, $O_٢$, O, and AR respectively.

There are ١٢ unknown' species, then ١٢ equations are required for the solution.

one of these is

$$\sum X_i = 1 \qquad (٣.١٦٤)$$

The remaining are the atomic mass balances for argon, carbon, hydrogen, oxygen and nitrogen.

(١) Argon: $X_{١٢}=a\nu$ (3.165)

(2) Carbon: $X_٨ + X_٩ = an$ (٣.١٦٦)

(٣) Hydrogen $٢X_١ + ٢X_٢ + X_٣ + X_٤ = am$ (٣.١٦٧)

(٤) Oxygen $X_١ + X_٣ + X_٦ + ٢X_٨ + X_٩ + ٢X_{١٠} + X_{١١} = ay$ (٣.١٦٨)

(٥) Nitrogen $٢X_٥ + X_٦ + X_٧ = az$ (٣.١٦٩)

Where **ν**, **n, m, y,** and **z** are the number of atoms of **A, C, H, O**, and **N** corresponding to ١mole of fuel plus the air, ١mole of product arises from a mole of fuel plus $(\frac{1}{\phi})$ stoichiometric quantity of air.

Since,

$$Y = \frac{1}{\phi}[n + ٠.٢٥m - ٠.٥\gamma] \qquad (٣.١٧٠)$$

$$z = ٣.٧٦y \qquad (٣.١٧١)$$

$$\nu = \frac{y}{2}\frac{1}{12} \qquad (٣.١٧٢)$$

a and **b** can be calculated as follows[٢٦].

When $\phi \geq 1$,

$$a = \frac{1.3}{\left[(n+0.5m) + \frac{1.863(2n+0.5m-\gamma)}{\phi}\right]\exp^{\frac{0.13T}{1000}}} \qquad (3.173)$$

And when $\phi < 1$,

$$a = \frac{1.3}{\left[(0.5m) + \frac{2.363(2n+0.5m-\gamma)}{\phi}\right]\exp^{\frac{0.13T}{1000}}} \qquad (3.174)$$

When T >٣٠٠٠K

$$b = \exp(10.3 - \frac{(3.1 - 0.17\log p)T}{1000} \qquad (٣.١٧٥)$$

And when T ≤ ٣٠٠٠K

$$b = \exp(-9 + 0.5\log p + \frac{30000}{T}) \qquad (٣.١٧٦)$$

Adjust b for the following case

B_X=٢-٩logϕ when B_X> ٣.٥,

B=exp [exp (٣.٥)+٠.٢٥logP] when $B_X \leq$ ٣.٥, (٣.١٧٧)

B=exp [exp (٢-٩logϕ)+٠.٢٥logP] when b> B, (٣.١٧٨)

b=B

Where **P** is the pressure in atmospheres

Having fixed the values of a and b, $X_٢$ can be calculated by the equilibrium constant, as before the ١٢ mole fractions can then be evaluated as following:

Using equations (٣.١٥٠), (٣.١٥٨), (٣.١٦٣), and (٣.١٦٧) and after rearranging in terms of $X_٢$ the mole fraction of hydrogen: -

$$X_2 = \left[\frac{-\left[\frac{K_{P_5} b}{\sqrt{P}} + \frac{K_{P_1}}{\sqrt{P}} \right] + \sqrt{\left(\frac{K_{P_5} b}{\sqrt{P}} + \frac{K_{P_1}}{\sqrt{P}} \right)^2 + 8(b+1)an}}{4(b+1)} \right]^2 \qquad (٣.١٧٩)$$

the mole fractions are calculated as follows: -

$$X_1 = BX_2 \qquad (٣.١٨٠)$$

$$X_3 = \frac{K_{P_5} b\sqrt{P}}{\sqrt{P}} \qquad (٣.١٨١)$$

$$X_4 = \frac{K_{P_1}\sqrt{X_2}}{\sqrt{P}} \qquad (٣.١٨٢)$$

$$X_8 = \frac{an}{\left[\frac{K_{P_6}}{b} + 1 \right]} \qquad (٣.١٨٣)$$

$X_9=an-X_8$ (٣.١٨٤)

Using equations (٣.١٥٤), (٣.١٦١) and (٣.١٦٩) after rearranging in terms of X_5 ,the mole fraction of nitrogen is obtained: -

$$X_5=\left[\frac{-\left[\frac{K_{P_7}b}{\sqrt{P}}+\frac{K_{P_3}}{\sqrt{P}}\right]+\sqrt{\left(\frac{K_{P_7}b}{\sqrt{P}}+\frac{K_{P_3}}{\sqrt{P}}\right)^2+8az}}{4}\right]^2$$

(٣.١٨٥)

The mole fraction of atomic nitrogen is obtained from equation (٣.١٥٤): -

$$X_7=\frac{K_{P_3}\sqrt{X_5}}{\sqrt{P}} \qquad (٣.١٨٦)$$

The mole fraction of nitric oxide is obtained from equation (٣.١٦١): -

$$X_6=\frac{K_{P_7}b\sqrt{X_7}}{\sqrt{P}} \qquad (٣.١٨٧)$$

The mole fraction of oxygen is obtained from equation (٣.١٥٦) and the mole fraction of atomic oxygen from equation (٣.١٥٢),

$$X_{10}=\frac{K_{P_4}b^2}{P} \qquad (٣.١٨٨)$$

$$X_{11}=\frac{K_{P_2}\sqrt{X_{10}}}{\sqrt{P}} \qquad (٣.١٨٩)$$

From equation (٣.١٦٥), the mole fraction of argon is obtained,

$X_{12}=a\nu$ (٣.١٩٠)

The values of ١٢ species mole fraction must satisfy the equation, (٣.١٦٤).

٣.١٠:*The Real cycle.*

It is the purpose of real cycle to show relationships between real cycles and their equivalent fuel-air cycles in this studies of time loss of combustion period and heat losses from the combustion chamber and

effect of mixture strength on flame speed and optimum spark timing for maximum cylinder pressure and torque.

Quantitative analysis of the differences between the actual cycle and its equivalent fuel-air cycle obviously required accurate pressure-volume diagrams of the real cycle.

For Real Cycles the Following Assumptions Can be Made: -

١-All gases are perfectly mixed, ideal gases with specific heat depending on temperature and gas composition.

٢-Valve opening and closing occur at the same timing ATDC and ABDC.

٣-Flow losses through valves are negligible only losses through the piston ring.

٤-Cylinder heat loss is considered.

٥-The compression and expansion strokes can be represented as real process compression and expansion.

٦- Combustion is taken as a two zone, separated by the flame front. And the duration of combustion is considered.

٧-Combustion products are ١٢ species.

٨-The cycle of events between valve closing and exhaust valve opening comprises:

a-Compression stroke

b-Ignition and propagation of flame front.

c-Expansion: (i) two zone; (ii) full products.

٣.١٠.a: Compression Stroke

The compression stroke process starts at the trapped condition. The state of the gas at this point is derived by using a perfect mixing model for air and fuel. During compression any reaction is neglected.

The first law of thermodynamic equation. (٣.٤١) is differentiated with respect to crank angle,

$$\frac{dQ}{d\theta} = MC_V \frac{dT}{d\theta} + P\frac{dV}{d\theta} \qquad (٣.١٩١)$$

The equation of state is

$$P_{(\theta+\Delta\theta)} V_{(\theta+\Delta\theta)} = mRT_{(\theta+\Delta\theta)} \qquad (٣.١٩٢)$$

Differentiating with respect to crank angle: -

$$\frac{dT}{d\theta} = T\left(\frac{1}{V}\frac{dV}{d\theta} + \frac{1}{P}\frac{dP}{d\theta}\right) \qquad (٣.١٩٣)$$

A combination of equations (٣.١٩٣), and (٣.١٩١) and rearranging gives

$$\frac{dP}{d\theta} = \frac{\left[-(1+\frac{R}{C_V})P\frac{dV}{d\theta} + (\frac{R}{C_V})\frac{dQ}{d\theta}\right]}{V} \qquad (٣.١٩٤)$$

The heat transfer rate from gas to wall is given by [٣٠]

$$\frac{Q}{A_S} = \frac{aK}{D}(Re)^b (T - T_W) \qquad (٣.١٩٥)$$

Differentiate equation. (٣.١٩٥) with respect to crank angle: -

$$\frac{dQ}{d\theta} = A_S \frac{C_1 K}{D}(Re)^{C_2}(T - T_W)\frac{1}{360\,rps} \qquad (٣.١٩٦)$$

Reynolds number is defined as [١٣]: -

$$Re = \frac{(\dot{m}_a + \dot{m}_f)D}{A_P \mu_g} \qquad (٣.١٩٧)$$

Where:

m_a = Mass flow rate of air into the cylinder (kg/sec)

m_f = Mass flow rate of fuel into the cylinder (kg/sec)

D= Cylinder bore (m)

A_P= area of piston face (m^2)

μ= Dynamic viscosity of gas in the cylinder (kg/sec. m)

Nusselt number in terms of Reynolds number as: -

$$Nu = \frac{h_g D}{K_g} = C_1 (Re)^{C_2} \qquad (٣.١٩٨)$$

Where: C_1=٠.٠٧ and C_2=٠.٧

K_g = thermal conductivity of cylinder gas (W/m. K)

h_g = average value of the convection heat transfer coefficient [٤], (W/m٢. K)

The viscosity of hydrocarbon-air mixture over the temperature range ٥٠٠ up to ٤٠٠K, for pressures from ١ up to ١٠٠atm, for ϕ=0 up to ϕ=4 . The viscosity of the mixture differs little from that of air. Therefore, a power law based on air viscosity data can be [١١],

$$\mu_{air} (\frac{kg}{m.sec}) = 3.3\ \ 10^{-7}\ \ T^{0.7} \qquad (٣.١٩٩)$$

Where **T** in Kelvin's, the viscosity of mixture

$$\mu_g = \frac{\mu_{air}}{1 + 0.027\phi} \qquad (٣.٢٠٠)$$

This correlation was corrected to include the effect of equivalence ratio ϕ. An expression to find the thermal conductivity of the mixture using Prandtl number data with fitting [١١]: -

$$K_g = \frac{\mu_g C_P}{pr} \qquad (٣.٢٠١)$$

And,

$pr = ٠.٠٥ + ٤.٢(\gamma - 1) - ٦.٧(\gamma - 1)^2$ For $\phi \le 1$

$$pr = \frac{0.05 + 4.2(\gamma - 1) - 6.7(\gamma - 1)^2}{1 + 0.015\ \ 10^{-6}(\phi T)}$$ For ١<$\phi \le 4$

The work is

$$\frac{dW}{d\theta} = P\frac{dV}{d\theta} \qquad (٣.٢٠٢)$$

Since,

$$\frac{dV}{d\theta} = \frac{V_C}{2}(CR-1)\left[\sin\theta + \frac{\sin\theta\cos\theta}{(R^2-\sin^2\theta)^{\frac{1}{2}}}\right] \qquad (٣.٢٠٣)$$

The numerical method used to solve equation (٣.١٩٣), (٣.١٩٤), (٣.١٩٦), and (٣.٢٠٢), is the fourth order Runge-Kutta method. Calculations done with crank angle steps.

$$Y_{i+1} = Y_i + \frac{\Delta\theta}{6}[k_1 + 2k_2 + 2k_3 + k_4] \qquad (٣.٢٠٤)$$

$$\frac{dY}{d\theta} = f(Y,\theta)$$

$$k_1 = f(Y_i,\theta)$$

$$k_2 = f(Y_i + \frac{k_1}{2}, \theta_i + \frac{\Delta\theta}{2})$$

$$k_3 = f(Y_i + \frac{k_2}{2}, \theta_i + \frac{\Delta\theta}{2})$$

$$k_4 = f(Y_i + k_3, \theta_i + \Delta\theta)$$

Where **Y** any variable of equations as before.

٣.١٠.b: Ignition and Propagation Flame Front

The combustion chamber is divided into two zones, corresponding to burned and unburned gas regions by an infinitesimal flame front with a spherical shape Fig.(٣.٣) [١]: -

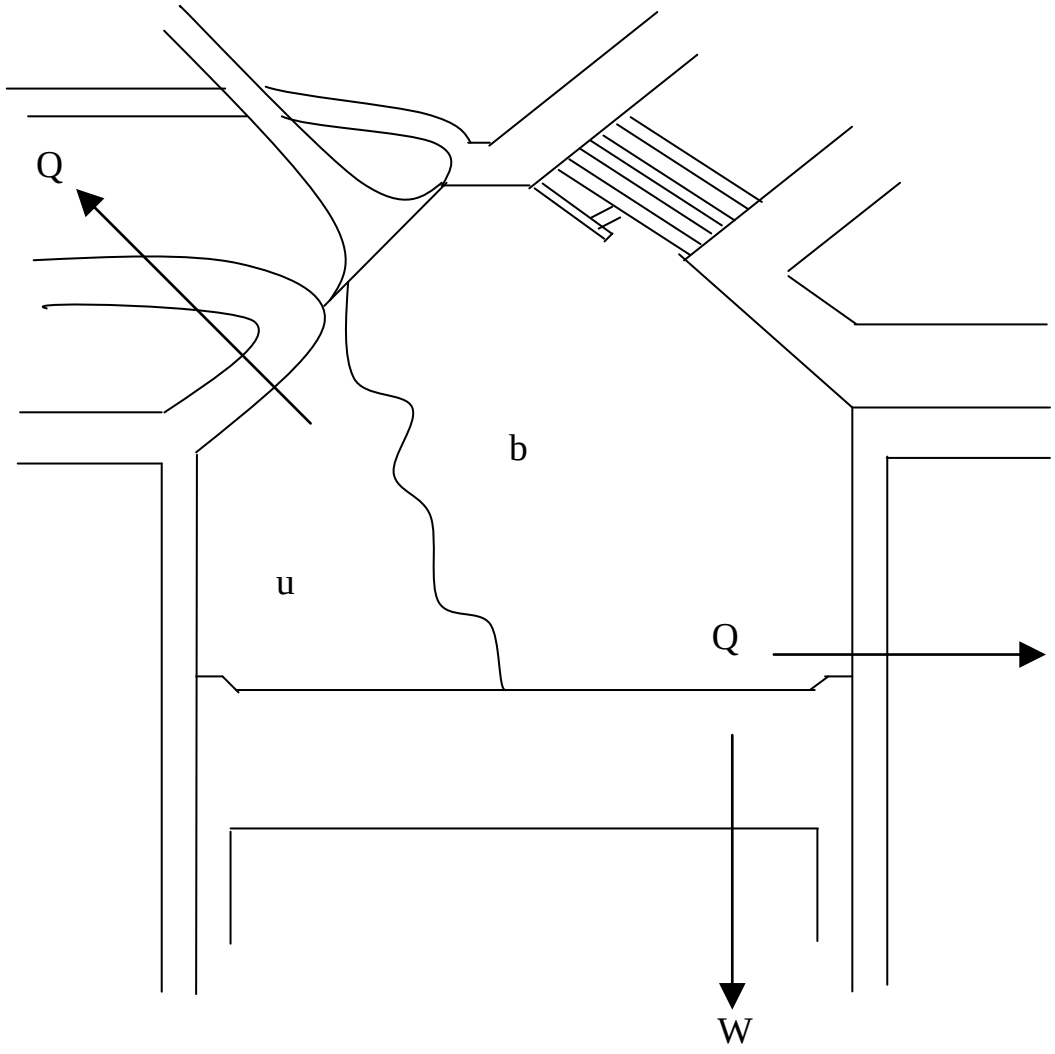


Figure. (٣.٣): schematic diagram of he combustion chamber[١] u.. unburned zone, b.. burned zone, W.. work, and Q.. heat.

Evaluation of instantaneous pressure, volume, temperature, thermodynamic properties and transport properties as a function of crank angle θ are obtained when an ordinary differential equation system is solved out. This ordinary differential equation system consists of equation of state and the first law of thermodynamics [٢٧] when applied to the open system formed by the gas inside cylinder.

$$m\frac{du}{d\theta}+u\frac{dm}{d\theta}=\frac{dQ}{d\theta}-P\frac{dV}{d\theta}-\frac{m_L h_L}{360\,rps} \qquad (٣.٢٠٥)$$

Where **m** total mass in the cylinder, **u** specific internal energy, **Q** amount of heat flowing into control volume, **P** gas pressure and **V** total volume. $\mathbf{m_L}$ and $\mathbf{h_L}$ mass loss and enthalpy loss, respectively.

Assuming **m** the total mass evolving in the cylinder. But **x** will be defined as the mass fraction of the cylinder contents that have been burned.

The specific internal energy is:

$$u = \frac{U}{m} = xu_b + (1-x)u_u \qquad (٣.٢٠٦)$$

Where subscript **u** and **b** refer to unburned and burned gas respectively. analogous relation is derived for the specific volume [٢]

$$\nu = \frac{V}{m} = x\nu_b + (1-x)\nu_u \qquad (٣.٢٠٧)$$

A set of ordinary differential equations will be derived describing the rates of change of pressure, burned temperature, unburned temperature, work, and heat loss with respect to crank angle. By simultaneously integrating these equations from the start of combustion until end of combustion. Using high order fourth Runge-Kutta method.

$$\nu_b = \nu_b(T_b, P) \qquad (٣.٢٠٨)$$

Differentiating with respect to crank angle: -

$$\frac{d\nu_b}{d\theta} = \frac{\partial\nu_b}{\partial T_b}\frac{dT_b}{d\theta} + \frac{\partial\nu_b}{\partial P}\frac{dP}{d\theta} \qquad (٣.٢٠٩)$$

After rearranging gives: -

$$\frac{d\nu_b}{d\theta} = \frac{\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\frac{dT_b}{d\theta} + \frac{\nu_b}{P}\frac{\partial \ln \nu_b}{\partial \ln P}\frac{dP}{d\theta} \qquad (٣.٢١٠)$$

By analogy

$$\frac{d\nu_u}{d\theta} = \frac{\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}\frac{dT_u}{d\theta} + \frac{\nu_u}{P}\frac{\partial \ln \nu_u}{\partial \ln P}\frac{dP}{d\theta} \qquad (٣.٢١١)$$

The relationship between u_b, T_b, and P is represented by [٣٦]: -

$$u_b = u_b(T_b, P) \qquad (٣.٢١٢)$$

$$\frac{du_b}{d\theta} = \frac{\partial u_b}{\partial T_b}\frac{dT_b}{d\theta} + \frac{\partial u_b}{\partial P}\frac{dP}{d\theta} \qquad (٣.٢١٣)$$

Where:

$$\frac{\partial u_b}{\partial T_b} = C_{P_b} - \frac{P\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b} \qquad \text{And,} \qquad \frac{\partial u_b}{\partial T_b} = -\nu_b\left(\frac{\partial \ln \nu_b}{\partial \ln T_b}\frac{\partial \ln \nu_b}{\partial \ln P}\right)$$

Rearranging : -

$$\frac{du_b}{d\theta} = \left(C_{P_b} - \frac{P\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\right)\frac{dT_b}{d\theta} - \nu_b\left(\frac{\partial \ln \nu_b}{\partial \ln T_b} + \frac{\partial \ln \nu_b}{\partial \ln P}\right)\frac{dP}{d\theta} \qquad (٣.٢١٤)$$

By analogy for the unburned gas,

$$\frac{du_u}{d\theta} = \left(C_{P_u} - \frac{P\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}\right)\frac{dT_u}{d\theta} - \nu_u\left(\frac{\partial \ln \nu_u}{\partial \ln T_u} + \frac{\partial \ln \nu_u}{\partial \ln P}\right)\frac{dP}{d\theta} \qquad (٣.٢١٥)$$

It is assumed that the pressures of the burned and unburned gases are equal, if the first law of thermodynamic equation (٣.٢٠٥) is taken term by term, equation (٣.٢٠٦) differentiate with respect to crank angle,

$$m\frac{du}{d\theta} = \left[x\frac{du_b}{d\theta} + (1-x)\frac{du_u}{d\theta} + (u_b - u_u)\frac{dx}{d\theta}\right]m \qquad (٣.٢١٦)$$

Substitution equations, (٣.٢١٤), and (٣.٢١٥), gives: -

$$\begin{aligned} m\frac{du}{d\theta} &= mx\left(C_{P_b} - \frac{P\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\right)\frac{dT_b}{d\theta} + \\ &m(1-x)\left(C_{P_u} - \frac{P\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}\right)\frac{dT_u}{d\theta} - \\ &\left[mx\nu_b\left(\frac{\partial \ln \nu_b}{\partial \ln T_b} + \frac{\partial \ln \nu_b}{\partial \ln P}\right) + m(1-x)\nu_u\left(\frac{\partial \ln \nu_u}{\partial \ln T_u} + \frac{\partial \ln \nu_u}{\partial \ln P}\right)\right]\frac{dP}{d\theta} \\ &+ m(u_b - u_u)\frac{dx}{d\theta} \end{aligned} \qquad (٣.٢١٧)$$

The change of mass with respect to crank angle, is

$$\frac{dm}{d\theta} = \frac{-m_\ell^{\cdot}}{360rps} = \frac{-Cm}{360rps} \qquad (٣.٢١٨)$$

Where: - $C = \frac{m_\ell}{m}$ which depend on ring design.

The heat flow into the system, on the right-hand side of equation (٣.٢٠٥), in terms of the heat loss is: -

$$\frac{dQ}{d\theta} = \frac{-Q_\ell}{360rps} = \frac{-Q_b - Q_u}{360rps} \qquad (٣.٢١٩)$$

Where,

$$Q_b = h_b A_b (T_b - T_W) \qquad (٣.٢٢٠)$$

$$Q_u = h_u A_u (T_u - T_W) \qquad (٣.٢٢١)$$

Where $\mathbf{A_b}$ and $\mathbf{A_u}$ are the combustion chamber wall surface areas in contact with burned and unburned gases, respectively, and $\mathbf{T_W}$ is the cylinder wall temperature. The surface areas are computed as function of cylinder bore D and instantaneous combustion chamber volume **V** by [٢]: -

$$A_b = (\frac{\pi D^2}{2} + \frac{4V}{D}) x^{\frac{1}{2}} \qquad (٣.٢٢٢)$$

$$A_u = (\frac{\pi D^2}{2} + \frac{4V}{D})(1 - x^{\frac{1}{2}}) \qquad (٣.٢٢٣)$$

Equations (٣.٢٢٢), and (٣.٢٢٣) are empirical functions that have the correct limits in the case of a cylinder where x=٠ and x=١.

The instantaneous convective heat transfer coefficient **h** ($W/m^٢ . K$) Is computed from the known correlation proposed by Woschni [٣٠] as function of cylinder bore **D** (cm), pressure **P** (bar), gas temperature **T** (K) and mean gas velocity ω (m/sec).

$$h = ٣٢٦ . D^{-٠.٢} . P^{٠.٨} . T^{-٠.٥٥} . \omega^{٠.٨} \qquad (٣.٢٢٤)$$

With mean gas velocity ω related to the mean piston speed $\overline{U}_P$ and combustion phenomena:

$$\omega = \left[C_1 \overline{U}_P + C_2 \frac{V T_O}{P_O V_O} \Delta P_C \right] \qquad (٣.٢٢٥)$$

Where $\mathbf{T_O}$, $\mathbf{P_O}$ and $\mathbf{V_O}$ are the temperature, pressure and volume at inlet valve closing, respectively, ΔP_C is the instantaneous pressure rise due to combustion. $\overline{U}_P = 2L\,rps$ (Mean piston speed). The constants $\mathbf{C_1}$ and $\mathbf{C_2}$ are derived from Woschni [٣٠] and equal to ٢.٢٨ and ٣.٢٤ 10^{-3} respectively. The enthalpy loss between piston ring [٢٧]: -

$$h_\ell = x h_b + (1-x) h_u \qquad (٣.٢٢٦)$$

And the change of cylinder mass m with crank angle is : -

$$m = m_1 \exp\left[-\frac{C(\theta-\theta_1)}{360 rps}\right] \qquad (٣.٢٢٧)$$

Where $\mathbf{m_1}$ is the initial mass at $\theta = \theta_1$ (start of combustion)

For the heat release law the wiebe function [١١] can be applied to compute the burned mass function $x(\theta)$:

$$x = \begin{cases} 0 & \theta < \theta_S \\ 1-\exp\left[-n\left(\frac{\theta-\theta_S}{\theta_b}\right)^{m+1}\right] & \theta < \theta < \theta_S + \theta_b \\ 1 & \theta > \theta_S + \theta_b \end{cases} \qquad (٣.٢٢٨)$$

Where θ_S is the ignition crank angle and θ_b is the combustion angular period. In accordance with wiebe function [١١] the values **n** and **m** has been fixed to a constant values ٥ and ٢, respectively.

The derivatives $\frac{dV}{d\theta}$ and $\frac{dx}{d\theta}$ in equation (٣.٢١٦) Are known by differentiating (٣.١٠٥) and (٣.٢٢٨), respectively as follows,

$$\frac{dV}{d\theta} = \frac{V_C}{2}(CR-1)\left[\sin\theta + \frac{\sin\theta\cos\theta}{(R^2-\sin^2\theta)^{\frac{1}{2}}}\right]$$

(٣.٢٢٩)

And,

$$\frac{dx}{d\theta} = \frac{15}{\theta_b}\left(\frac{\theta-\theta_S}{\theta_b}\right)^2 \exp -5\left(\frac{\theta-\theta_S}{\theta_b}\right)^3 \qquad (٣.٢٣٠)$$

Differentiating equation (٣.٢٠٧) with respect to crank angle θ, and substitute equations (٣.٢٠٩) and (٣.٢١٠), gives : -

$$\frac{1}{m}\frac{dV}{d\theta} - \frac{V}{m^2}\frac{dm}{d\theta} = x\frac{d\nu_b}{d\theta} + (1-x)\frac{d\nu_u}{d\theta} + (\nu_b - \nu_u)\frac{dx}{d\theta}$$

Then,

$$\frac{1}{m}\frac{dV}{d\theta} + \frac{CV}{360mrps} = x\frac{\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\frac{dT_b}{d\theta} + (1-x)\frac{\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}\frac{dT_u}{d\theta} + \left[x\frac{\nu_b}{P}\frac{\partial \ln \nu_b}{\partial \ln P} + (1-x)\frac{\nu_u}{P}\frac{\partial \ln \nu_u}{\partial \ln P}\right]\frac{dP}{d\theta} + (\nu_b - \nu_u)\frac{dx}{d\theta} \qquad (٣.٢٣١)$$

Treating the unburned gas as an open system losing mass via leakage between piston rings, and using the entropy of unburned gas [٢٧] gives: -

$$-Q_u = 360\,rps\,m(1-x)T_u\frac{dS_u}{d\theta} \qquad (٣.٢٣٢)$$

The entropy relating to temperature and pressure must be introduced.

$$S_u = S_u(T_u, P) \qquad (٣.٢٣٣)$$

Differentiating equation (٣.٢٣٣) with respect to crank angle θ, we get

$$\frac{dS_u}{d\theta} = \left(\frac{\partial S_u}{\partial T_u}\right)\frac{dT_u}{d\theta} + \left(\frac{\partial S_u}{\partial P}\right)\frac{dP}{d\theta} \qquad (٣.٢٣٤)$$

From relations $\left(\frac{\partial S_u}{\partial T_u}\right) = \frac{C_{P_u}}{T_u}$ and $\left(\frac{\partial S_u}{\partial P}\right) = -\left(\frac{\partial \nu_u}{\partial T_u}\right)$

And $\frac{\partial \nu_u}{\partial T_u} = \frac{\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}$ rearranging gives [٣٦] : -

$$\frac{dS_u}{d\theta} = \left(\frac{C_{P_u}}{T_u}\right)\frac{dT_u}{d\theta} - \frac{\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}\frac{dP}{d\theta} \qquad (٣.٢٣٥)$$

Substitute equation (٣.٢٣٥), and (٣.٢٢١) into equation (٣.٢٣٢), gives : -

$$C_{P_u}\frac{dT_u}{d\theta} - \nu_u\frac{\partial \ln \nu_u}{\partial \ln T_u}\frac{dP}{d\theta} = \frac{-h_u(A_u)}{360\,rps\,m}(T_u - T_W) \qquad (٣.٢٣٦)$$

The ordinary differential equation of unburned gas with crank angle from equation (٣.٢٣٧) is : -

$$\frac{dT_u}{d\theta} - \nu_u \frac{\partial \ln \nu_u}{\partial \ln T_u}\frac{dP}{d\theta} = \frac{-h_u(A_u)}{360\,rps\,C_{P_u}}(T_u - T_W) + \frac{\nu_u}{C_{P_u}}\frac{\partial \ln \nu_u}{\partial \ln T_u}\frac{dP}{d\theta} \qquad (٣.٢٣٧)$$

And for burned gas, let $B = \frac{dT_u}{d\theta}$, then

$$\frac{dT_b}{d\theta} = \left[\frac{\frac{1}{m}\frac{dV}{d\theta} + \frac{CV}{360mrps} - (1-x)\frac{\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}B + \left[x\frac{\nu_b}{P}\frac{\partial \ln \nu_b}{\partial \ln P} + (1-x)\frac{\nu_u}{P}\frac{\partial \ln \nu_u}{\partial \ln P}\right]\frac{dP}{d\theta} + (\nu_b - \nu_u)\frac{dx}{d\theta}}{\left(x\frac{\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\right)}\right]$$

(٣.٢٣٨)

The ordinary differential equation of pressure change with respect to crank angle, is obtained by substituting equations, (٣.٢٣٨), (٣.٢٣٧), (٣.٢١٧), (٣.٢٢٦), (٣.٢١٨), and (٣.٢١٩) in equation (٣.٢٠٥) and after rearranging gives : -

$$\frac{dP}{d\theta} = \frac{AA + BB + CC}{DD + EE + FF} \qquad (٣.٢٣٩)$$

Where,

$$AA = -mx(C_{P_b} - \frac{P\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b})$$

$$\left[\frac{\left(\frac{1}{m}\frac{dV}{d\theta} + \frac{VC}{360\,rps\,m}\right) + \left((1-x)\frac{\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}\right)\left(\frac{h_u A_u (T_u - T_W)}{360\,rps\,m\,C_{P_u}(1-x)}\right) + (\nu_b - \nu_u)\frac{dx}{d\theta}}{(x\frac{\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b})}\right]$$

$$BB = \left[\begin{array}{l}\left(m(1-x)(C_{P_U} - \frac{P\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u})\right)\left(\frac{h_u A_u (T_u - T_W)}{360\,rps\;C_{P_U}(1-x)}\right) \\ -m(u_b - u_u)\frac{dx}{d\theta}\end{array}\right]$$

$$CC = (xu_b + (1-x)u_U)\frac{Cm}{360\,rps} - \frac{1}{360\,rps}(h_b A_b (T_b - T_W) +$$

$$h_u A_u (T_u - T_W)) + \frac{Cm}{360\,rps}\left[xh_b + (1-x)h_u\right] - P\frac{dV}{d\theta}$$

$$DD =$$

$$\left[\begin{array}{l}-mx\left(C_{P_b} - \frac{P\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\right)\left((1-x)\frac{\nu_u^{\,2}}{C_{P_U}T_u}(\frac{\partial \ln \nu_u}{\partial \ln T_u})^2\right) + \\ \frac{\left(mx\left(C_{P_b} - \frac{P\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\right)\left(x\frac{\nu_b}{P}\frac{\partial \ln \nu_b}{\partial \ln P}\right) + (1-x)\frac{\partial \ln \nu_b}{\partial \ln P}\right)}{\left(x + \frac{\nu_b}{T_b}\frac{\partial \ln \nu_b}{\partial \ln T_b}\right)}\end{array}\right]$$

$$EE = \left[m(1-x)\left(C_{P_U} - \frac{P\nu_u}{T_u}\frac{\partial \ln \nu_u}{\partial \ln T_u}\right)\left(\frac{\nu_u}{C_{P_U}}\frac{\partial \ln \nu_u}{\partial \ln T_u}\right)\right]$$

And,

$$FF = -\left[mx\nu_b\left(\frac{\partial \ln \nu_b}{\partial \ln T_b} + \frac{\partial \ln \nu_b}{\partial \ln P}\right) + m(1-x)\nu_u\left(\frac{\partial \ln \nu_u}{\partial \ln T_u} + \frac{\partial \ln \nu_u}{\partial \ln P}\right)\right]$$

If the heat transfer coefficient of burned and unburned gas are equals, i.e, $h_b=h_u=h$.

The equation of work and heat loss as follows,

$$\frac{dW}{d\theta} = P\frac{dV}{d\theta} \qquad (٣.٢٤٠)$$

And,

$$\frac{dQ}{d\theta} = \frac{h}{360\,rps}\left(\frac{\pi D^2}{2} + \frac{4V}{D}\right)\left[x^{\frac{1}{2}}A_b(T_b - T_W) + (1 - x^{\frac{1}{2}})A_u(T_u - T_W) \right] \qquad (٣.٢٤١)$$

Equations, (٣.٢٣٩), (٣.٢٣٨), (٣.٢٣٧), (٣.٢٤٠), and (٣.٢٤١) are integrated using high order Runge-Kutta method as follows,

$$Y_{i+1} = Y_i + \frac{\Delta\theta}{6}\left[k_1 + 2k_2 + 2k_3 + k_4\right] \qquad (٣.٢٤٢)$$

$$\frac{dY}{d\theta} = f(Y,\theta)$$

$$k_1 = f(Y_i,\theta)$$

$$k_2 = f\left(Y_i + \frac{k_1}{2}, \theta_i + \frac{\Delta\theta}{2}\right)$$

$$k_3 = f\left(Y_i + \frac{k_2}{2}, \theta_i + \frac{\Delta\theta}{2}\right)$$

$$k_4 = f(Y_i + k_3, \theta_i + \Delta\theta)$$

Where Y is any variable of equations as before.

٣.١٠.b.١: Flame Front Propagation.

After ignition and flame development, the combustion process is well established and the flame moves very quickly through the combustion chamber. The flame front, which would expand spherically from the spark plug in stationary mixture. Unburned temperature and

pressure at combustion process influence the flame speed. The temperature and pressure of unburned gas is raised to maximum value at the end of combustion process. Useful correlation of flame speed with unburned gas temperature and pressure for various fuel- air mixtures associated with reciprocating internal combustion engine is follows [١١], [٤], [٣٨]

$$S_u = S_o\left(\frac{T_u}{T_{ref}}\right)^{\alpha}\left(\frac{P}{P_{ref}}\right)^{\varphi}(1-2.1f) \qquad (٣.٢٤٣)$$

Where:

f: Residual mass fraction

S_u: flame speed (cm/sec)

S_O: reference flame speed (cm/sec)

T_u: unburned gas temperature (K)

P: pressure inside the cylinder (atm)

T_{ref}, P_{ref} temperature and pressure at reference condition

The temperature and pressure exponents α and φ are independent of fuel types within the estimated experiments and can be represented by the expression,

$$\alpha = 2.18 - 0.8(\phi - 1)$$

And

$$\varphi = -0.16 + 0.22(\phi - 1)$$

The reference velocities S_O are a weak function of fuel type and can be fit by a second-order polynomial of the form.

$$S_o = B_m + B_2(\varphi - \varphi_m)^2$$

B_m, B_2 and ϕ_m are constants for a given fuel, and their values are given in table (٣.١) for several types for fuel [٣٨].

Table (٣.١): -Constant values for S_o

Fuels	Formula	ϕ_m	$B_m(\frac{cm}{sec})$	$B_2(\frac{cm}{sec})$
Methanol	CH_4O	١.١١	٣٦.٩٢	-١٤٠.٥١
Propane	C_3H_8	١.٠٨	٣٤.٤٤	-١٣٨.٠٥
Isooctane	C_8H_{18}	١.١٣	٢٦.٣٢	-٨٤.٧٢
RMFD-٣٠٣	C_7H_{17}	١.١٣	٢٧.٥٨	-٧٨.٣٤

٣.١٠.b.٢ *Time of Flame Propagation.*

The burn angle is the angle through which the crankshaft turns during combustion. If combustion is to be completed at ATDC then ignition occurs at BTDC, the combustion period is normally in the range of ٤٠^o to ٦٠^o of crankshaft rotation, if the engine is running at any rotational speed, the time of flame propagation is [٤]

$$\text{time} = \theta_b{}^o \frac{1\text{min}}{\text{revolution}} \frac{1\text{revolution}}{360^o} \frac{60\text{sec}}{1\text{min}} \qquad (٣.٢٤٤)$$

٣.١٠.C:*The Expansion Period Can Be:*

The two-zone expansion starts from TDC, until end of combustion ATDC, because the combustion period finish for maximum crank angles ٢٠^o ATDC. The same procedure of analysis of two zone combustion in two zone expansion and the calculation in a step by step using Runge-Kutta method. Once the combustion process is complete the all variable are ordered to calculate for single zone only. The Runge-Kutta method is used for the same analysis for compression stroke.

٣.١١: *Engine Design and Operating Parameters*

٣.١١.١: *Indicated Work Per Cycle*

Pressure data for the gas in the cylinder over the operating cycle of the engine can be used to calculate the work transfer from the gas to the

piston. The cylinder pressure and corresponding cylinder volume throughout the engine cycle can be plotted on a p-v diagram. The indicated work per cycle w_i (per cylinder) is obtained by integrating the area enclosed on the diagram,

$$w_i = \oint PdV \qquad (٣.٢٤٥)$$

The indicated power per cylinder is related to the indicated work per cycle.

$$P_i = \frac{w_i \, rps}{n_R} \qquad (٣.٢٤٦)$$

Where n_R is the number of crank turns for each power stroke per cylinder, for four-stroke cycles, n_R equals ٢; for two-stroke cycles, n_R equals ١.

٣.١١.٣: Indicated Thermal Efficiency

The indicate thermal efficiency is the ratio of the work done to the energy content of the fuel supplied. If the heat of reaction is Q_{VS} at the standard reference temperature and the fuel rate is m_f units of mass per unit time, we can defined the indicated thermal efficiency as

$$\eta_i = \frac{\oint PdV \, rps}{m_f Q_{VS}} \qquad (٣.٢٤٧)$$

٣.١١.٣: Indicated Mean Effective Pressure

The indicated mean effective pressure imep is the area of the indicated diagram divided by the swept volume V_S. Hence

$$imep = \frac{\oint PdV}{V_{TDC} - V_{BDC}} \qquad (٣.٢٤٨)$$

٣.١١.٤: Indicated Specific Fuel Consumption

A more useful parameter is the specific fuel consumption (isfc) is indicated fuel flow rate per unit power output,

$$isfc = \frac{m_f}{P_i} \qquad (٣.٢٤٩)$$

٣.١١.٥: Indicated Torque

Another parameter is indicated torque is defined by divided the indicated power to angular velocity of crankshaft, as

$$T_i = \frac{P_i}{\omega} \qquad (٣.٢٥٠)$$

Where ω is the angular velocity (rad/sec)?

٣.١١.٦: Friction Mean Effective Pressure

The total friction mean effective pressure can be quite accurately be related to engine speed by the empirical equation [١١] and [٤]

$$fmep = A + B(rpm/١٠٠٠) + C(rpm/١٠٠٠)^{٢} \quad \text{(bar)} \qquad (٣.٢٥١)$$

Where : - **A, B,** and **C** are empirical constants related to a specific engine and their values are ٠.٩٧**,** ٠.١٥**,** and ٠.٠٥**,** respectively.

Work delivered by the crankshaft is less than indicated work due to mechanical friction and loads of the engine. Loads include oil pump, supercharger, air conditioner compressor, alternator, etc.

The actual work available at the crankshaft is called brake work W_b.

$$W_b = W_i - W_f \qquad (٣.٢٥٢)$$

Where W_f is the specific work lost due to friction and parasitic loads. And all parameters in indicated relations can be used to calculate the brake relations, like brake power, brake thermal efficiency (total efficiency), brake mean effective pressure (bmep), brake specific fuel consumption, and brake torque.

The normal operating range for a conventional SI engine using gasoline $١٢ \le A/F \le ١٨$.

٣.١١.٧: Volumetric Efficiency

Another performance parameter of importance is the volumetric efficiency, η_V, it is defined as the mass of fuel and air induced into the cylinder divided by the mass that would occupy the displaced volume at the density ρ_i in the intake manifold,

$$\eta_V = \frac{n_R (m_a + m_f)}{\rho_i V_S \, rps} \qquad (٣.٢٥٣)$$

Where:

m_a = mass flow of air into the engine (or cylinder) for one cycle.

m_f = mass flow of fuel into the engine (or cylinder) for one cycle.

ρ_i = Mixture density evaluated at atmospheric conditions outside the engine.

V_S = displacement volume.

rps = engine speed.

n_R = number of revolutions per cycle.

$$m_f = \frac{\phi F_S (1-\beta)}{1+\phi F_S}(m_f + m_a) \qquad (٣.٢٥٤)$$

Where:

$F_S = (F/A)_{St}$

f=Residual mass fraction from previous cycle.

Substitute equation (٣.٢٥٤) into equation (٣.٢٤٦), gives : -

$$\eta_i = \frac{w_i (1+\varphi F_S)}{(m_f + m_a)\varphi\varphi_S (1-f) Q_{VS}} \qquad (٣.٢٥٥)$$

And

$$\eta_b = \frac{w_b (1+\varphi F_S)}{(m_f + m_a)\varphi\varphi_S (1-f) Q_{VS}} \qquad (٣.٢٥٦)$$

٣.١٢: Relationships Between Performance Parameters

The following relationships between engine performance parameters can be developed:

$$\text{power} = \frac{\eta m_a \, \text{rps} \, Q_{VS} (F/A)}{n_R} \qquad (٣.٢٥٧)$$

For four-stroke cycle, volumetric efficiency can be introduced,

$$P = \frac{\eta \eta_V \, \text{rps} \, V_S \, Q_{VS} \, \rho_i (F/A)}{2} \qquad (٣.٢٥٨)$$

For torque T:

$$T = \frac{\eta \eta_V \;\; V_S \, Q_{VS} \, \rho_i (F/A)}{2\pi} \qquad (٣.٢٥٩)$$

For mean effective pressure:

$$\text{mep} = \eta \eta_V \;\; Q_{VS} \, \rho_i (F/A) \qquad (٣.٢٦٠)$$

CHAPTER 4

COMPUTER PROGRAM

٤.١: Introduction

In this chapter , an iterative quick basic computer programs are developed to calculate pressure and temperature distribution in cylinder. Prediction of the performance of the ٤-stroke, spark ignition engine is performed. These calculations are based on the theoretical analysis presented in chapter three.

٤.٢: Input Data

The input data required for running the programs are as follows: -

-Number of cylinders

-Cylinder bore

-Cylinder stroke

-Compression ratio

-Connecting rod length

-Crank radius

-Piston surface area/bore area

-Head surface area

-Inlet temperature

-Inlet pressure

-Reference temperature

-Reference pressure

-Air-fuel ratio

-Engine speed

-Crank angle steps or output

-Angle of advance

-Combustion duration in crank angle degrees

-Cylinder wall temperature

fuel identification : -

-Polynomial coefficient of properties of fuel

-Percentage carbon content in fuel

-Percentage hydrogen content in fuel

٤.٣: Programs Output

-Temperature, pressure and volume are all listed versus crank angle

-Mean effective pressure

-Torque

-Output power

-Specific fuel consumption

-Thermal efficiency

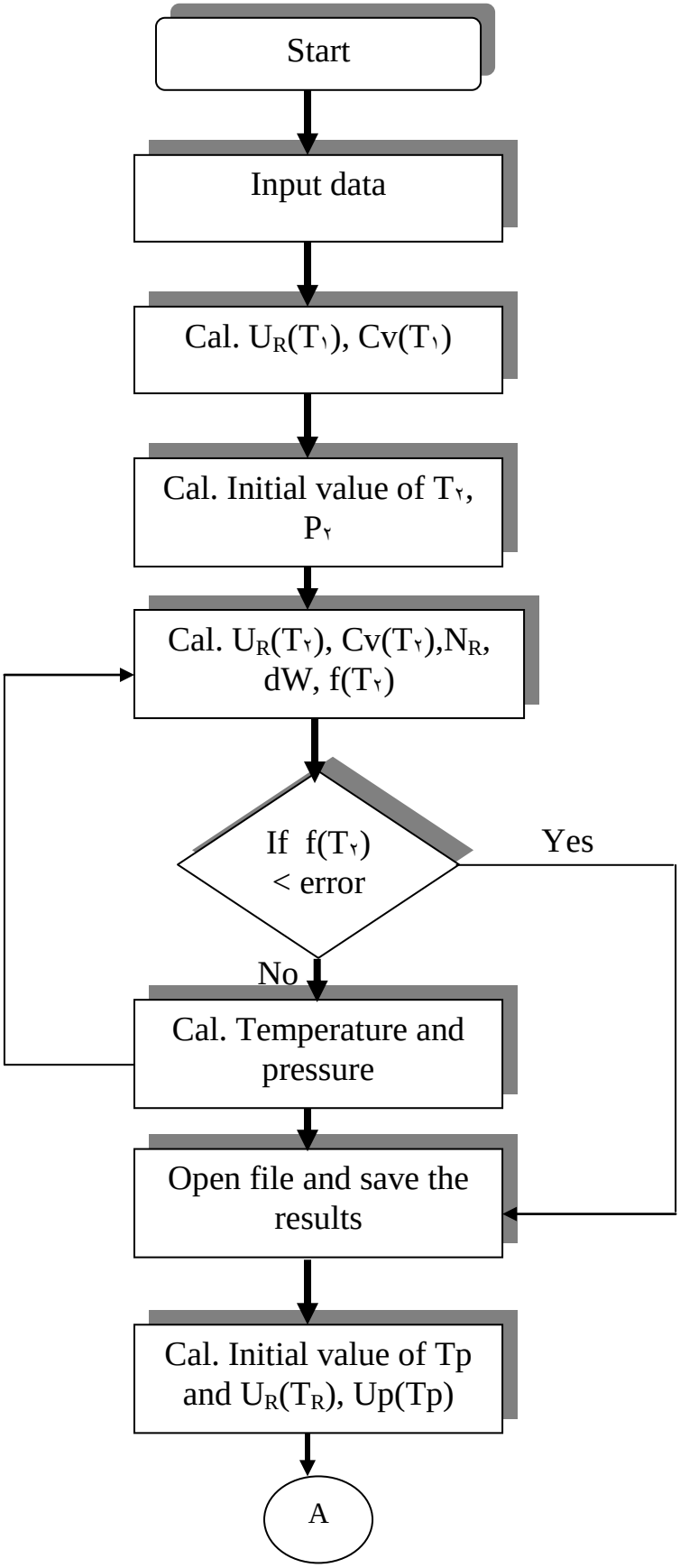
-Mechanical efficiency

-Volumetric efficiency

-Flame speed

٤.٤: Programs Layout

There are two main programs: ideal and real. The ideal program with the flow chart shown in figure (٤.١). the real program with the flow chart shown in figure (٤.٢).The ideal take the input data and computational is done step by step using Newton-Raphson methods, and check the results. The real take the input data and computational is done step by step using Runge-Kutta methods, and then check the results.


Start
Input data
Cal. $U_R(T_1)$, $Cv(T_1)$
Cal. Initial value of T_2, P_2
Cal. $U_R(T_2)$, $Cv(T_2)$,N_R, dW, $f(T_2)$
If $f(T_2)$ < error
Yes
No
Cal. Temperature and pressure
Open file and save the results
Cal. Initial value of Tp and $U_R(T_R)$, Up(Tp)
A

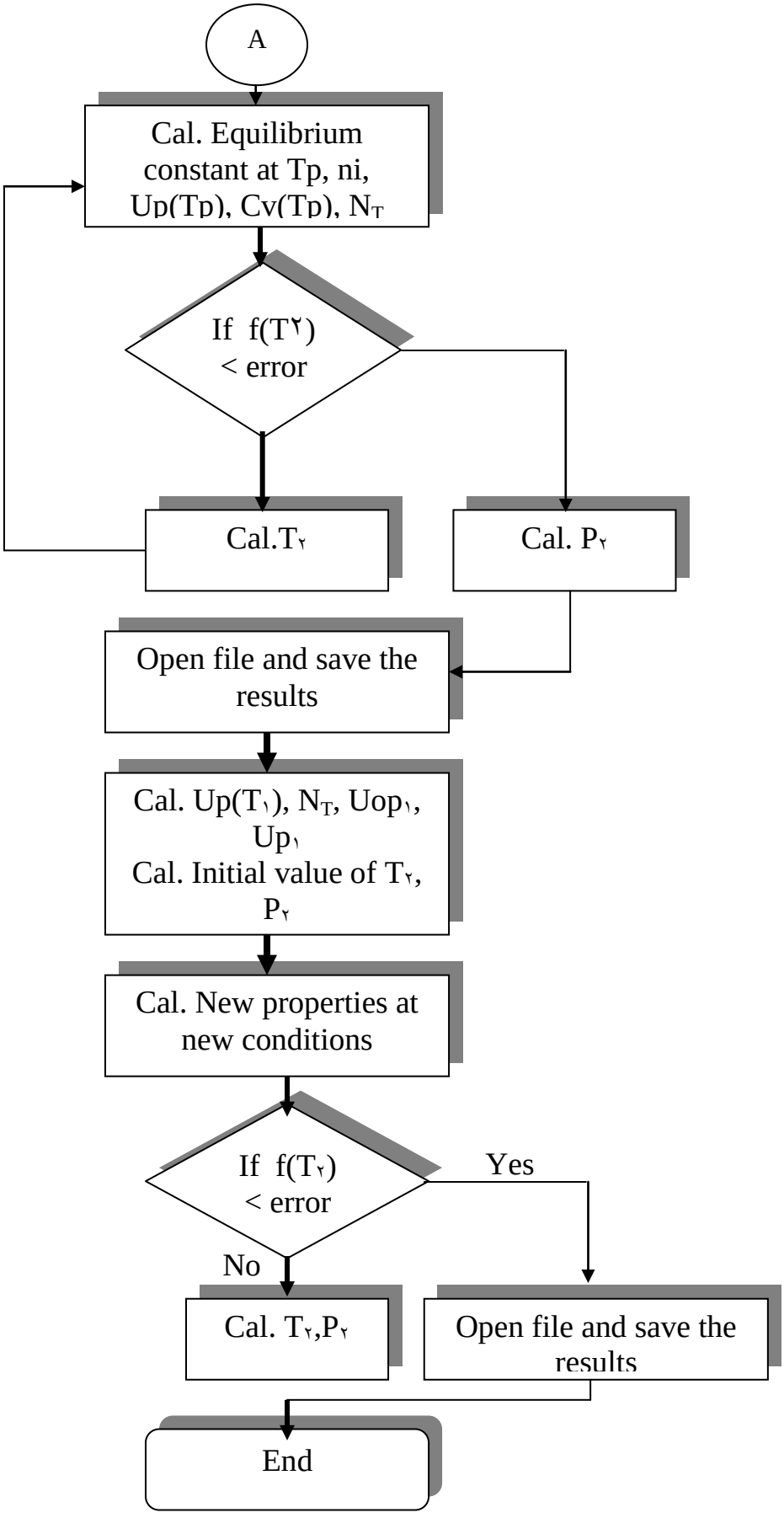
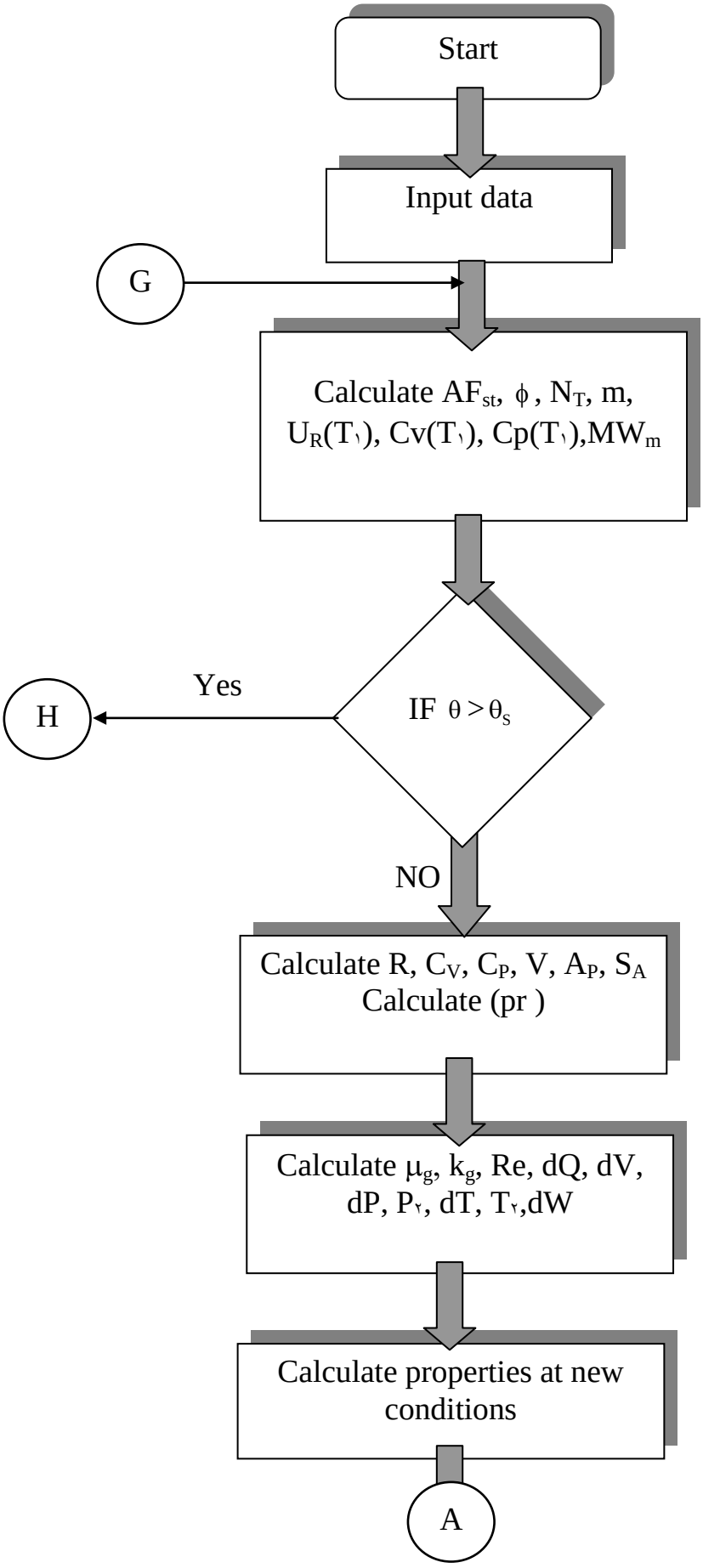
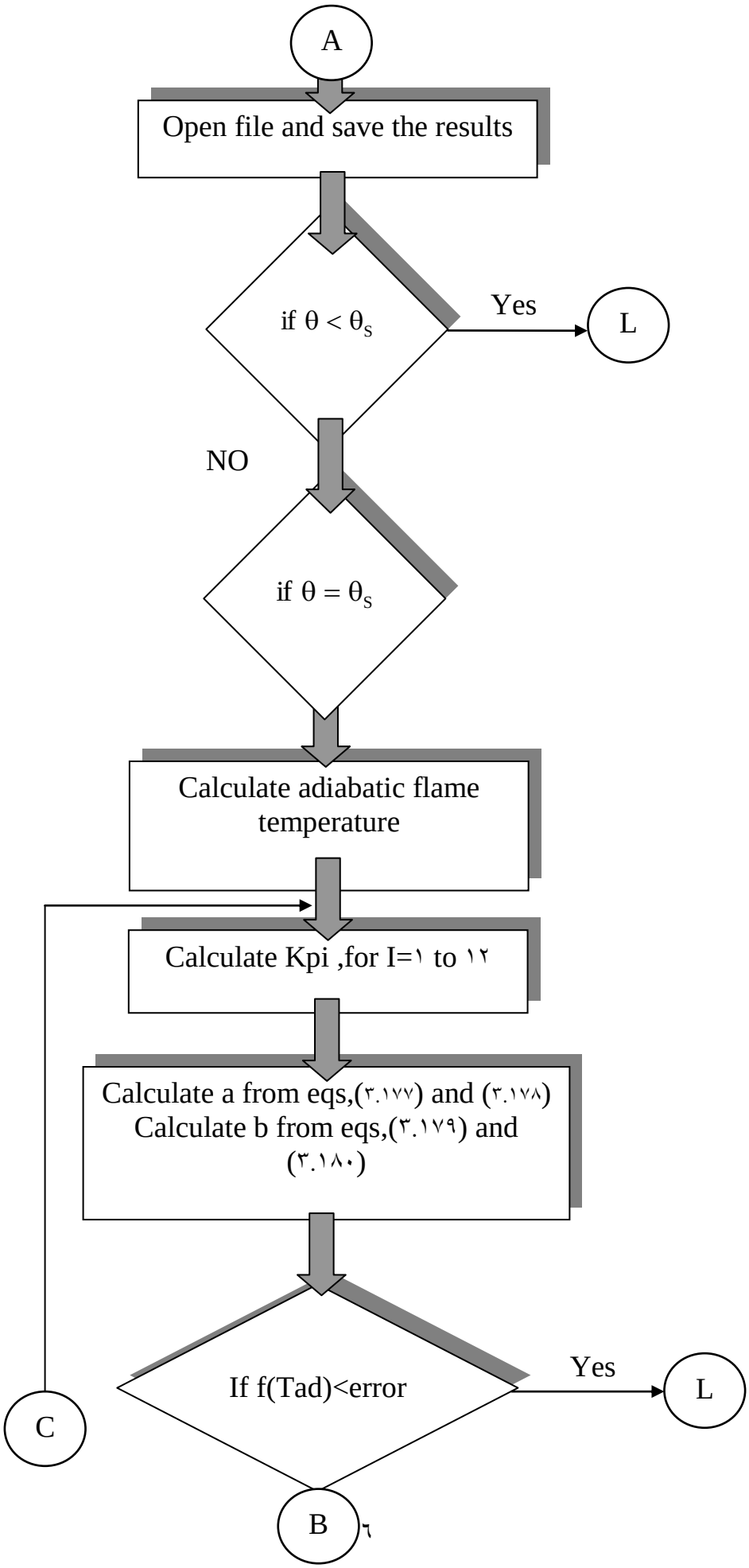
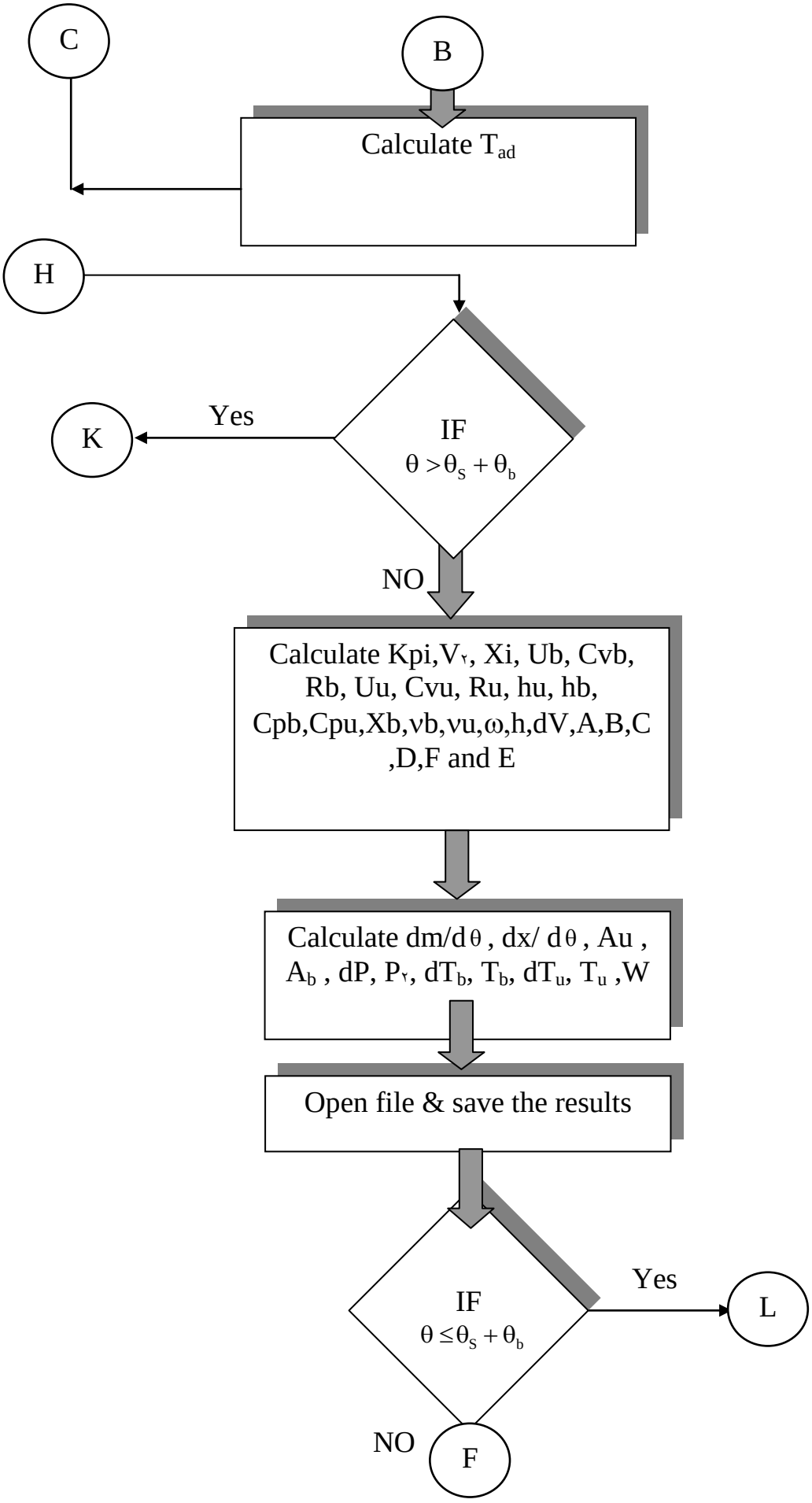


Fig.(٤.١): Flow chart of ideal main program.


Start
Input data
G
Calculate AF_{st}, ϕ, N_T, m,
$U_R(T_1)$, $Cv(T_1)$, $Cp(T_1)$,MW_m
IF $\theta > \theta_S$
Yes
H
NO
Calculate R, C_V, C_P, V, A_P, S_A
Calculate (pr)
Calculate μ_g, k_g, Re, dQ, dV,
dP, P_2, dT, T_2,dW
Calculate properties at new conditions
A


A
Open file and save the results
if θ < θS
Yes
L
NO
if θ = θS
Calculate adiabatic flame temperature
Calculate Kpi ,for I=١ to ١٢
Calculate a from eqs,(٣.١٧٧) and (٣.١٧٨)
Calculate b from eqs,(٣.١٧٩) and (٣.١٨٠)
If f(Tad)<error
Yes
L
C
B


C
B
Calculate T_{ad}
H
IF
$\theta > \theta_S + \theta_b$
Yes
K
NO
Calculate Kpi,V۲, Xi, Ub, Cvb, Rb, Uu, Cvu, Ru, hu, hb, Cpb,Cpu,Xb,νb,νu,ω,h,dV,A,B,C ,D,F and E
Calculate dm/dθ , dx/ dθ , Au , A_b , dP, P۲, dT_b, T_b, dT_u, T_u ,W
Open file & save the results
IF
$\theta \leq \theta_S + \theta_b$
Yes
L
NO
F

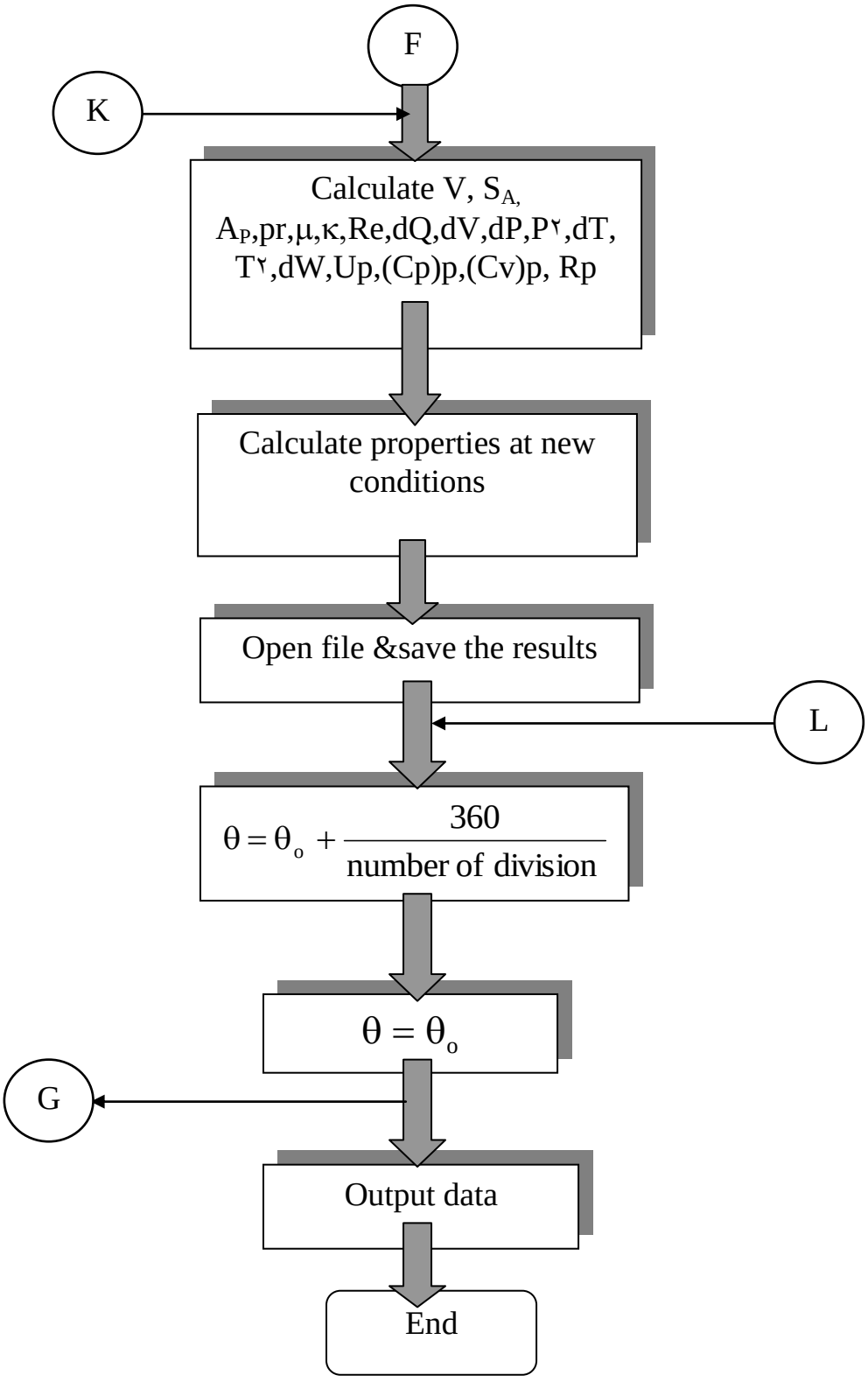


Fig.(٤.٢): Flow chart of real main program.

CHAPTER 5

RESULTS AND DISCUSSION

٥.١: Introduction

The results presented in this chapter were obtained by running the computer program written to simulate the engine with the input data mentioned in article (٤.٢) and optimum engine design parameters as shown in tables [١] and [٢].

٥.٢: The Effect of Mixture Strength and Load on Ideal Cycle .

Figures (٥.١) , (٥.٢) and (٥.٣) show the variation of the cylinder inside pressure with cylinder volume (ideal . The maximum pressure increases as the load increases , due to increase in mass flow rate. The cylinder pressure is found to be maximum when the equivalence ratio equal ١ Fig.(٥.٢), where the combustion process is perfect. This cycle show the pressure , and volume distributions in the cylinder through cycle processes. From this results it can be concluded that the process is polytropic (PV^n= C) , for compression and expansion . The constant **C** changes with load and mixture strength . This cycle was used to study the behavior of ideal engine . So equivalence ratio is chosen to be (١) for optimum operation.

٥.٣: The Effect of Ignition Timing.

Figure (٥.٤) shows real diagram the pressure-volume variations in spark timing. If combustion starts too early in the cycle , the work transfer from the piston to the gases in the cylinder at the end of the compression stroke is too large , if combustion starts too late , the peak cylinder pressure is reduced and the expansion stroke work transfer from gas to the piston decreases . There exists a particular spark timing which gives maximum power and minimum specific fuel consumption . Spark timing affects peak cylinder pressure and therefore peak unburned and burned gas temperatures . Retarding spark timing from the optimum reduces these variables . Retarding timing is some times used therefor for NOx emission control and to avoid knock . The decrease in maximum pressure when correspond with ideal maximum pressure is due to heat loss from combustion chamber and loss time available for combustion . So an advance angle of ٤٠^{o} BTDC is optimum.

Figure (٥.٥) shows the variation of cylinder pressure with crank angle .the maximum cylinder pressure P_{max} ; the crank angle at which this maximum pressure occurs θ_{max} . The optimum spark setting will depend on the rate of flame development and propagation , the length of the flame travel path across combustion chamber , and the details of the flame termination process after it reaches the wall . These depend on engine design and operating conditions and the properties of the fuel , air , burned gas mixture , The evolution of maximum pressure is at a crank angle ٢٠٠ ATDC, corresponding to angle of advance of ٤٠^{o} BTDC.

٥.٢: The Effect of Compression Ratio.

the higher is the compression ratio the higher is the pressure and temperature in the end gas during normal conditions ; this increases the tendency to knock for a given engine setting and fuel there will be a critical compression ratio above which knock occurs . This compression ratio is called the highest useful compression ratio (HUCR) . For isooctease the (HUCR) is between (٩-١٠) . for different mixture strengths , therefore , a compression ratio of (٨)is chosen to avoid engine knock .

Figures (٥.٦) and (٥.٧) show the variation of mean pressure with equivalence ratio . The mean pressure is proportional to the product equivalence ratio and thermal efficiency . This exhibits a maximum for equivalence ratio equal one , for equivalence ratio less than the value corresponding to this maximum , the decrease in fuel mass per unit displaced volume more than offsets , the increasing thermal efficiency . For equivalence ratio greater than this value , the decreasing thermal efficiency (due to decreasing combustion efficiency) more than offset , the increasing fuel mass . The difference of results between brake and indicated (due to friction loss) .

Figures (٥.٨) and (٥.٩) show the variation of torque with equivalence ratio . The maximum torque is found for equivalence ratio equal one . For equivalence ratio less than unity , decreasing fuel mass , will increasing thermal efficiency .

For equivalence ratio less than unity , the fuel mass decreases, therefore , the increasing thermal efficiency . For equivalence ratio greater than unity , the fuel mass increases per unit displaced volume , will decrease thermal efficiency (due to lack of sufficient air for complete oxidation of the fuel).

Figures (٥.١٠) and (٥.١١) show the variation of power with equivalence ratio . the maximum power is for equivalence ratio equal one . For weaker mixture the power is less than the value corresponding to the maximum values of power . For richer mixture the power is also less than maximum values .

Figures (٥.١٢) and (٥.١٣) show the variation of specific fuel consumption with equivalence ratio . The specific fuel consumption minimum for equivalence ratio equal one . For rich side the specific fuel consumption increases , because the amount of fuel is greater than air , for weak side specific fuel consumption is less than air .

The specific fuel consumption is opposite to power , so the maximum specific fuel consumption is for compression ratio equal eight .

Figures (٥.١٤) and (٥.١٥) show the variation of the thermal efficiency with equivalence ratio . As the equivalence ratio is decreased below unity (i.e.; the fuel – air mixture is made progressively leaner than stiochiometric) , the efficiency increase . This occurs because the burned gas temperature after combustion decreases , decreasing the burn gas specific heats . The efficiency increase because of the given volume expansion ratio , the burn gases expand through a larger temperature to exhaust , therefore , per unit mass of fuel , the expansion stroke work is increased .

As the equivalence ratio increases above unity (i.e.; the mixture is made progressively richer) , the efficiency decreases because of lack of sufficient air for complete oxidation of the fuel , the effect is to decrease the temperature of the burned gas which decrease the mixture specific heats .

Figure (٥.١٦) shows the variation of volumetric efficiency with equivalence ratio . The volumetric efficiency is maximum for equivalence ratio equal one .

The volumetric efficiency is proportional to mass of fuel , therefore for richer and leaner mixture the volumetric efficiency is less than the value corresponding to maximum volumetric efficiency .

٥.٥: The Effect of Engine Speed , Loads and Mixture Strength .

Figures (٥.١٧) , (٥.١٨) and (٥.١٩) show the variation of mean brake pressure with engine speed . The mean brake pressure decreases when the engine speed increases (due to the increase in total friction mean effective pressure) . Therefore the higher values of mean break pressure is at full load and equivalence ratio equal one , and the lower values at part load , this is due to the change in density which is higher at full load than part load .

Figures (٥.٢٠), (٥.٢١) and (٥.٢٢) show the variation of brake torque with engine speed . The brake torque decrease when the engine speed increases , because the torque is proportional inversely to engine speed . At low speed , the torque is higher . As engine speed increases further, torque decreases because the engine is unable to ingest a full charge of air at higher speeds .

Figures (٥.٢٣) , (٥.٢٤) and (٥.٢٥) show the variation of indicated power with engine speed . The indicated power is proportional to the engine speed , The indicated power increase when the engine speed increase . The indicated power is high at full load , and lower at part load.

The figures show the higher power of equivalence ratio equal one .

Figures (٥.٢٦) , (٥.٢٧), and (٥.٢٨) show the variation of brake power with engine speed , the brake power at full load increases with increasing the engine speed (due to high power at full load) , but at part load the brake power increases with engine speed until it reaches a maximum value then it decreases , due to low power at part load and increasing friction loss .

Figures (٥.٢٩) and (٥.٣٠) show the variation of brake specific fuel consumption with engine speed . The specific fuel consumption increases with increasing engines speed , it can be shown from the figure that the lower specific fuel consumption is for Φ=٠.٩ and than rises for value above that .

Figures (٥.٣١) and (٥.٣٢) shows the variation of brake thermal efficiency with engine speed . Thermal efficiency decreases with increasing engine speed , due to increase in fuel mass per unit displaced volume . The higher thermal efficiency was found to happen at equivalence ratio equal to Φ=٠.٩ and than reduced below this value , this is due to the reasons mentioned before .

Figures (٥.٣٣) , (٥.٣٤) and (٥.٣٥) show the variation of mechanical efficiency with engine speed . The mechanical efficiency decreases with increasing engine speed due to increasing friction loss . The higher value of mechanical efficiency at fuel load and equivalence ratio equal one and the lower value of mechanical efficiency at part load .

Figures (٥.٣٦) and (٥.٣٧) show the variation of mechanical efficiency with brake power . The mechanical efficiency decreases clearly with increasing brake power , due to increasing friction loss .

٥.٦: The Effect of Ambient Temperature .

Figures (٥.٣٨) , (٥.٣٩) , (٥.٤٠) ,(٥.٤١) , (٥.٤٢) , (٥.٤٣) and (٥.٤٤) show the variation of mean break pressure , brake torque , indicated power , brake power , brake specific fuel consumption , brake thermal efficiency and mechanical efficiency with engine speed . The higher value of operating parameters at minimum inlet temperature , due to fuel mass is proportional inversely to the temperature . The lower specific fuel consumption at minimum inlet temperature because the high power output .

٥.٧: The Effect of Dissociation

Figures (٥.٤٥) , (٥.٤٦) , (٥.٤٧) (٥.٤٨), (٥.٤٩) and (٥.٥٠) show the variation of mean brake pressure , brake torque , indicated power , brake power , brake thermal efficiency and mechanical efficiency with engine speed . All variables without dissociation are higher than that with dissociation , because the dissociation is endothermic reaction which needs energy to dissociate the components into number of species , this energy will be taken from the system , However there are some conditions at which dissociation becomes important . The first is for very high flame temperatures (such as the result from near –stiochiometric combustion of highly compressed mixtures) . The second is when the product of dissociation is a pollutant such as carbon monoxide or nitrogen oxide . In these cases dissociation and chemical equilibrium can be very important .

٥.٨: The Effect of Flame Speed and Residual Gases

flames in premixed fuel , air , residual gas mixture are characterized by a flame speed . The flame speed is the velocity at which the flame propagates into quiescent premixed unburned mixture ahead of the flame .

Figures (٥.٥١) , (٥.٥٢) (٥.٥٣) (٥.٥٤) , (٥.٥٥) and (٥.٥٧) show the variation of flame speed with equivalence ratio . The flame speed peaks slightly rich of stoichiometric , and the decreases for richer and leaner mixture . Flame speed is higher at full load than part load , and the maximum flame speed decreases , by increasing the inlet temperature , due to decrease in the fuel mass per unit displaced volume). The maximum speed at residual mass fraction ٠٪ is higher than at residual mass fraction ٥٪ and ١٠٪, due to decreasing in charge mass which led to decreasing the fuel mass .

Table[١]: -Optimum engine parameters.

Cylinder bore (cm)	Stroke length (cm)	Connecting rod to crank radius	CR	Engine speed (rpm)	No. of Cylinder	Crank radius (cm)
٨	١٠	٣.٥	٨	٣٠٠٠	٤	٤

Table[٢]: -Optimum engine design.

ϕ	Maxi. Pressure (bar) atdc (٢٠˚)	Maxi. Temperature (K)	Spark advance btdc (degree)	Period of combustion (degree)	Power output (kW)
١	٦٠	٢٨٥٠	-٤٠˚	٦٠˚	٦٠

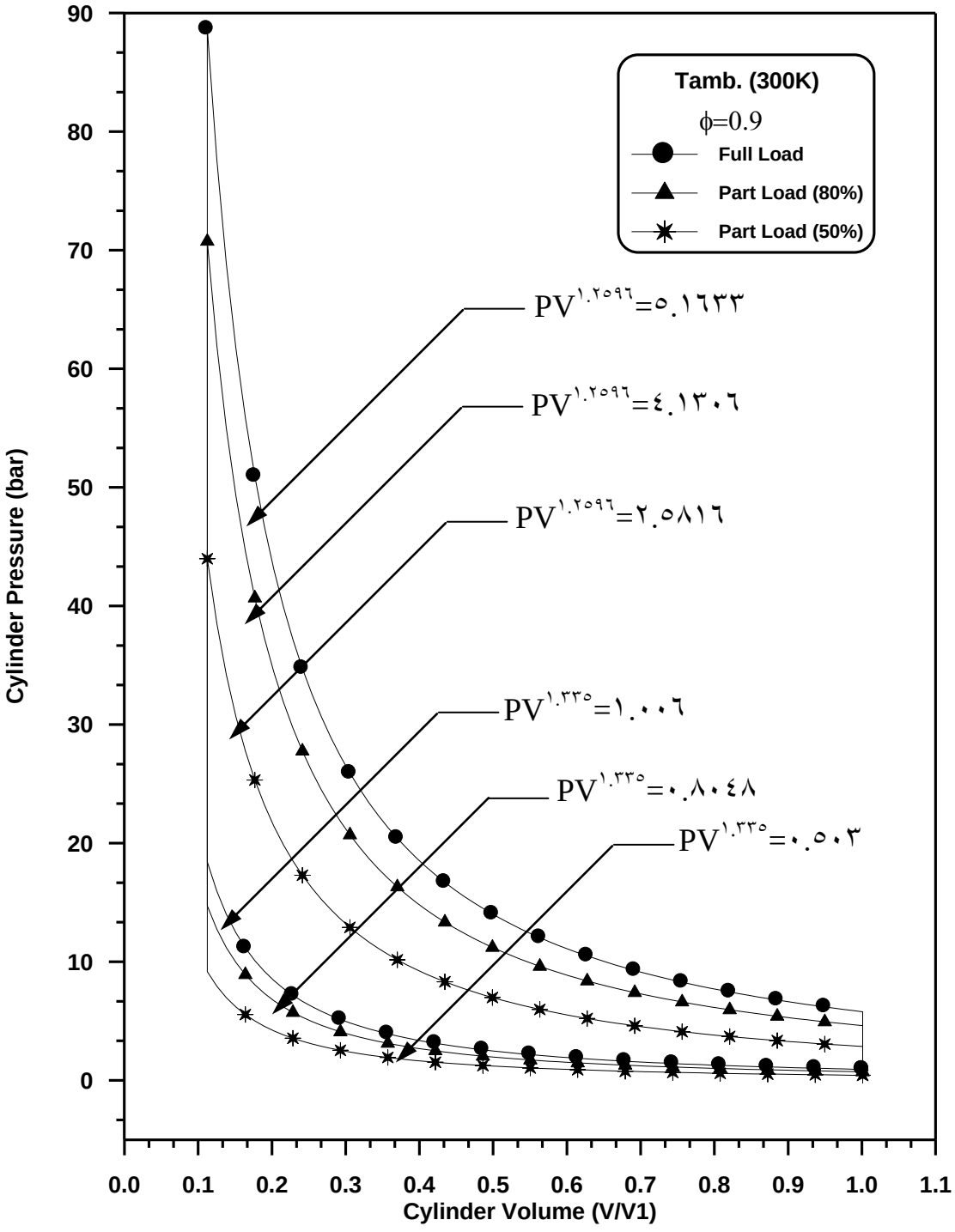


Fig. (٥.١): Pressure-Volume Ideal Diagram for Different Loads for Equivalence Ratio of ٠.٩ .

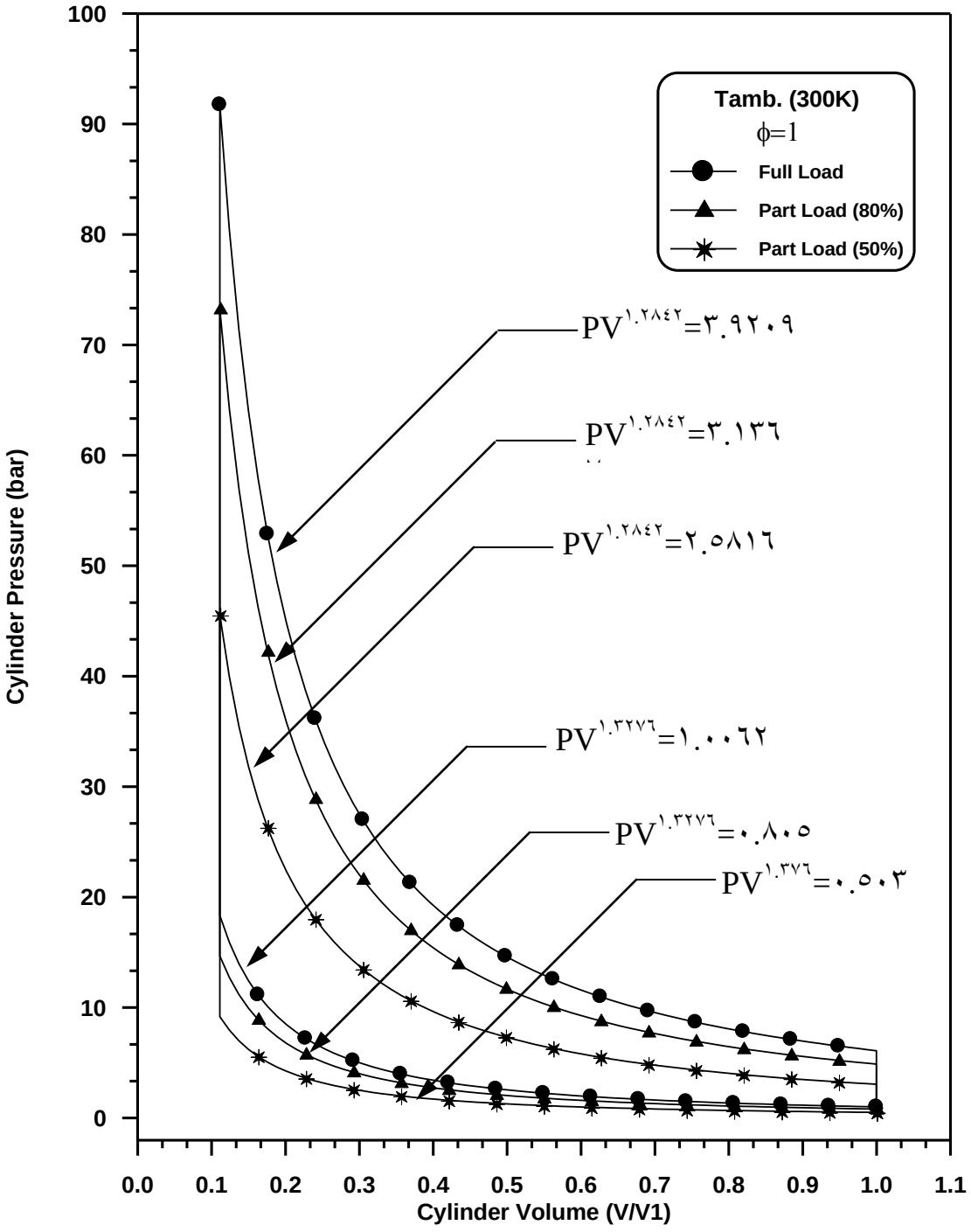


Fig. (٥.٢): Pressure-Volume Ideal Diagram for Different Loads for Equivalence Ratio of ١.

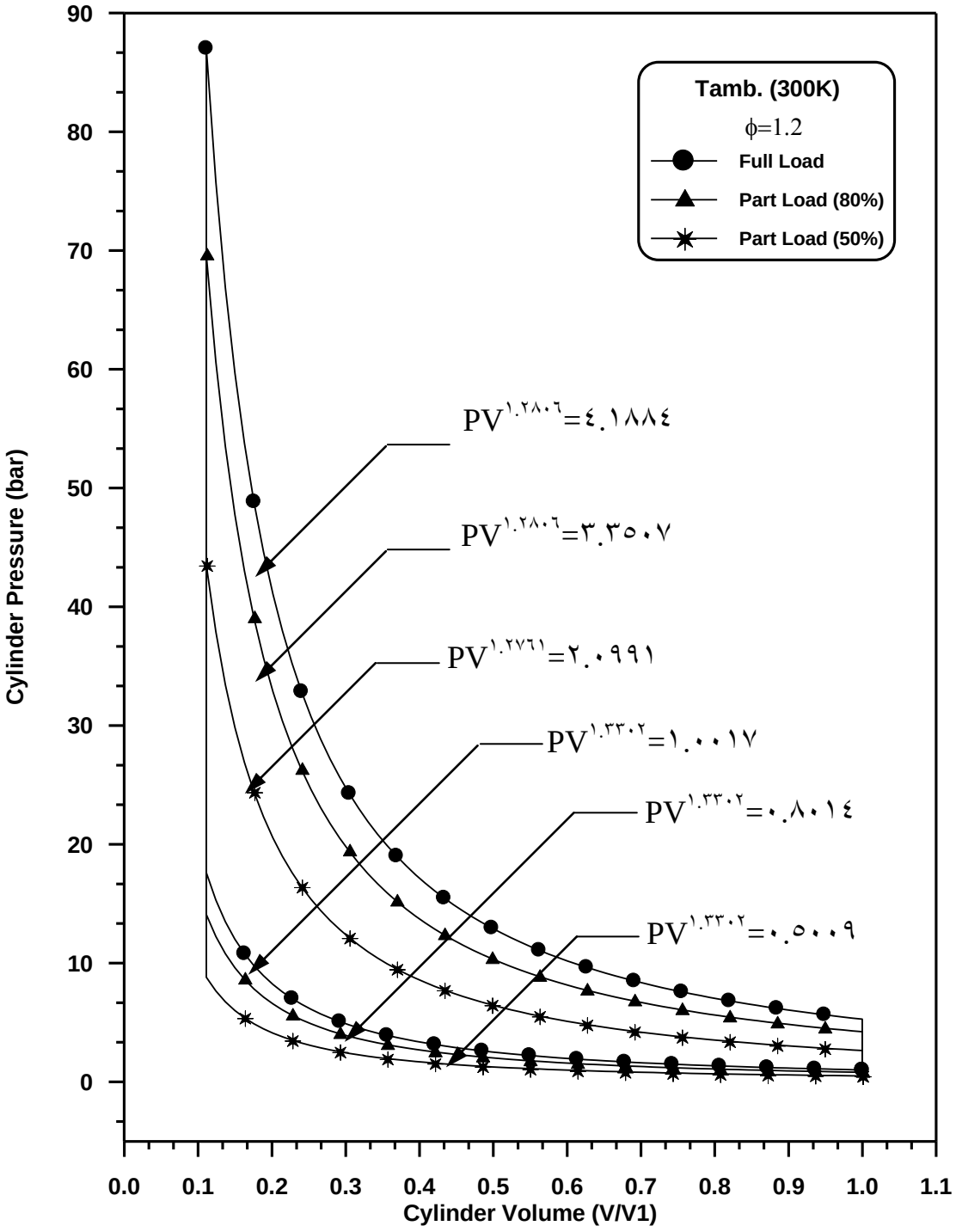


Fig. (٥.٣): Pressure-Volume Ideal Diagram for Different Loads for Equivalence Ratio of ١.٢ .

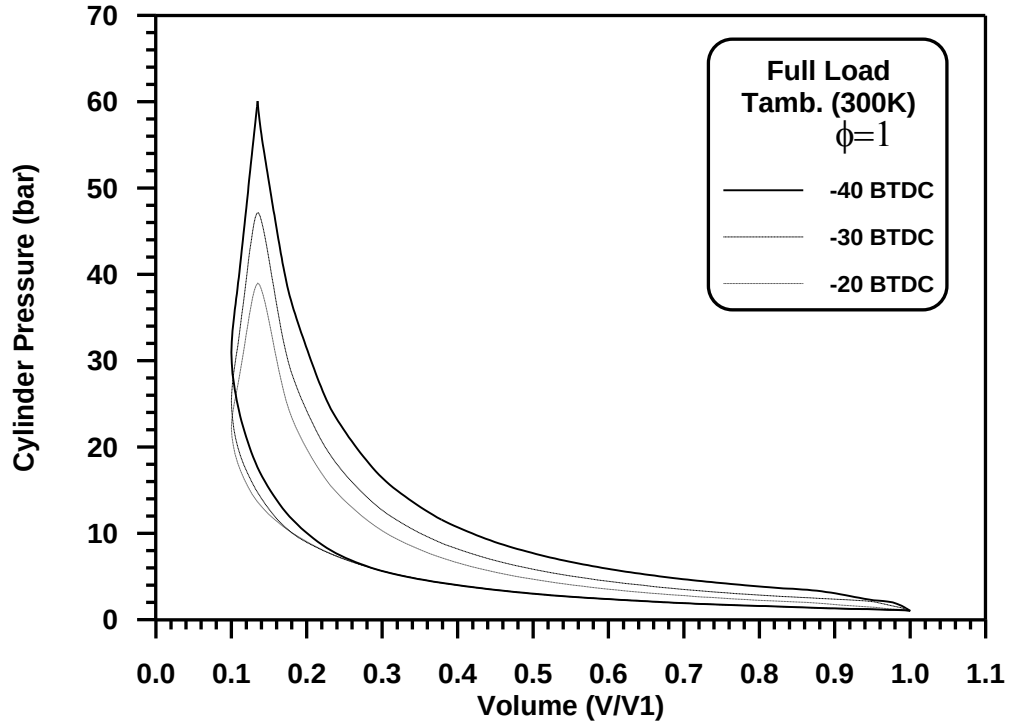


Fig. (٥.٤): Pressure-Volume Real Diagram for Different Spark Advance (degree), Before Top Dead Center.

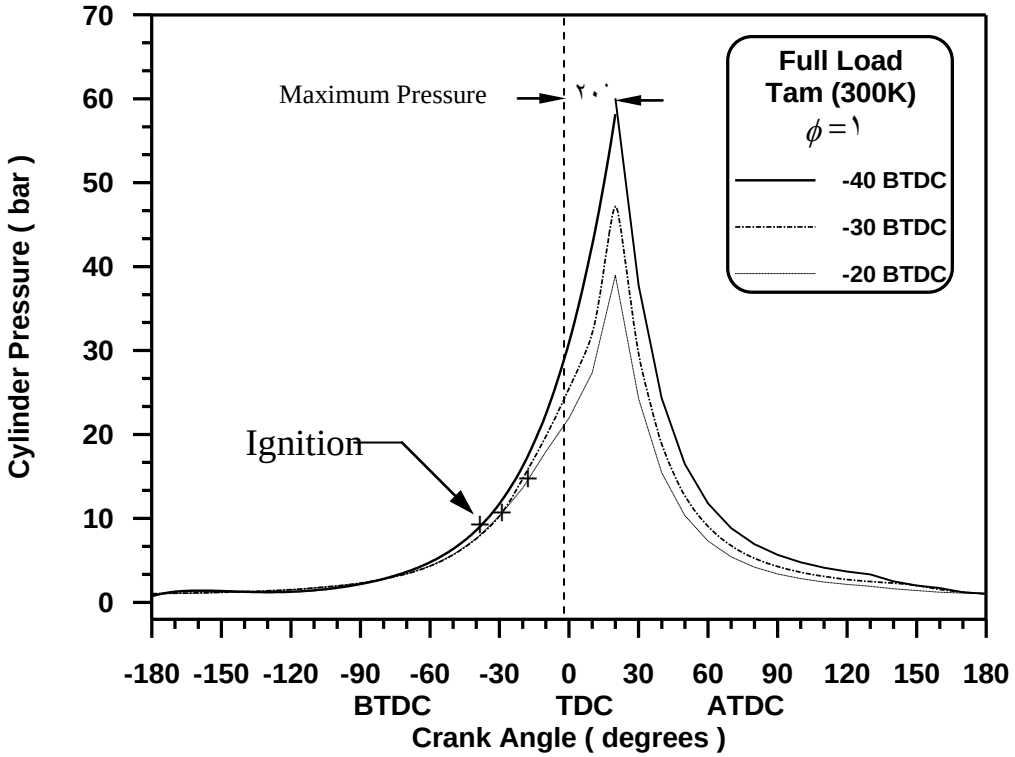


Fig. (٥.٥): Variation of Cylinder Pressure with Crank Angle for Different Angles of Advance.

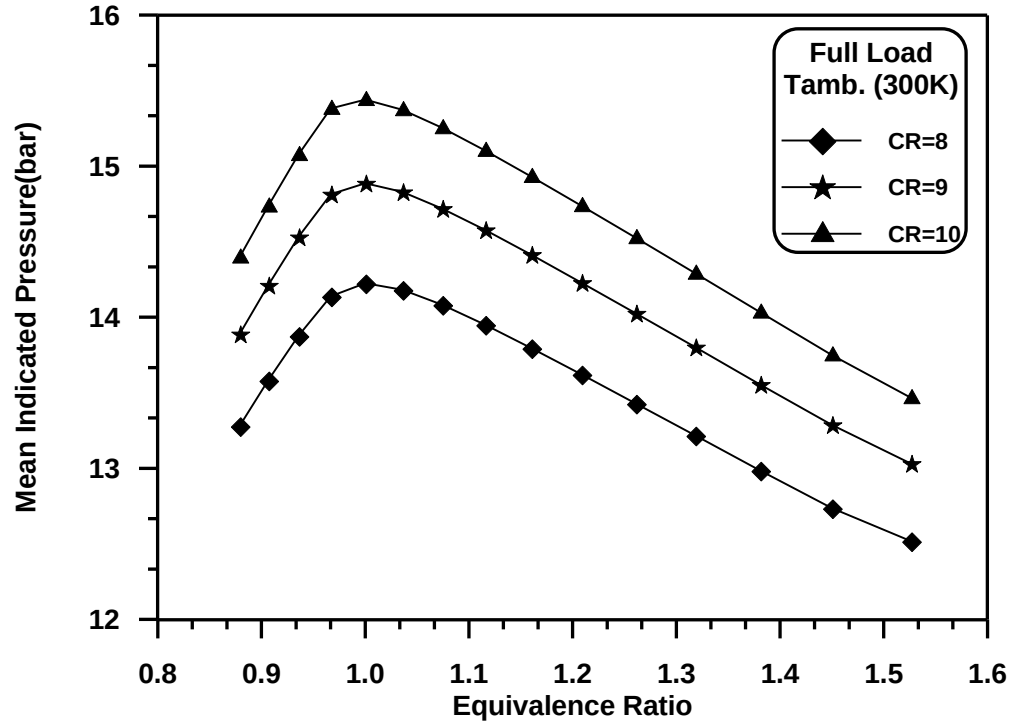


Fig. (٥.٦): Variation of Mean Indicated Pressure with Air to Fuel Ratio for Different Compression Ratios.

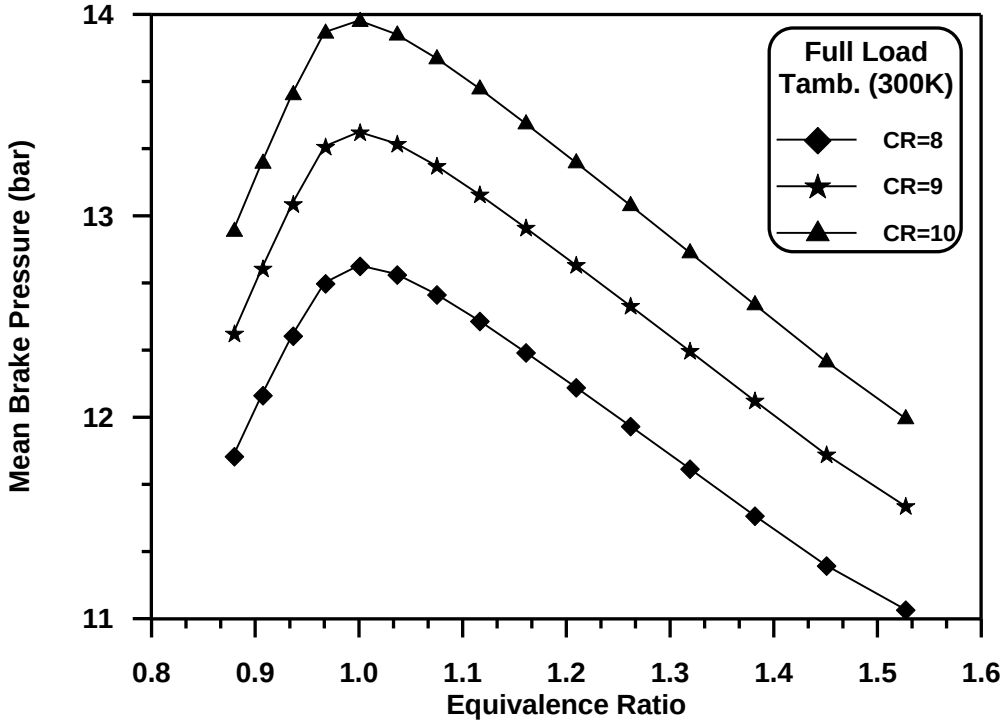


Fig. (٥.٧): Variation of Mean Brake Pressure with Air to Fuel Ratio for Different Compression Ratios.

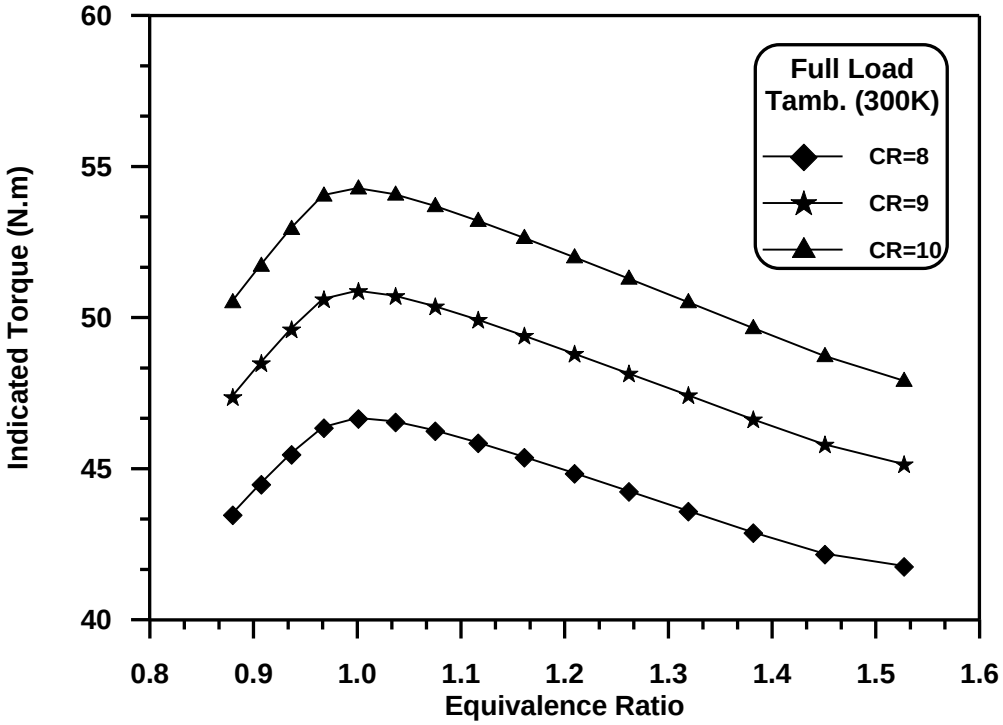


Fig. (٥.٨): Variation of Indicated Torque with Air to Fuel Ratio for Different Compression Ratios.

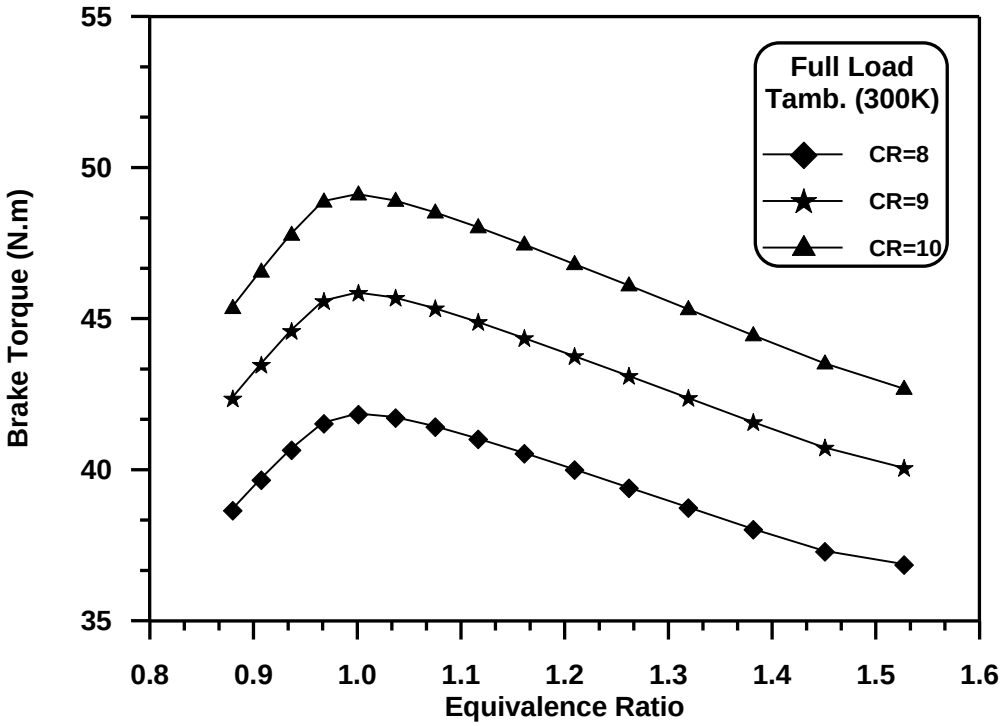


Fig. (٥.٩): Variation of Brake Torque with Air to Fuel Ratio for Different Compression Ratios.

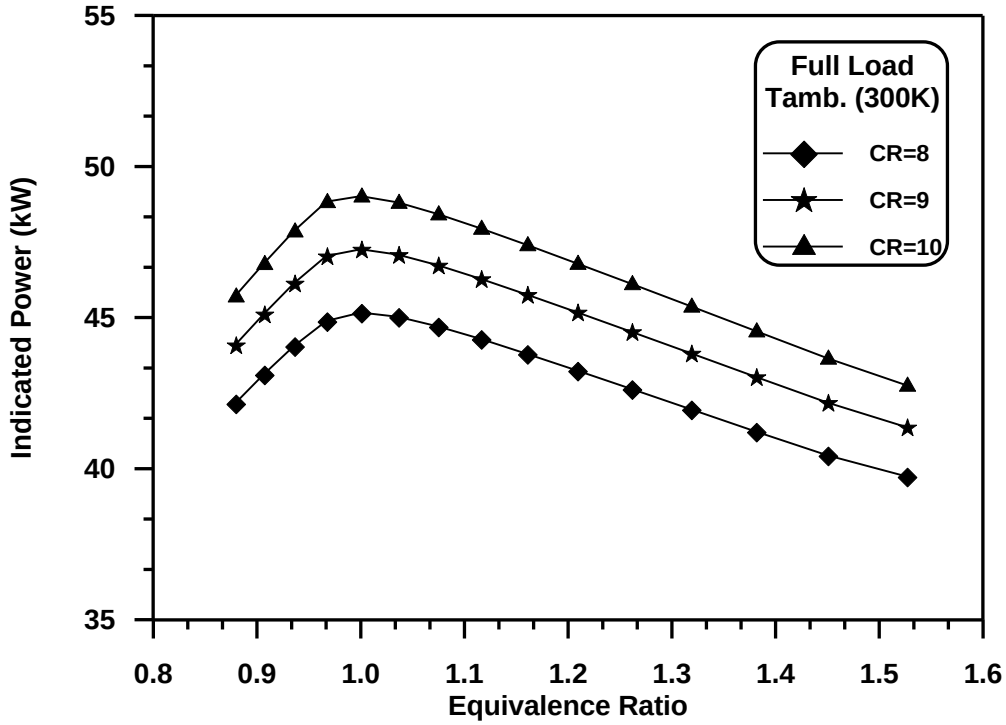


Fig. (٥.١٠): Variation of Indicated Power with Air to Fuel Ratio for Different Compression Ratios.

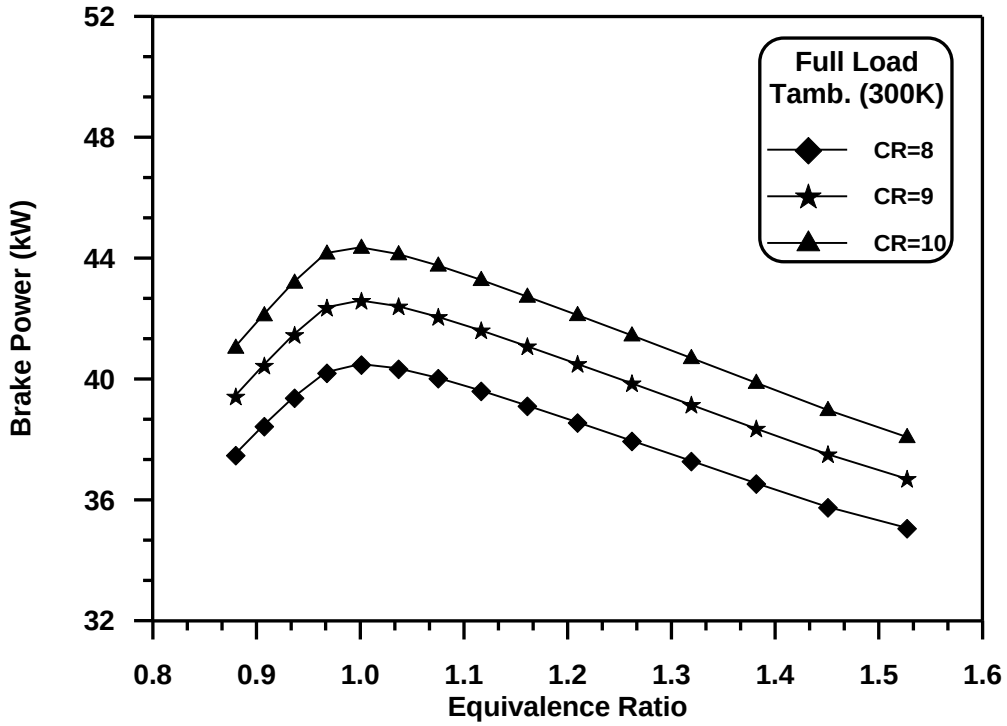


Fig. (٥.١١): Variation of Brake Power with Air to Fuel Ratio for Different Compression Ratios.

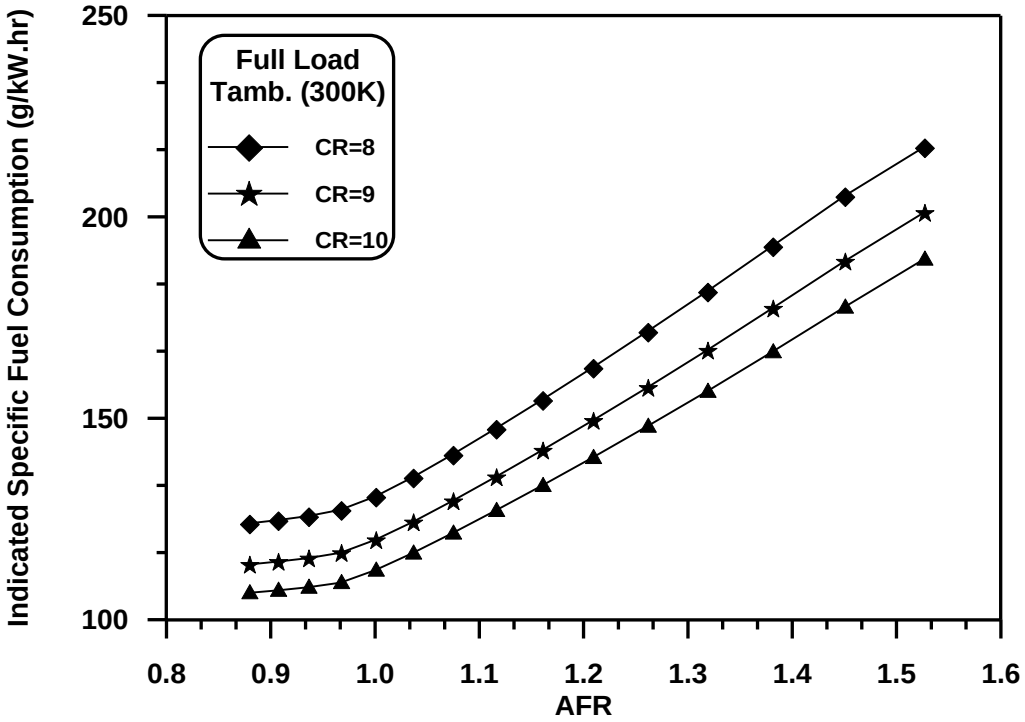


Fig. (٥.١٢): Variation of Indicated Specific Fuel Consumption with Air to Fuel Ratio for Different Compression Ratios.

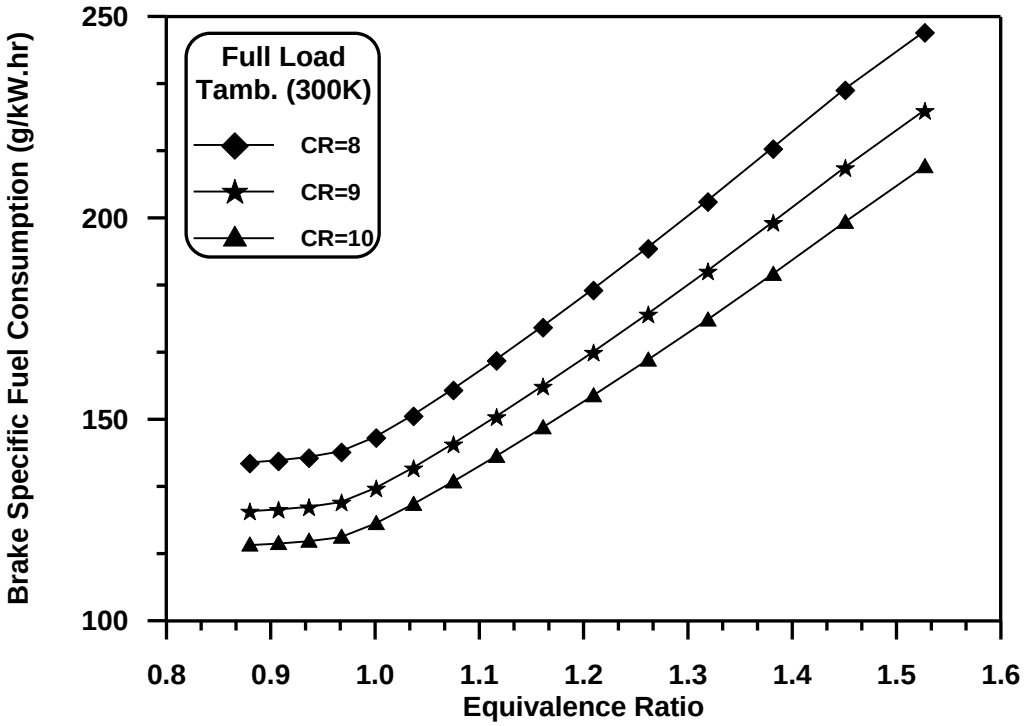


Fig. (٥.١٣): Variation of Brake Specific Fuel Consumption with Air to Fuel Ratio for Different Compression Ratios.

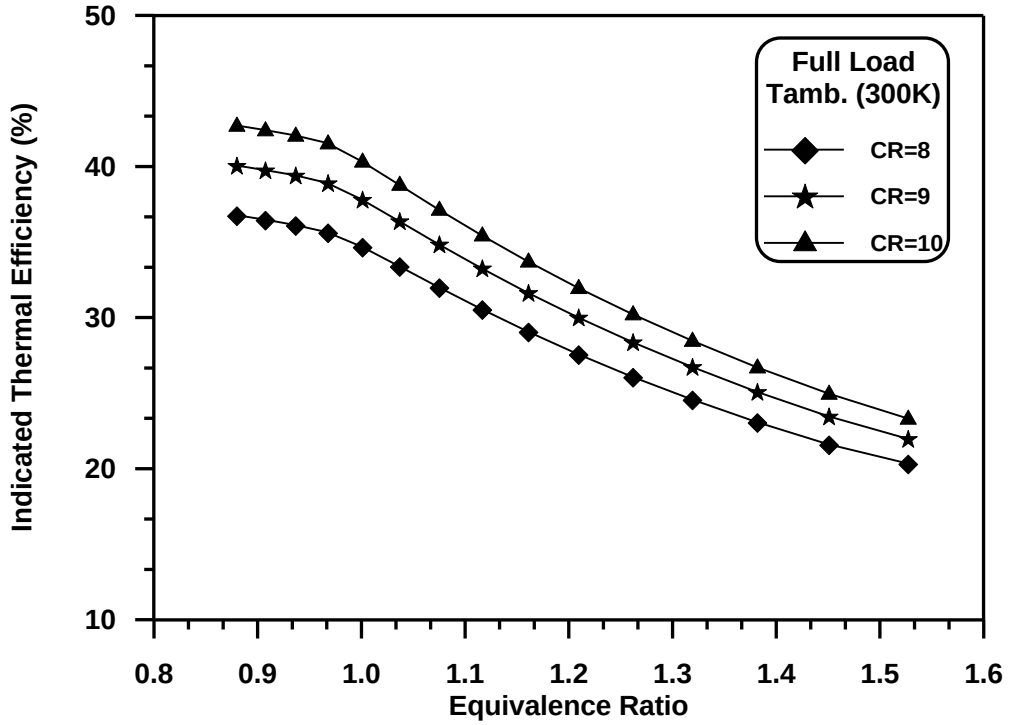


Fig. (٥.١٤): Variation of Indicated Thermal Efficiency with Air to Fuel Ratio for Different Compression Ratios.

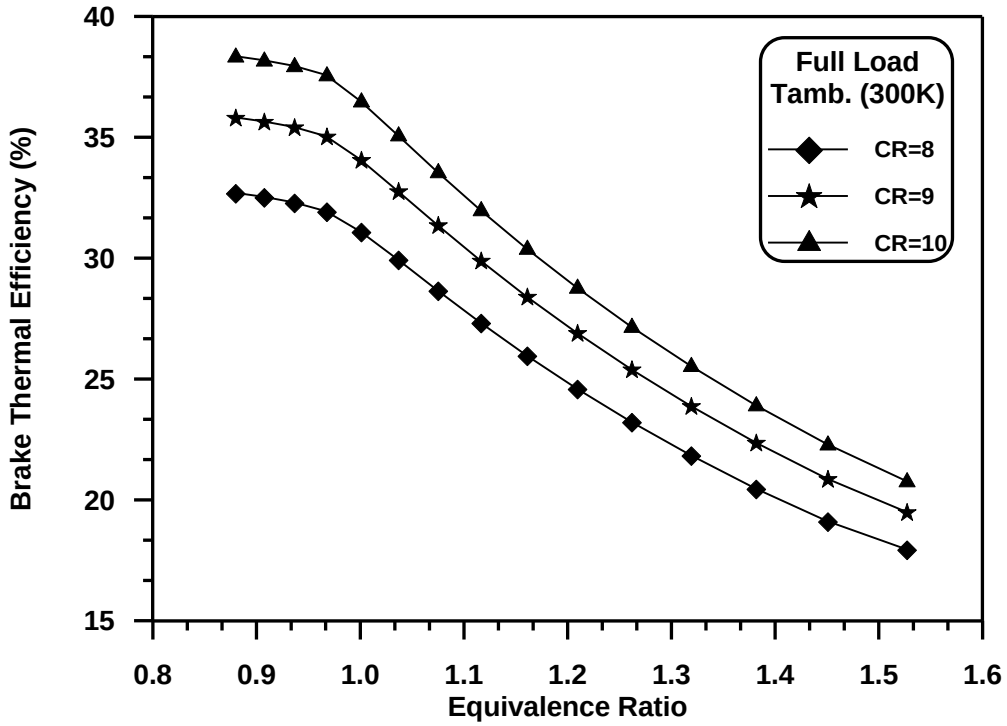


Fig. (٥.١٥): Variation of Brake Thermal Efficiency with Air to Fuel Ratio for Different Compression Ratios.

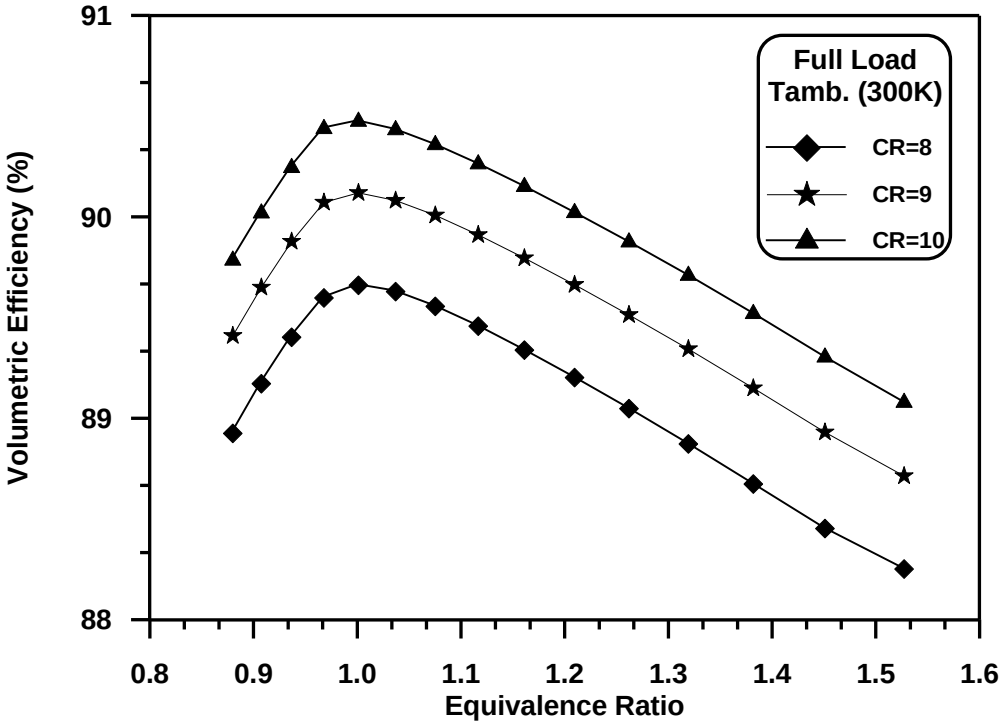


Fig. (٥.١٦): Variation of Volumetric Efficiency with Air to Fuel Ratio for Different Compression Ratios.

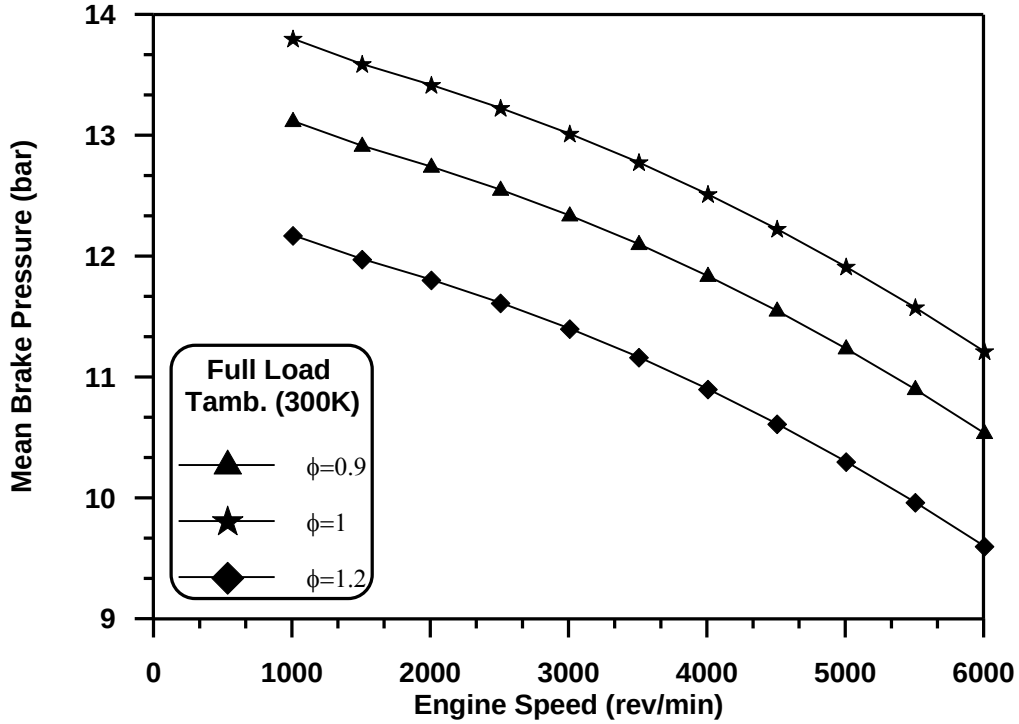


Fig. (٥.١٧): Variation of Mean Brake Pressure with Engine Speed for Different Equivalence Ratios.

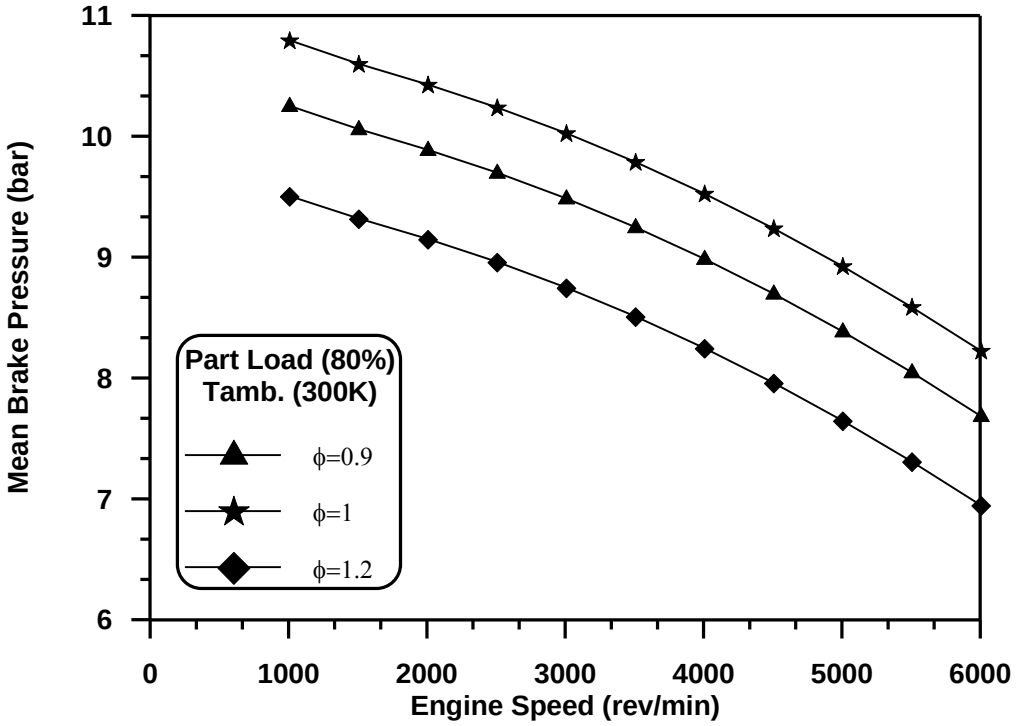


Fig. (٥.١٨): Variation of Mean Brake Pressure with Engine Speed for Different Equivalence Ratios.

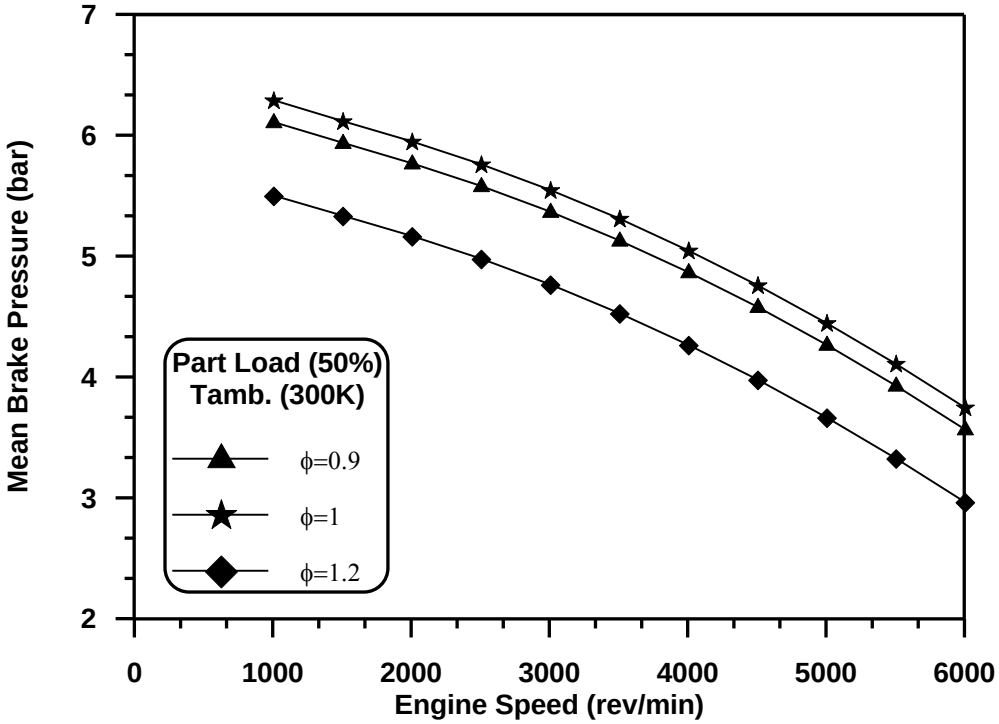


Fig. (٥.١٩): Variation of Mean Brake Pressure with Engine Speed for Different Equivalence Ratios.

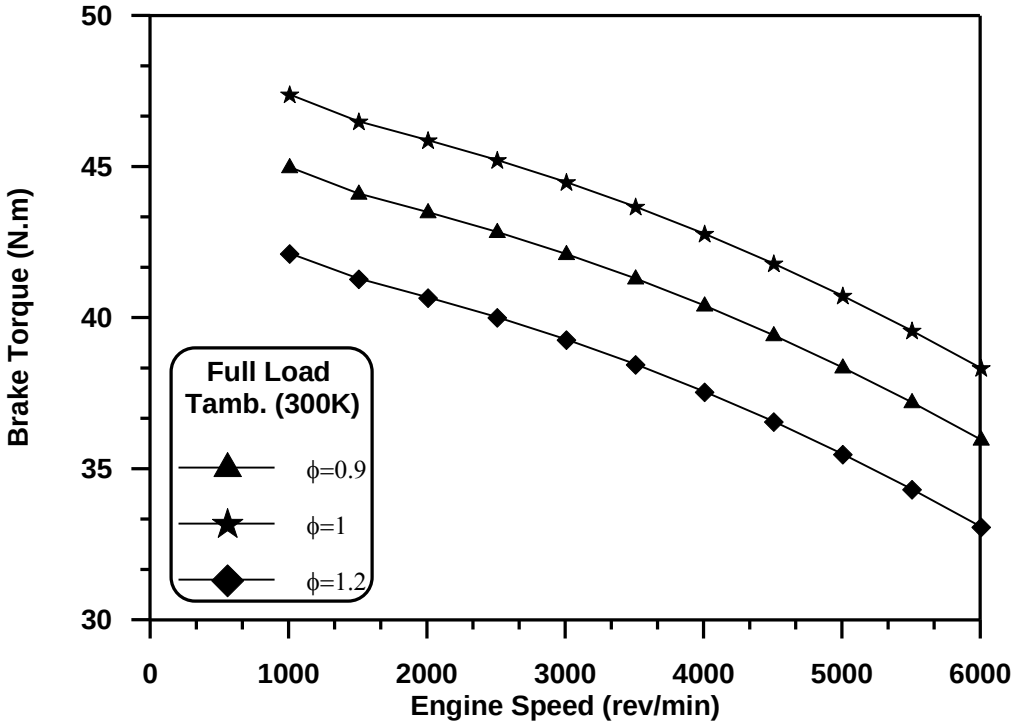


Fig. (٥.٢٠): Variation of Brake Torque with Engine Speed for Different Equivalence Ratios.

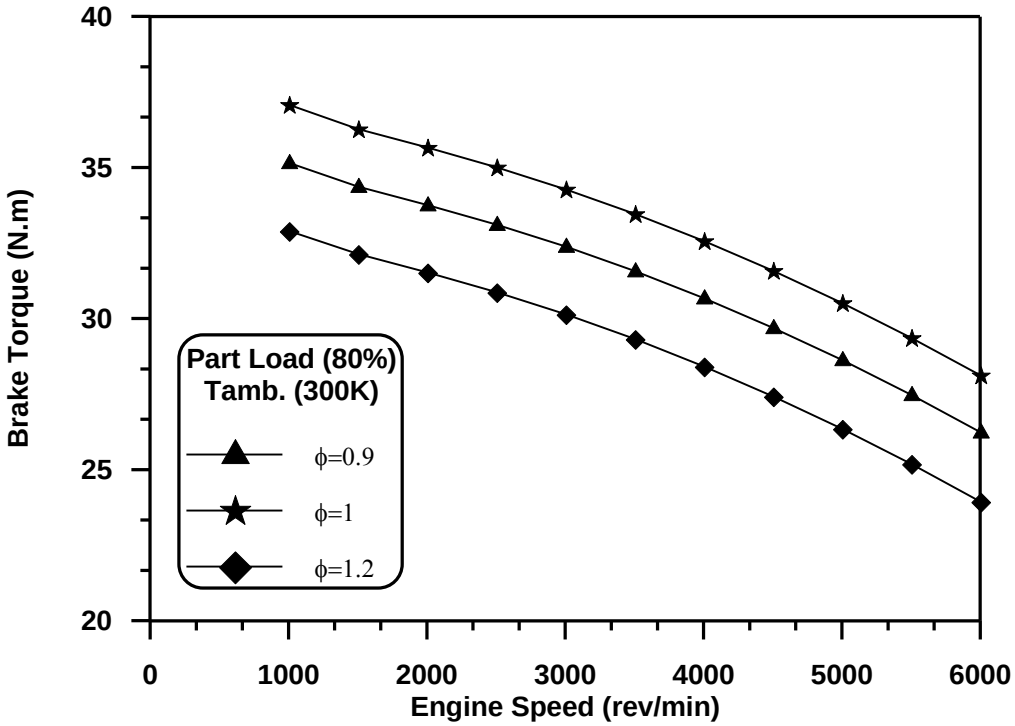


Fig. (٥.٢١): Variation of Brake Torque with Engine Speed for Different Equivalence Ratios.

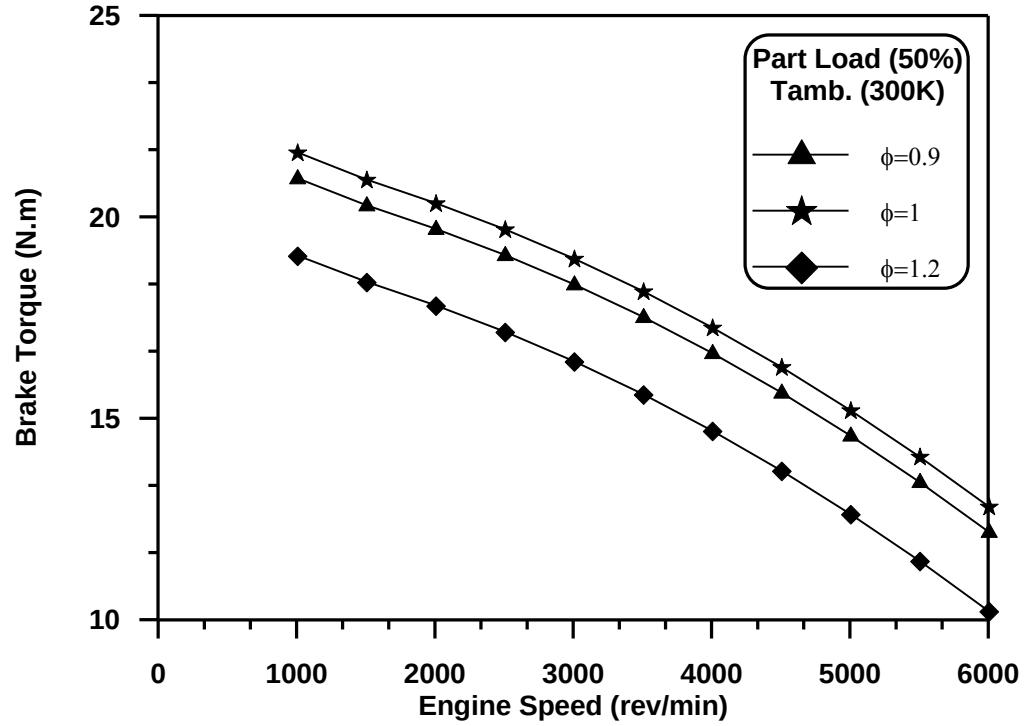


Fig. (٥.٢٢): Variation of Brake Torque with Engine Speed for Different Equivalence Ratios.

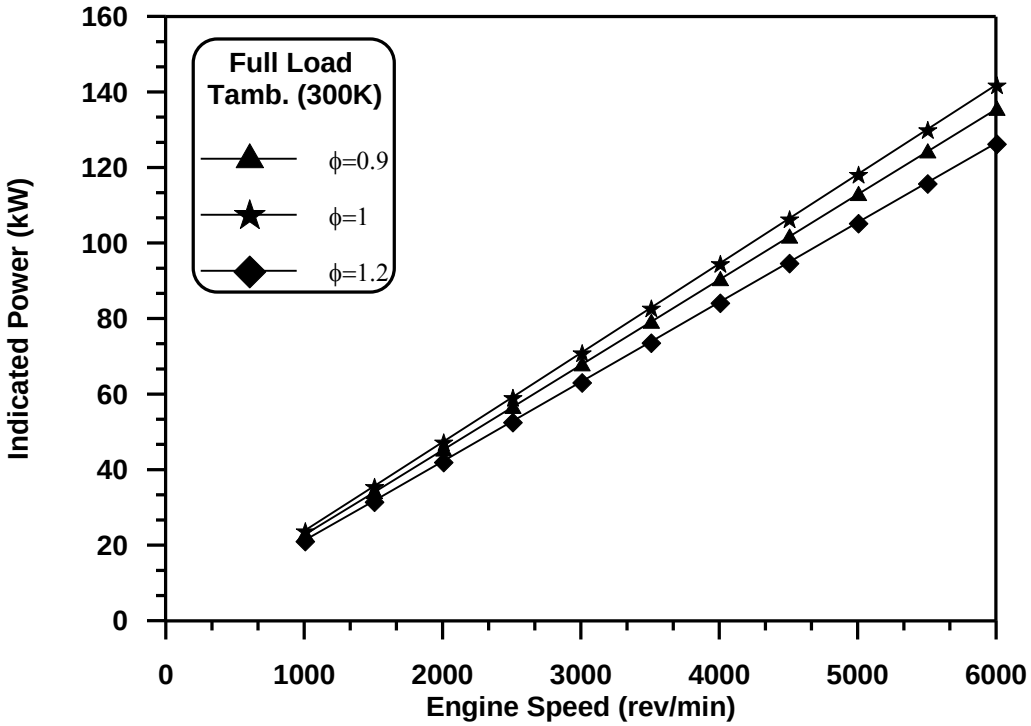


Fig. (٥.٢٣): Variation of Indicated Power with Engine Speed for Different Equivalence Ratios.

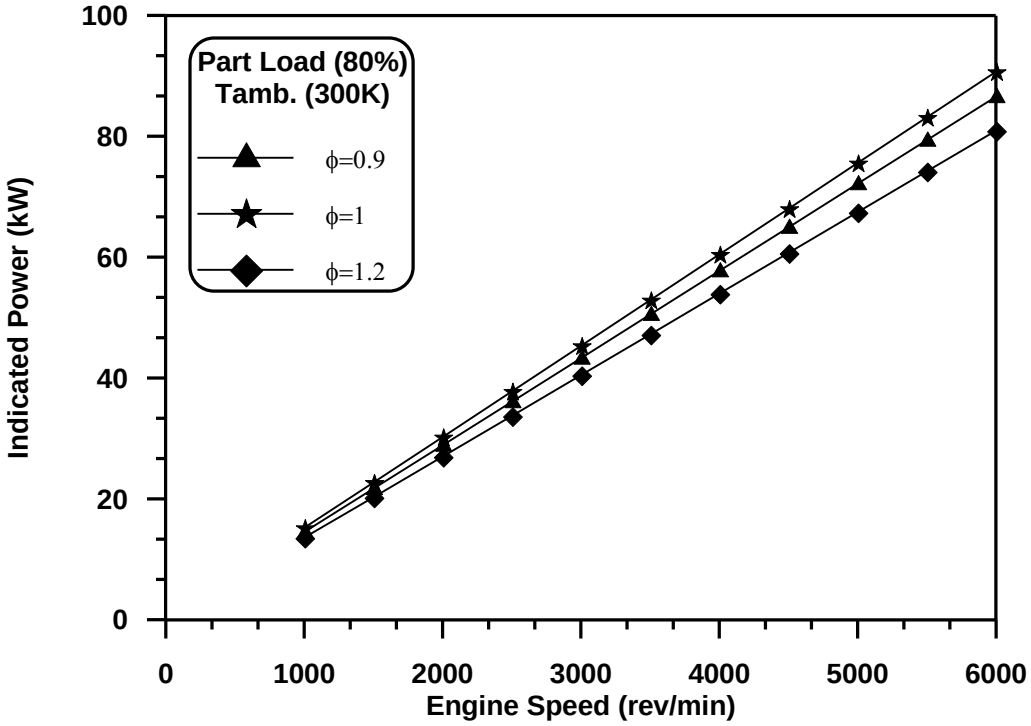


Fig. (٥.٢٤): Variation of Indicated Power with Engine Speed for Different Equivalence Ratios.

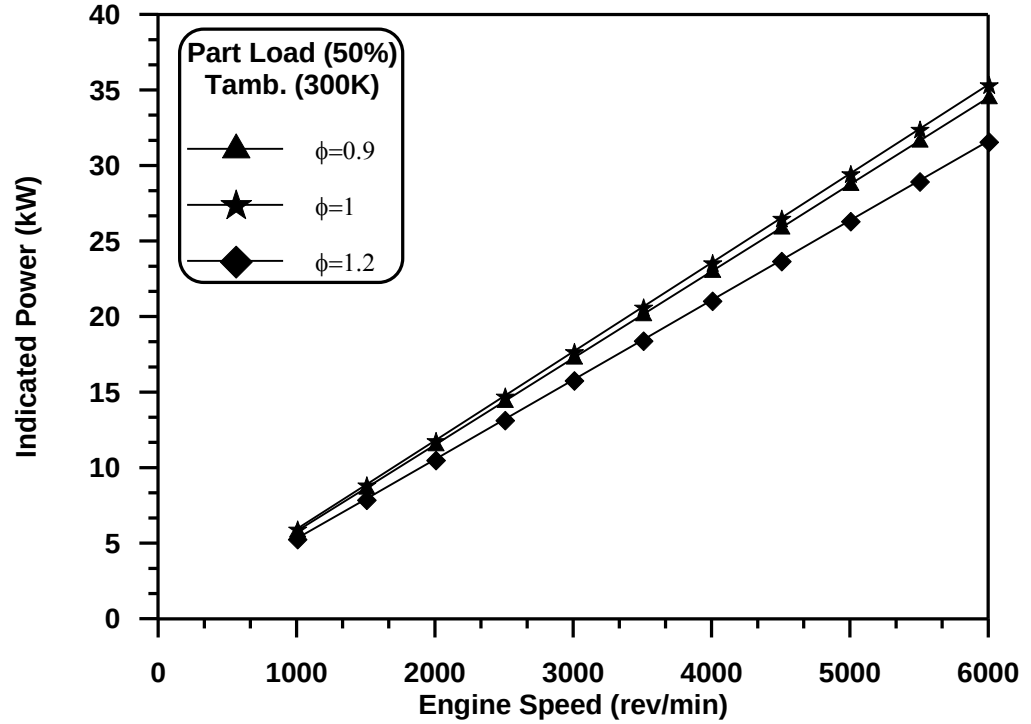


Fig. (٥.٢٥): Variation of Indicated Power with Engine Speed for Different Equivalence Ratios.

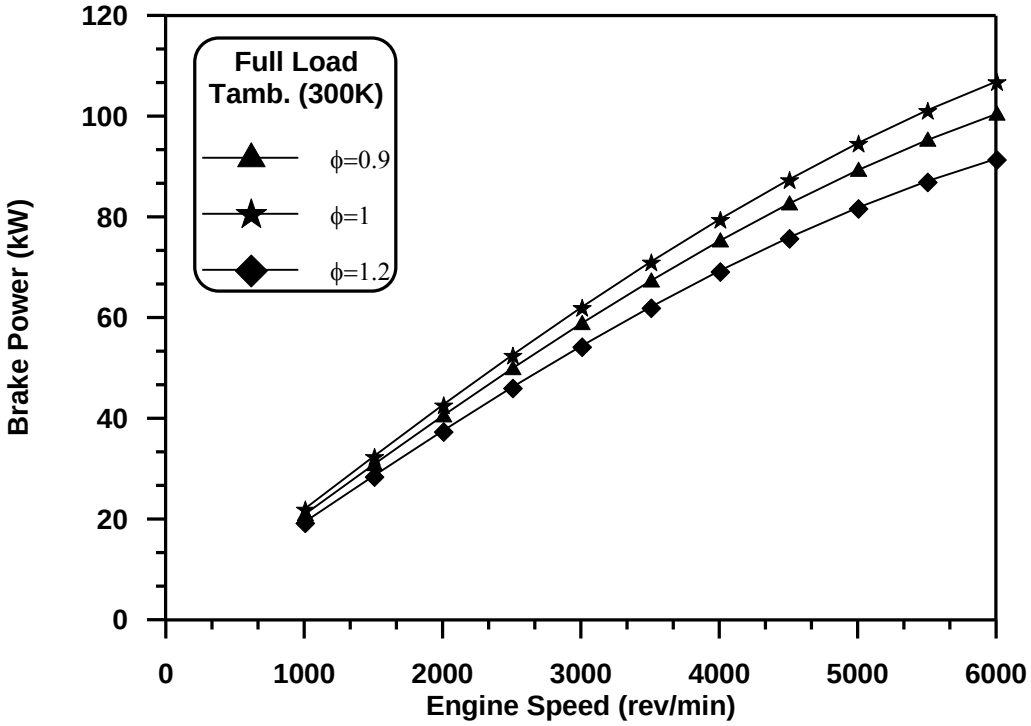


Fig. (٥.٢٦): Variation of Brake Power with Engine Speed for Different Equivalence Ratios.

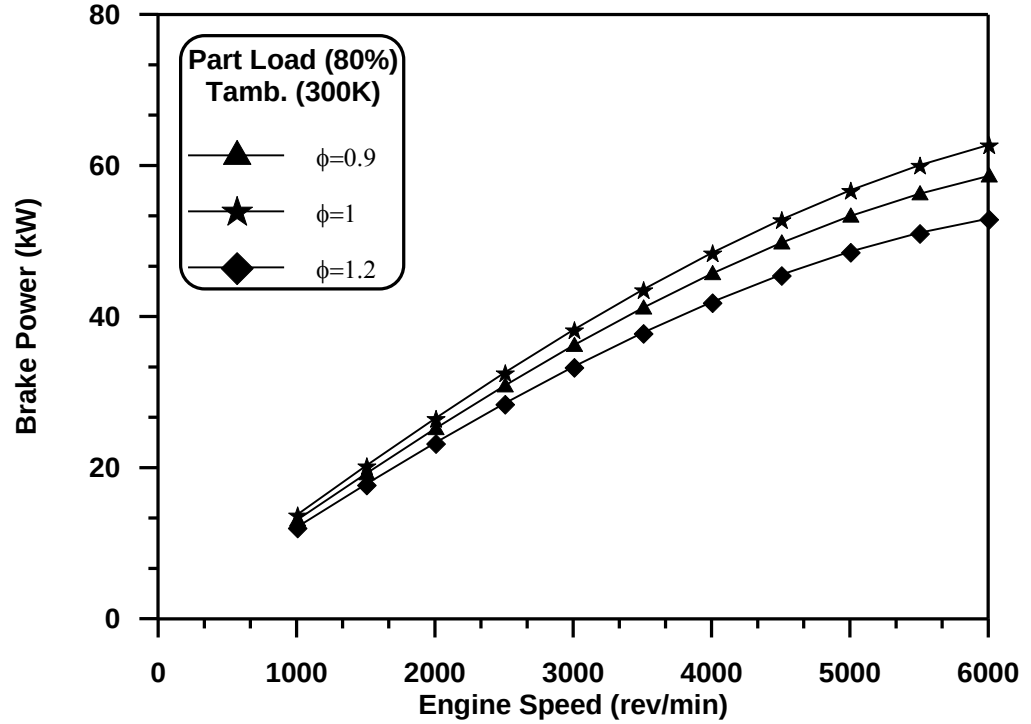


Fig. (٥.٢٧): Variation of Brake Power with Engine Speed for Different Equivalence Ratios.

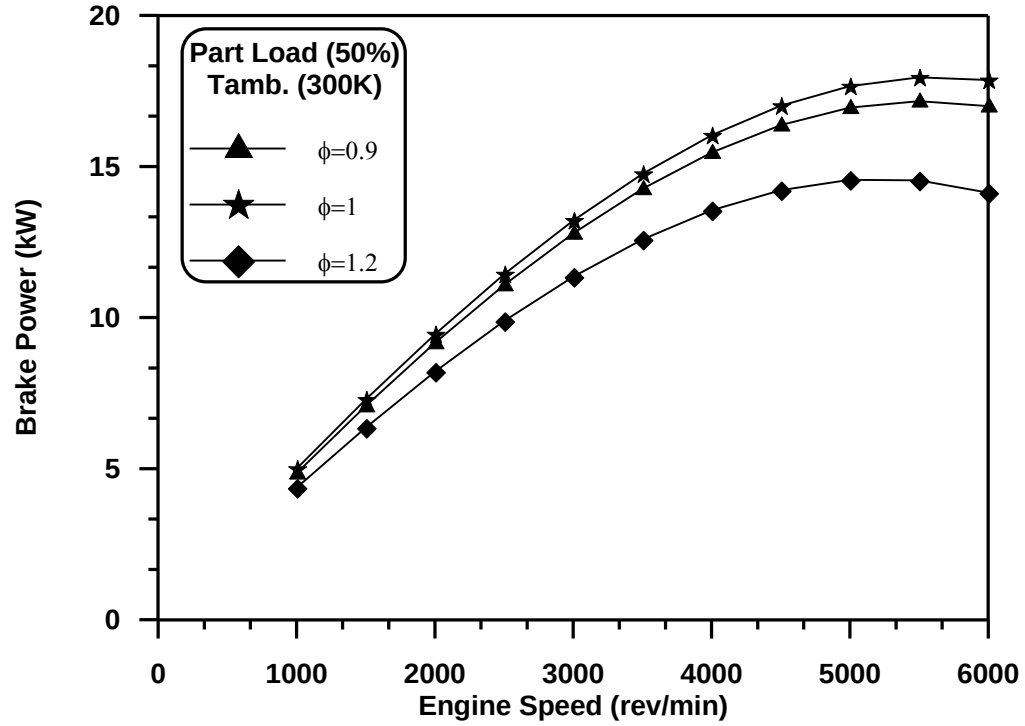


Fig. (٥.٢٨): Variation of Brake Power with Engine Speed for Different Equivalence Ratios.

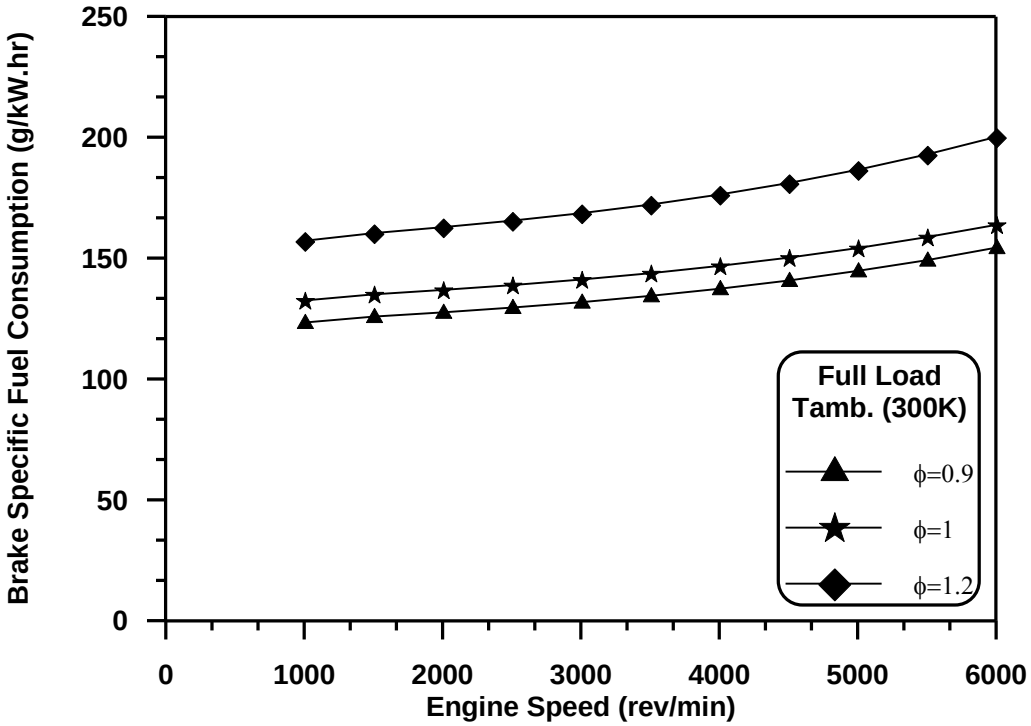


Fig. (٥.٢٩): Variation of Brake Specific Fuel Consumption with Engine Speed for Different Equivalence Ratios.

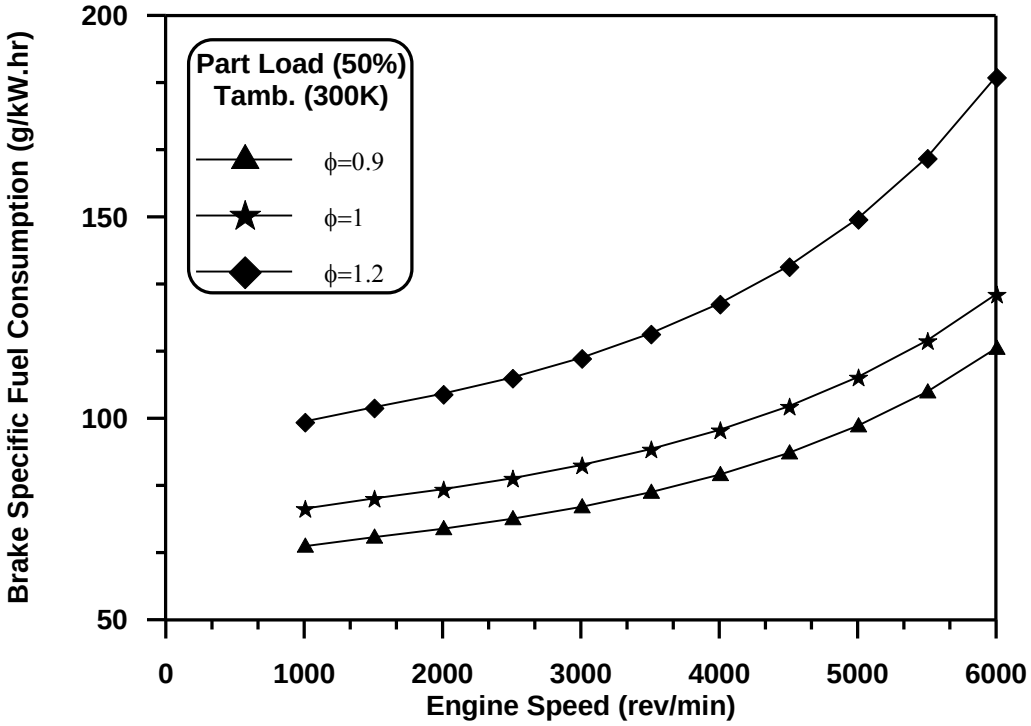


Fig. (٥.٣٠): Variation of Brake Specific Fuel Consumption with Engine Speed for Different Equivalence Ratios.

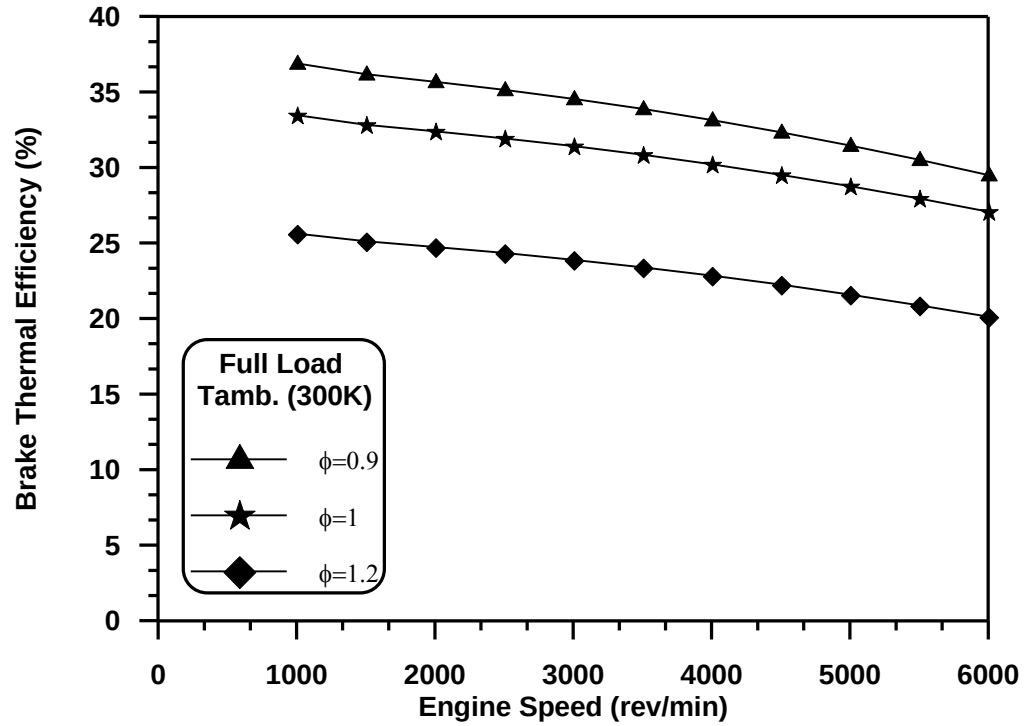


Fig. (٥.٣١): Variation of Brake Thermal Efficiency with Engine Speed for Different Equivalence Ratios.

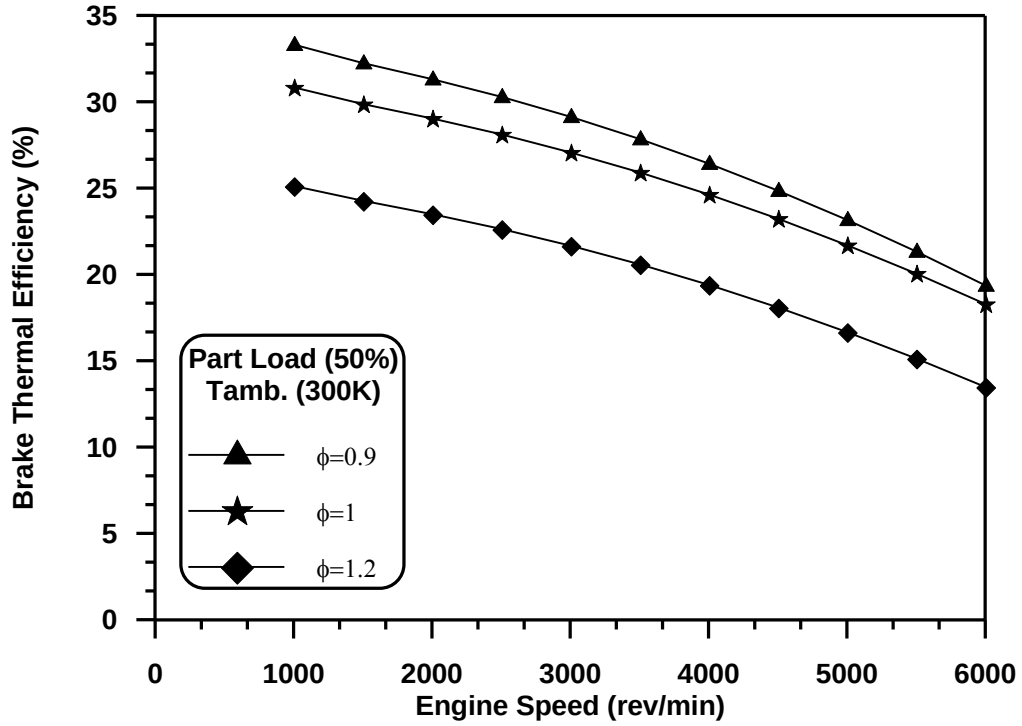


Fig. (٥.٣٢): Variation of Brake Thermal Efficiency with Engine Speed for Different Equivalence Ratios.

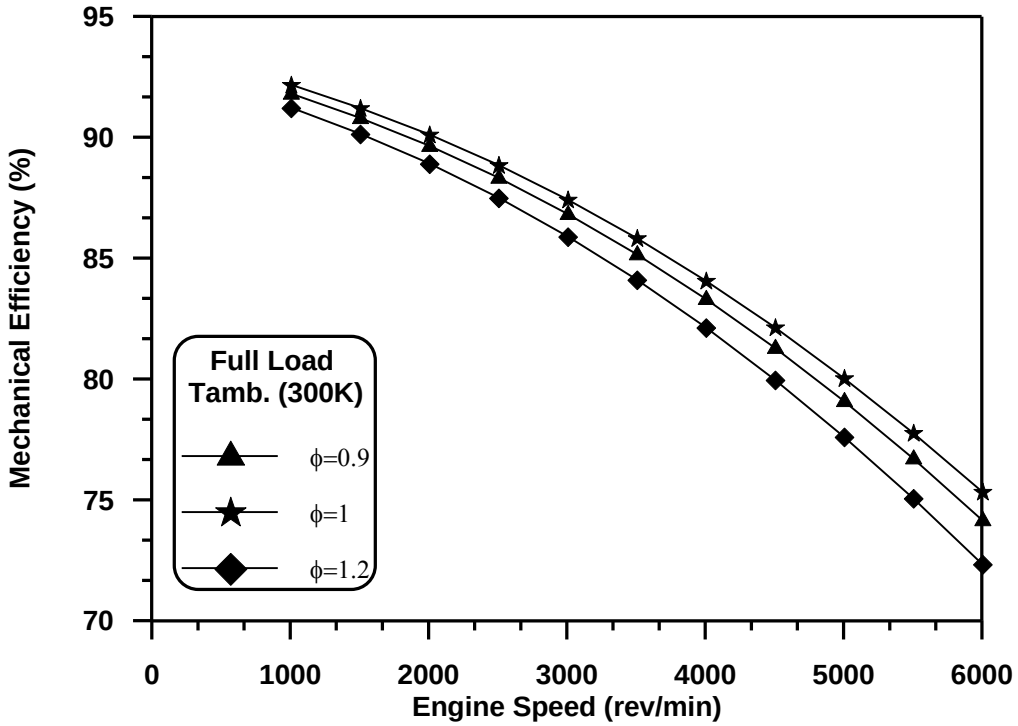


Fig. (٥.٣٣): Variation of Mechanical Efficiency with Engine Speed for Different Equivalence Ratios.

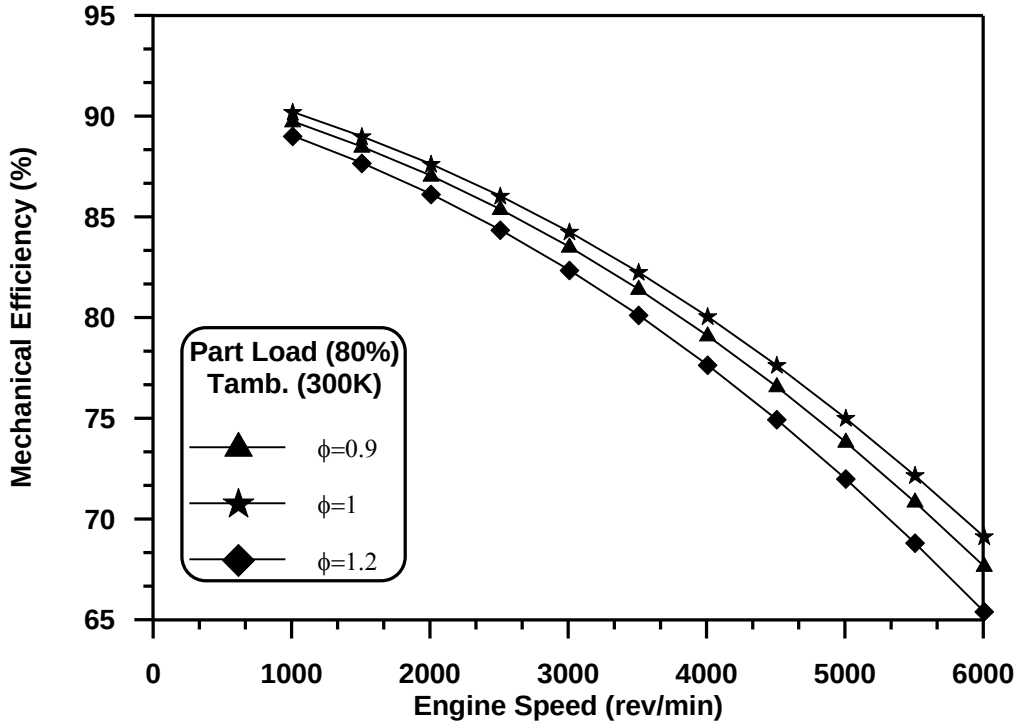


Fig. (٥.٣٤): Variation of Mechanical Efficiency with Engine Speed for Different Equivalence Ratios.

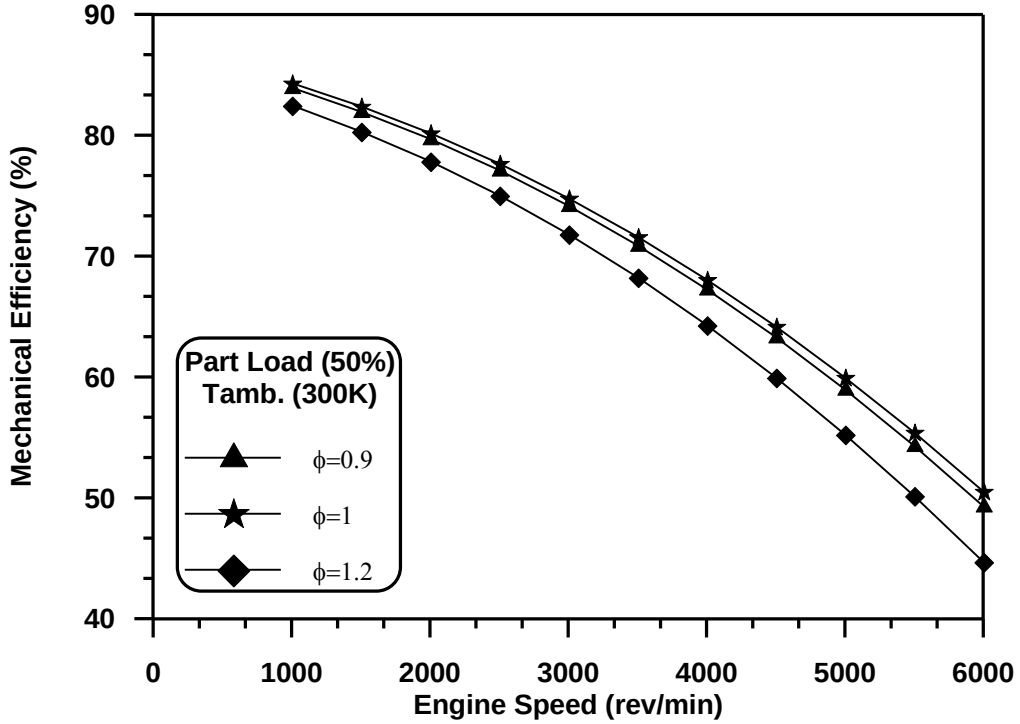


Fig. (٥.٣٥): Variation of Mechanical Efficiency with Engine Speed for Different Equivalence Ratios.

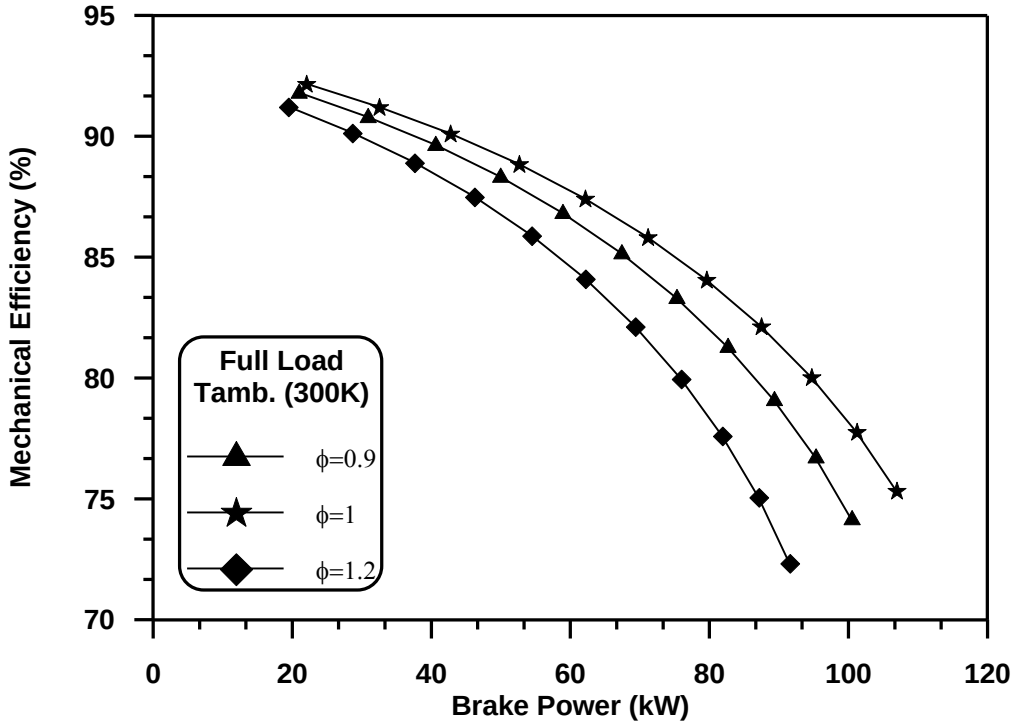


Fig. (٥.٣٦): Variation of Mechanical Efficiency with Brake Power for Different Equivalence Ratios.

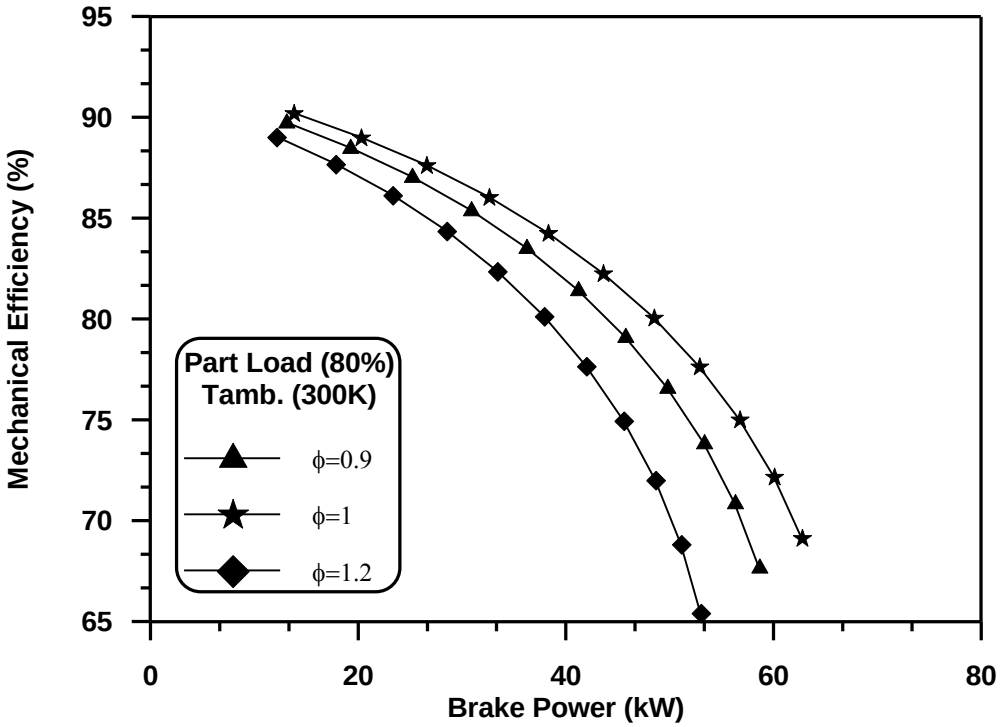


Fig. (٥.٣٧): Variation of Mechanical Efficiency with Brake Power for Different Equivalence Ratios.

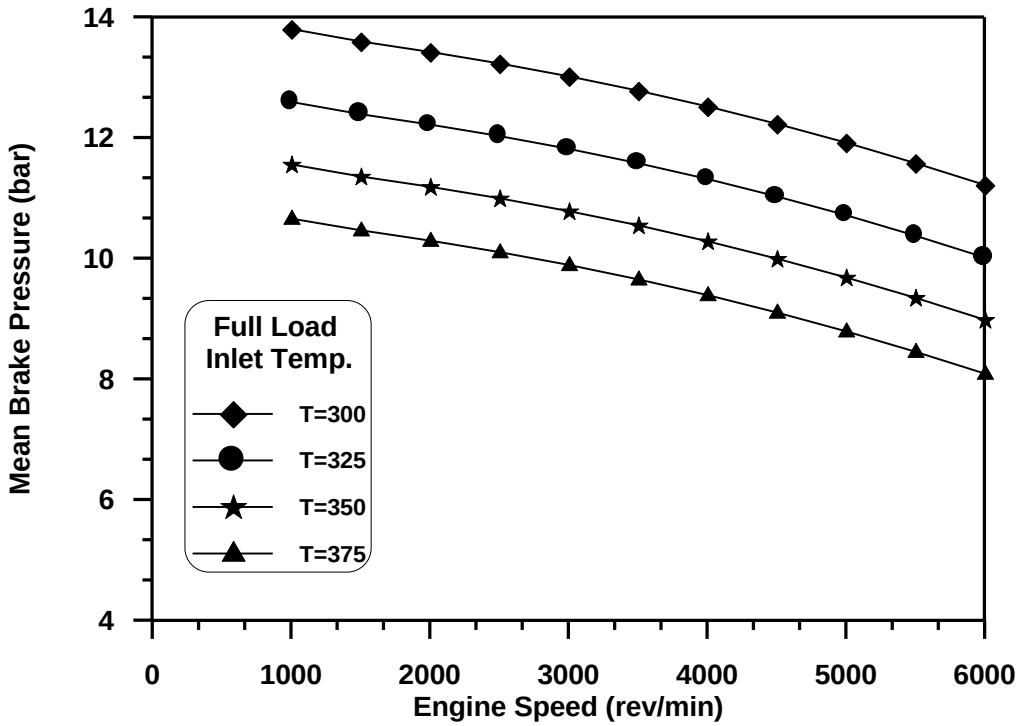


Fig. (٥.٣٨): Variation of Mean Brake Pressure with Engine Speed for Different Inlet Temperatures.

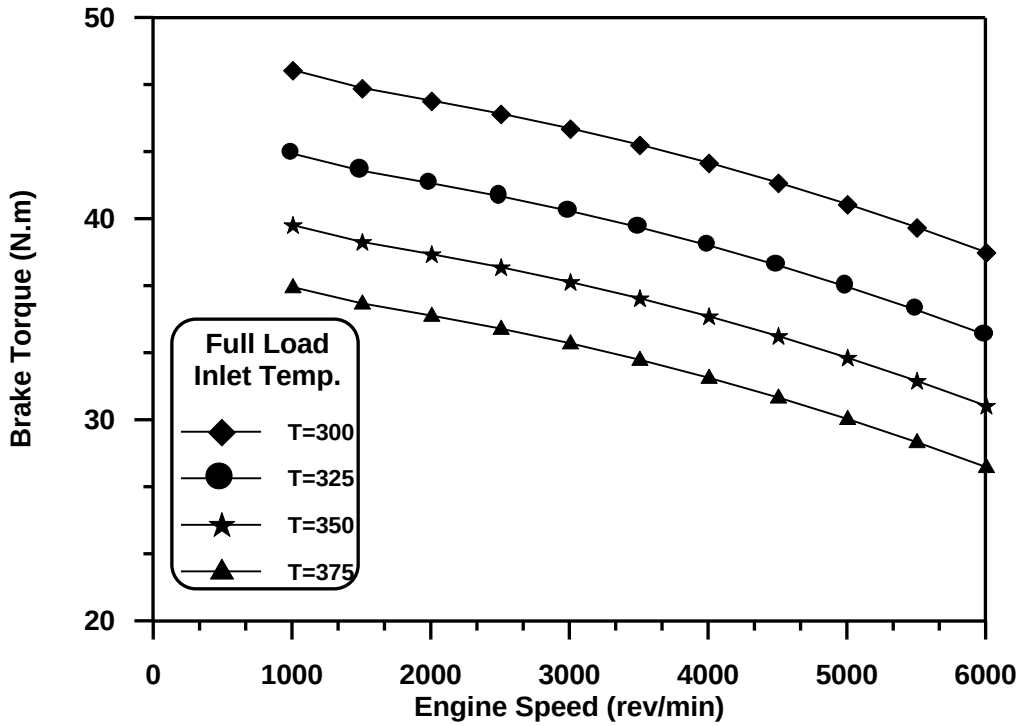


Fig. (٥.٣٩): Variation of Mean Brake Pressure with Engine Speed for Different Inlet Temperatures.

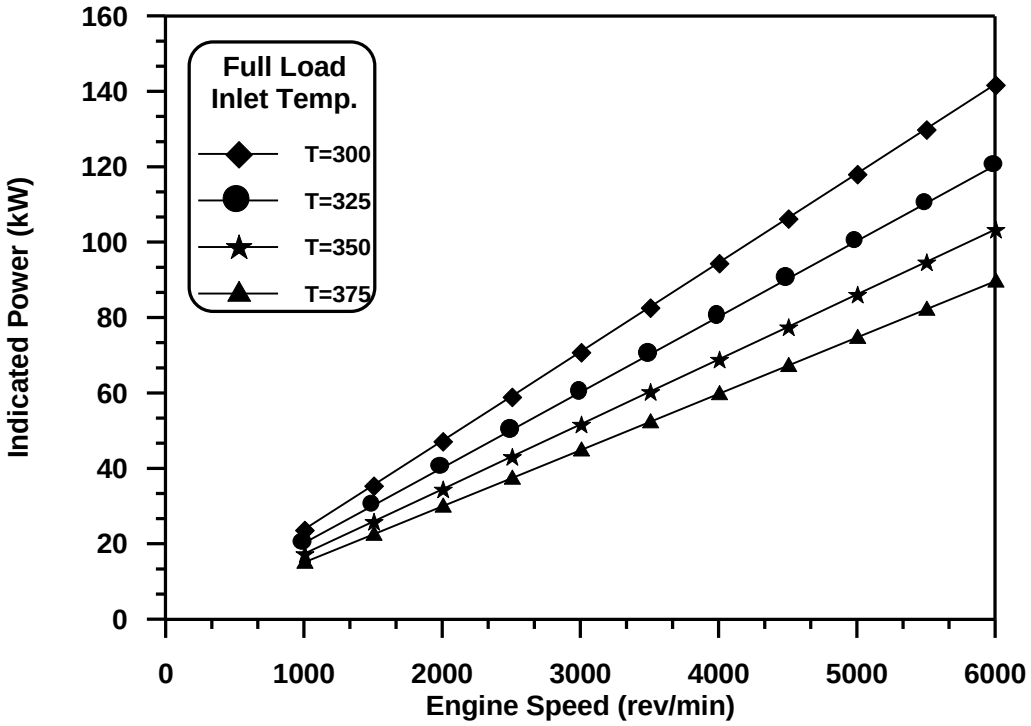


Fig. (٥.٤٠): Variation of Indicated Power with Engine Speed for Different Inlet Temperatures.

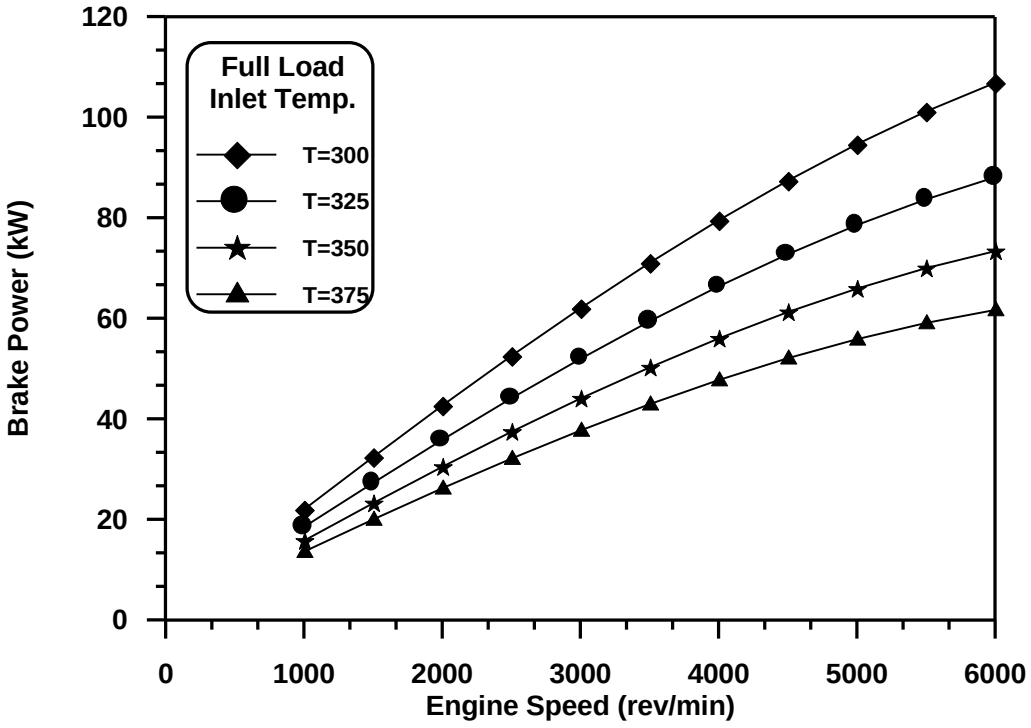


Fig. (٥.٤١): Variation of Brake Power with Engine Speed for Different Inlet Temperatures.

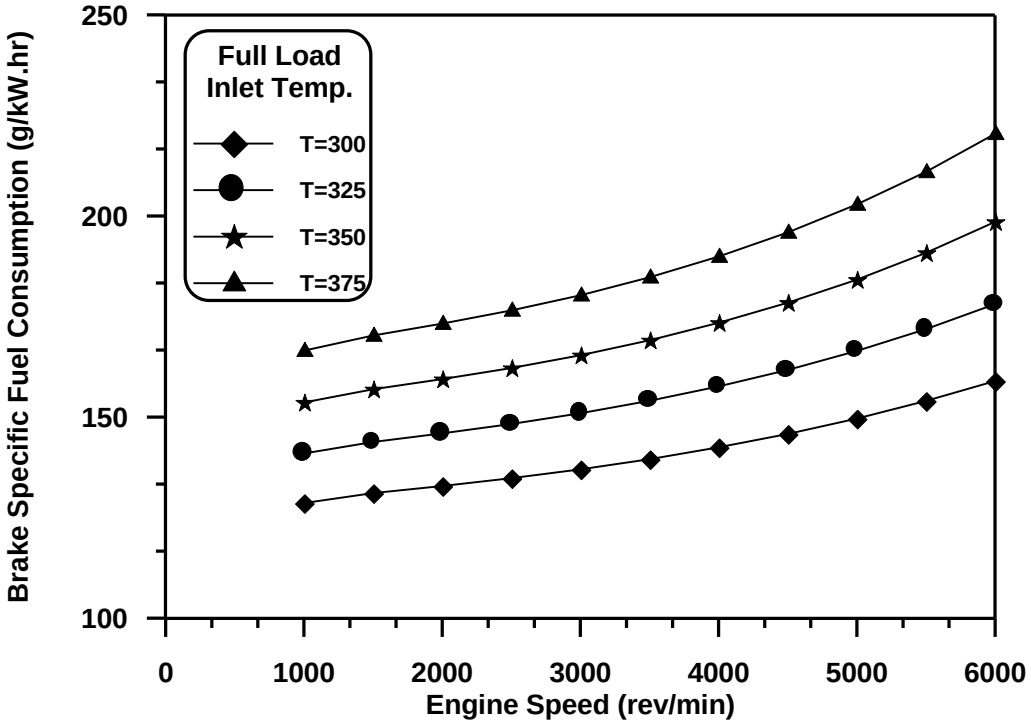


Fig. (٥.٤٢): Variation of Brake Specific Fuel Consumption with Engine Speed for Different Inlet Temperatures.

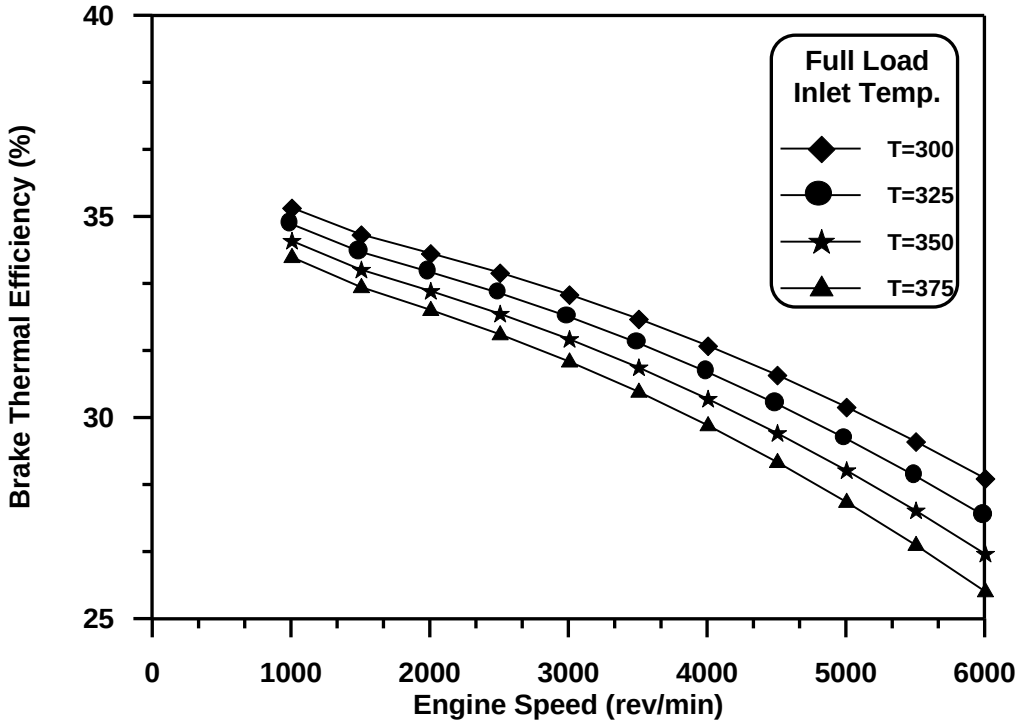


Fig. (٥.٤٣): Variation of Brake Thermal Efficiency with Engine Speed for Different Inlet Temperatures.

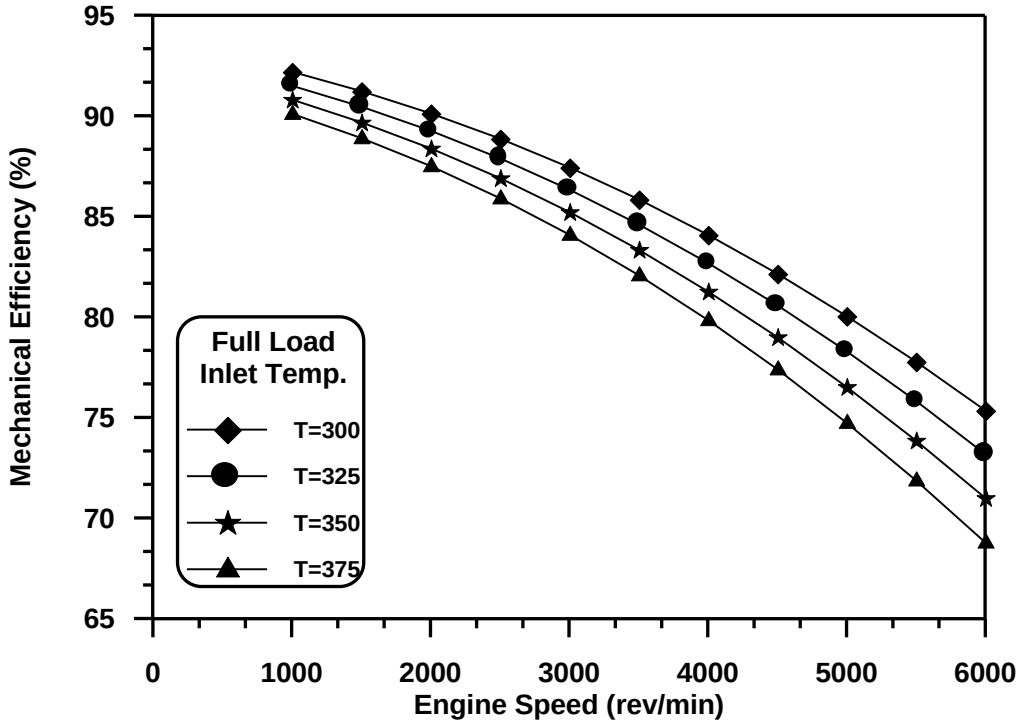


Fig. (٥.٤٤): Variation of Mechanical Efficiency with Engine Speed for Different Inlet Temperatures.

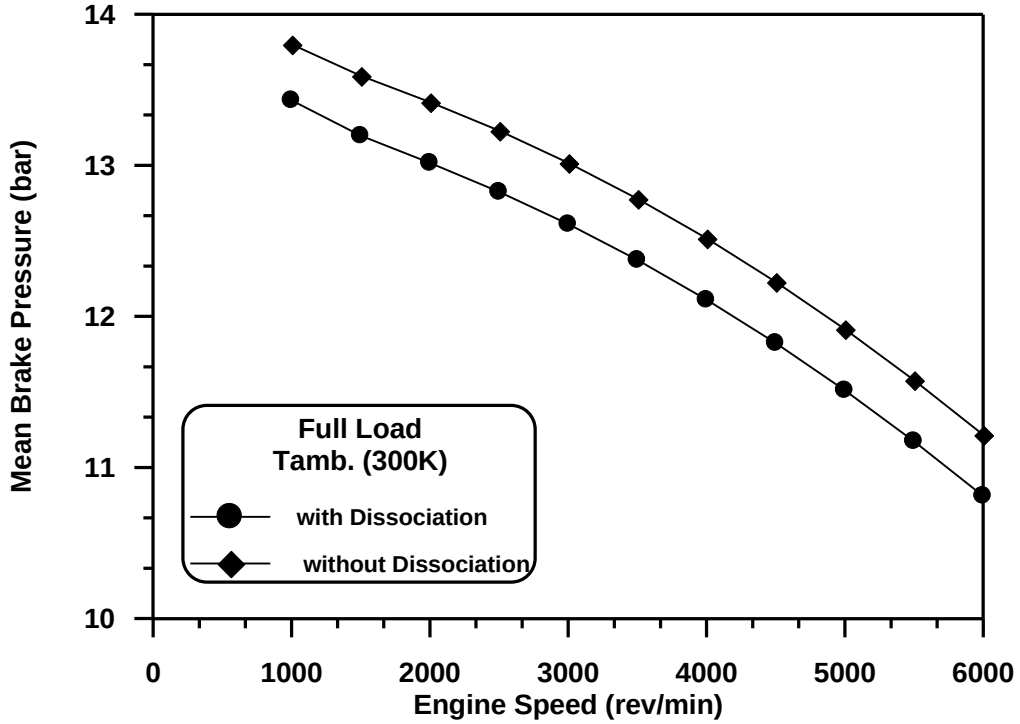


Fig. (٥.٤٥): Variation of Mean Brake Pressure with Engine Speed for Dissociation and without Dissociation.

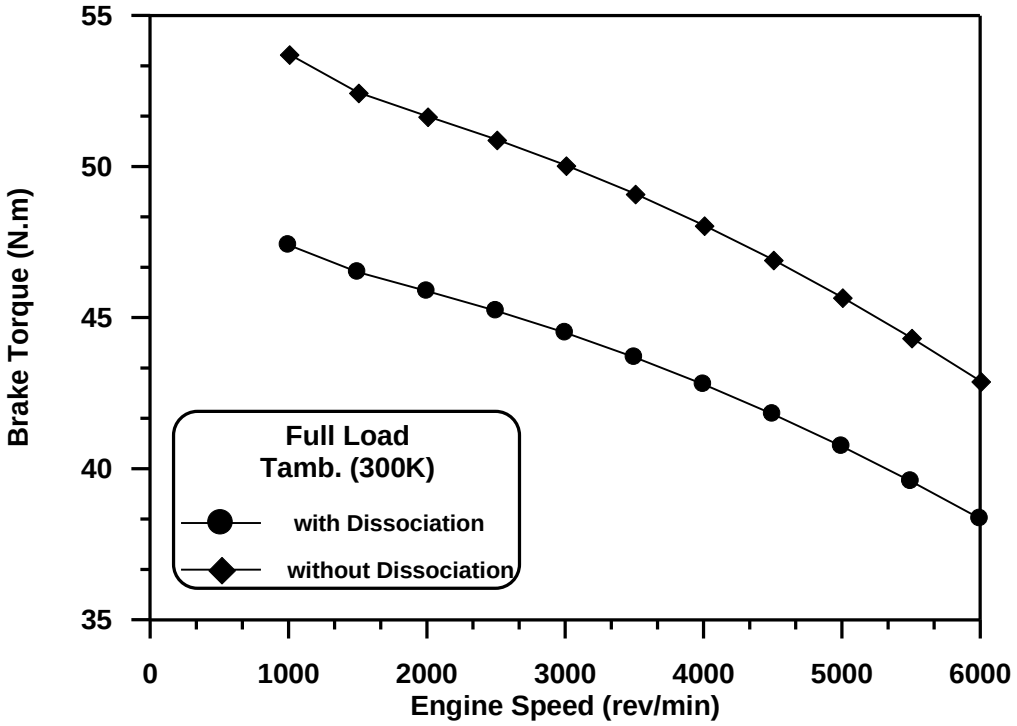


Fig. (٥.٤٦): Variation of Brake Torque with Engine Speed for Dissociation and without Dissociation.

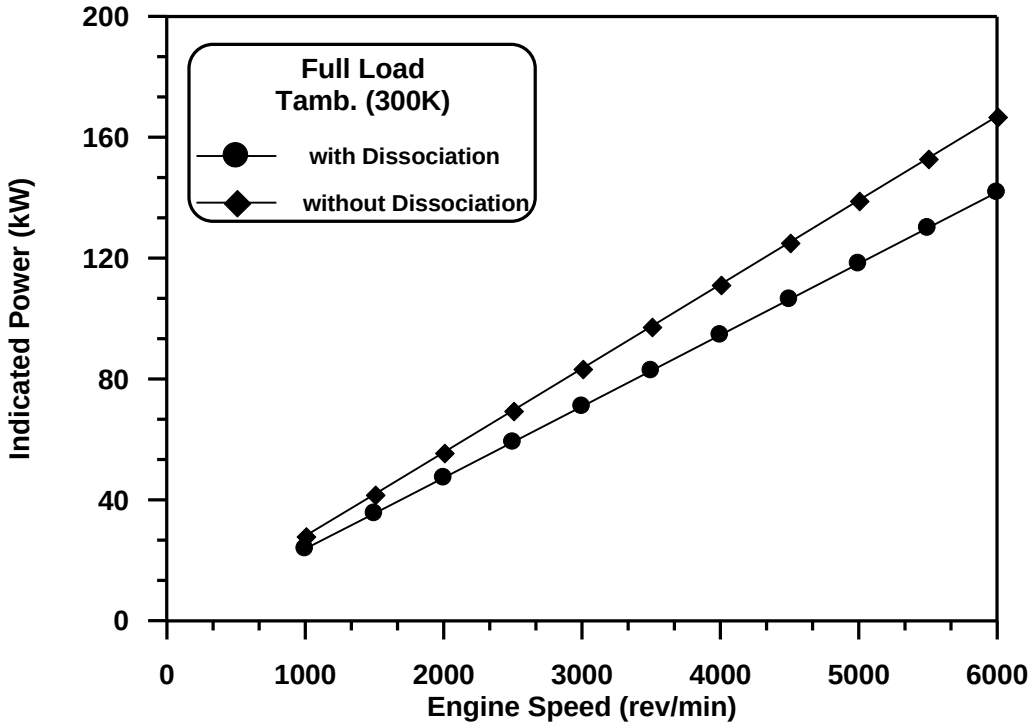


Fig. (٥.٤٧): Variation of Indicated Power with Engine Speed for Dissociation and without Dissociation.

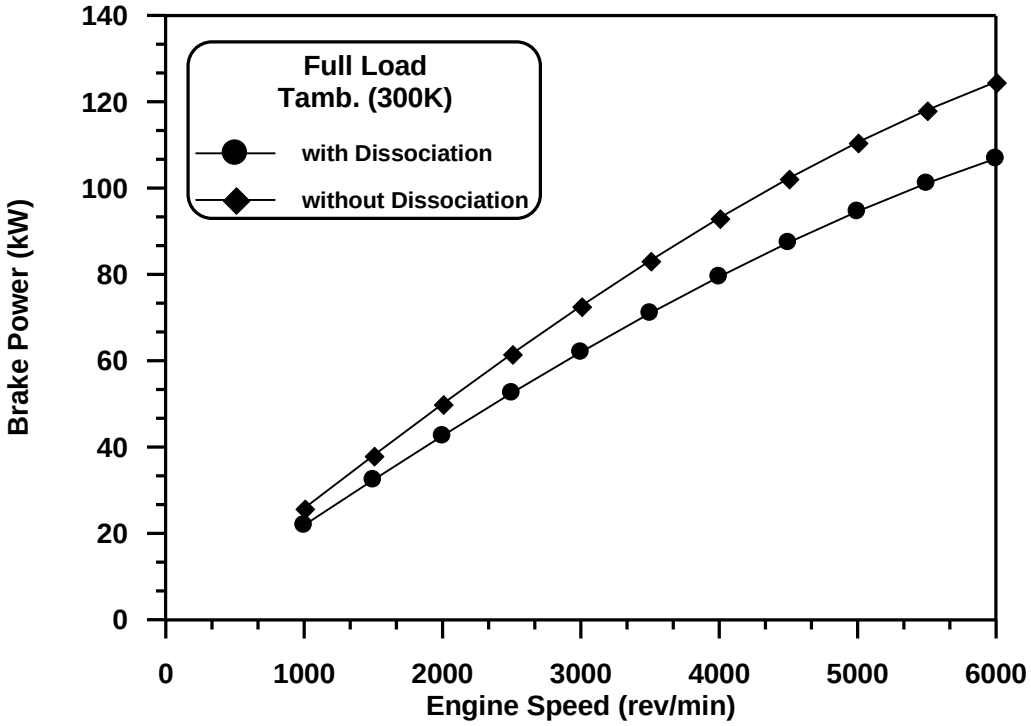


Fig. (٥.٤٨): Variation of Brake Power with Engine Speed for Dissociation and without Dissociation.

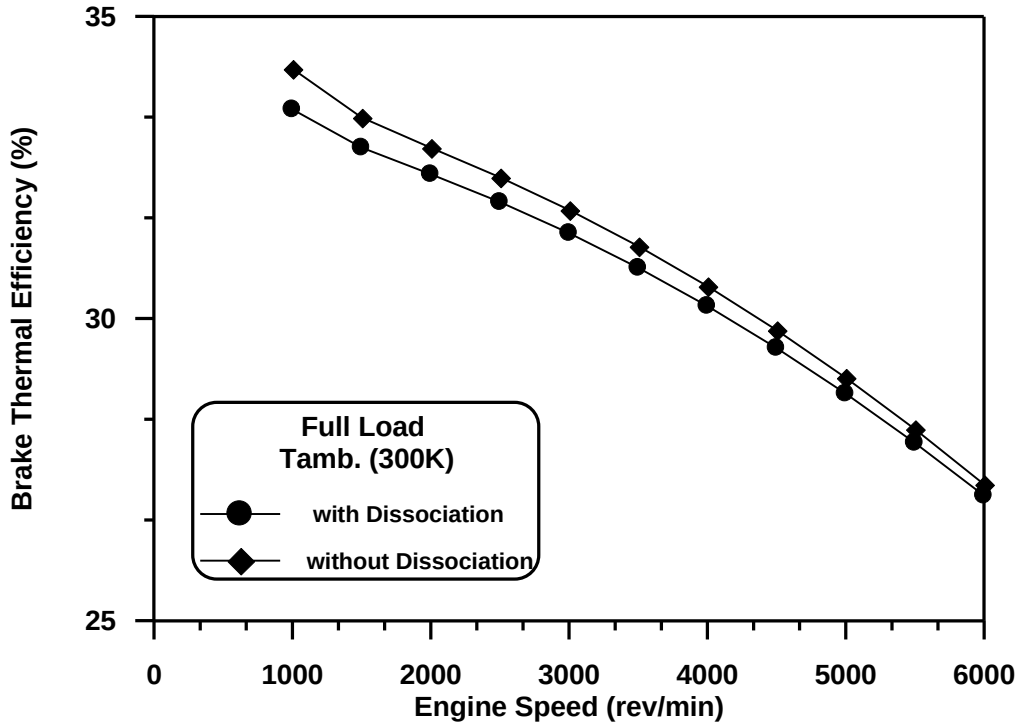


Fig. (٥.٤٩): Variation of Brake Thermal Efficiency with Engine Speed for Dissociation and without Dissociation.

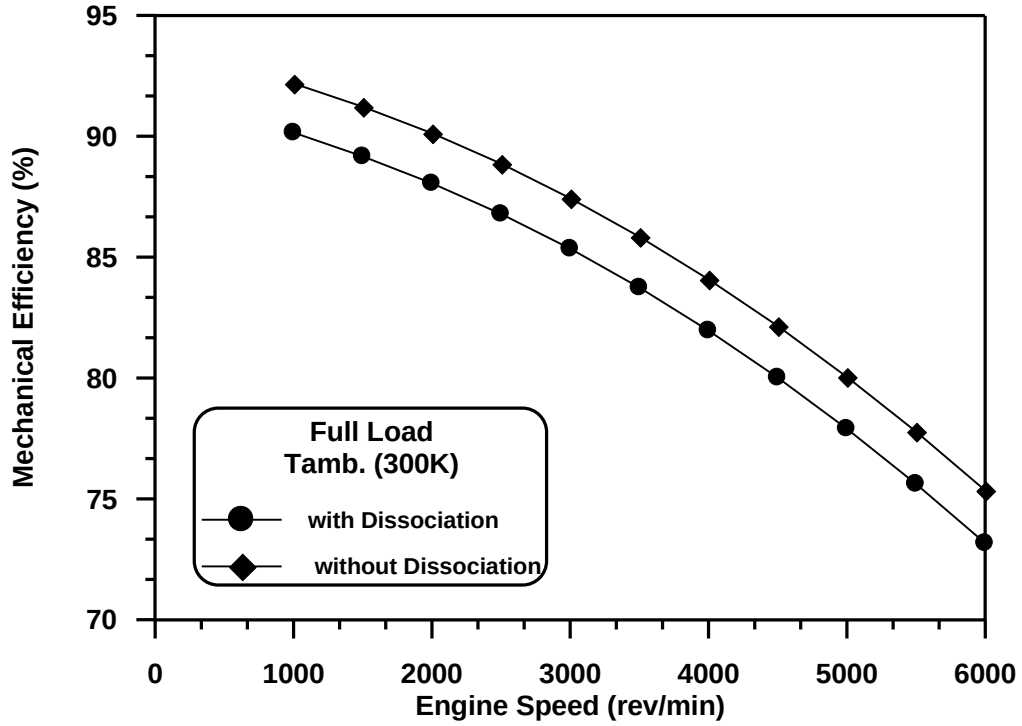


Fig. (٥.٥٠): Variation of Mechanical Efficiency with Engine Speed for Dissociation and without Dissociation.

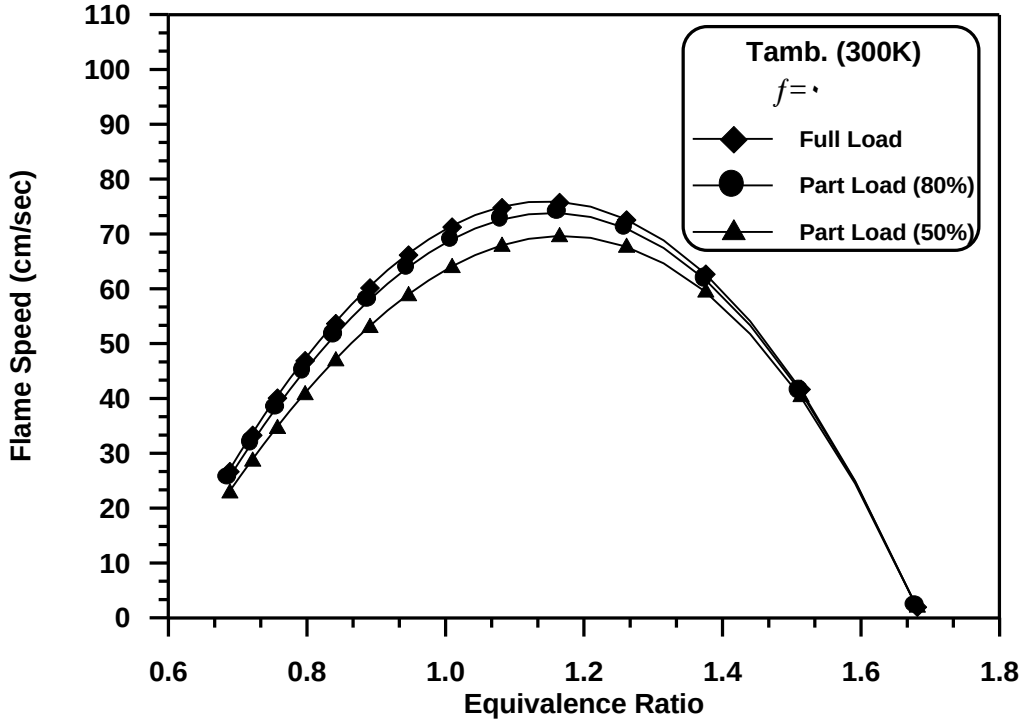


Fig. (٥.٥١): Variation of Flame Speed with Equivalence Ratio for Different Loads.

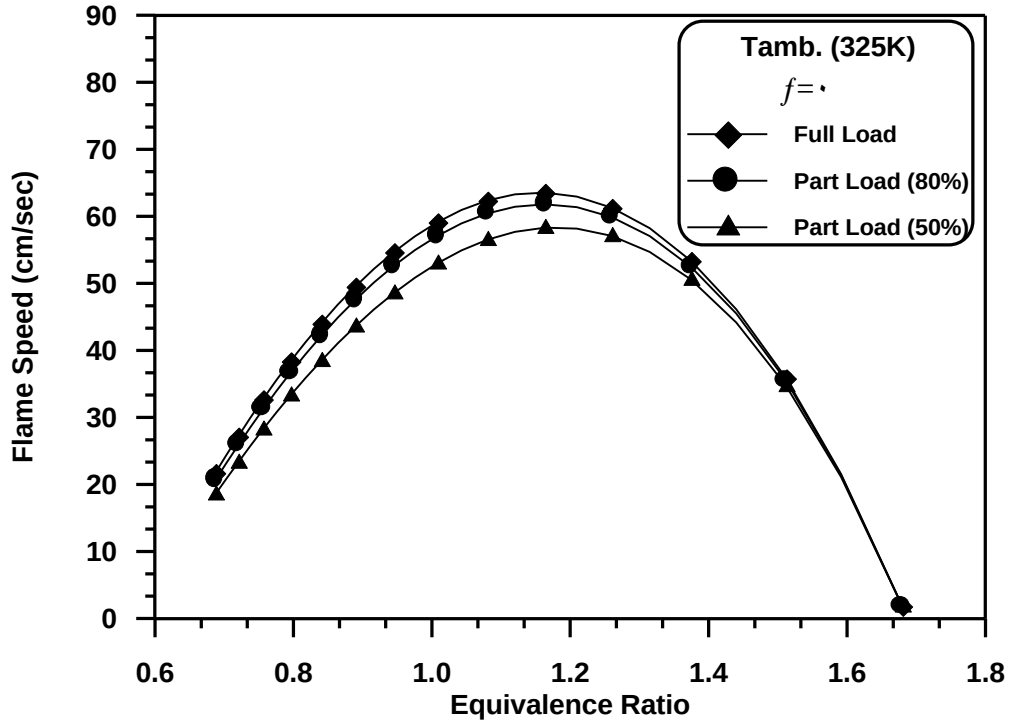


Fig. (٥.٥٢): Variation of Flame Speed with Equivalence Ratio for Different Loads.

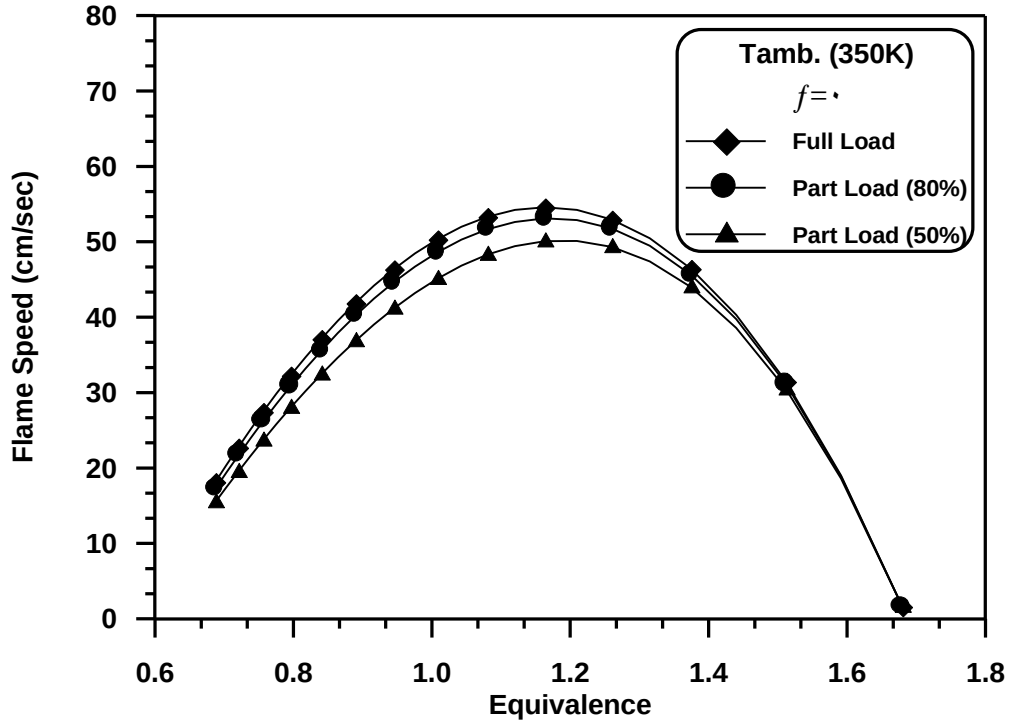


Fig. (٥.٥٣): Variation of Flame Speed with Equivalence Ratio for Different Loads.

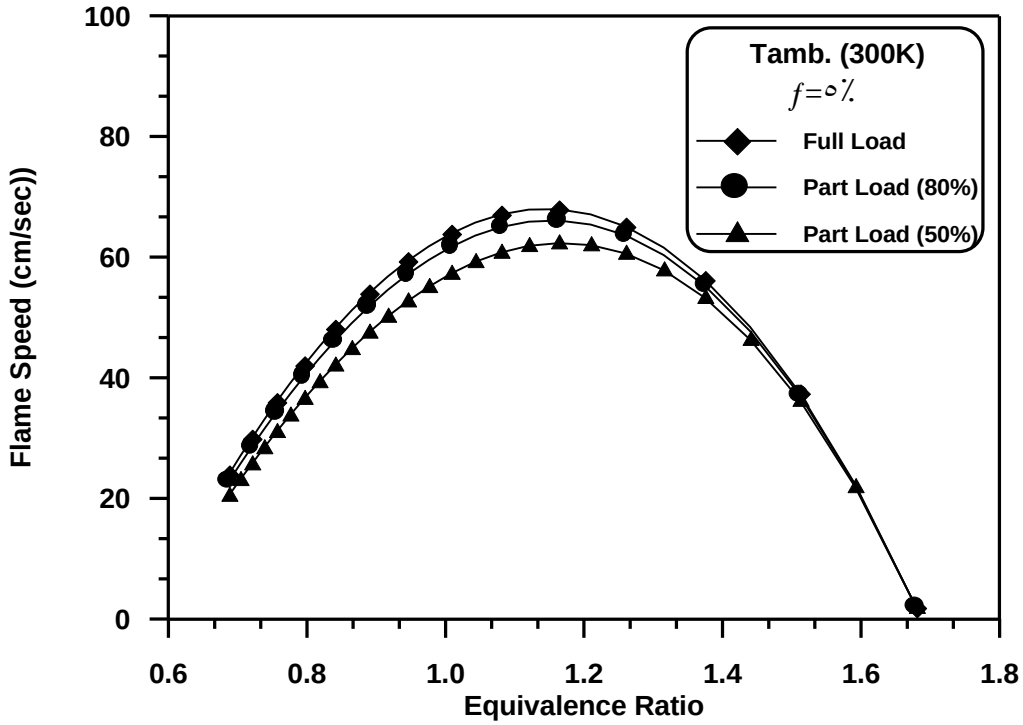


Fig. (٥.٥٤): Variation of Flame Speed with Equivalence Ratio for Different Loads.

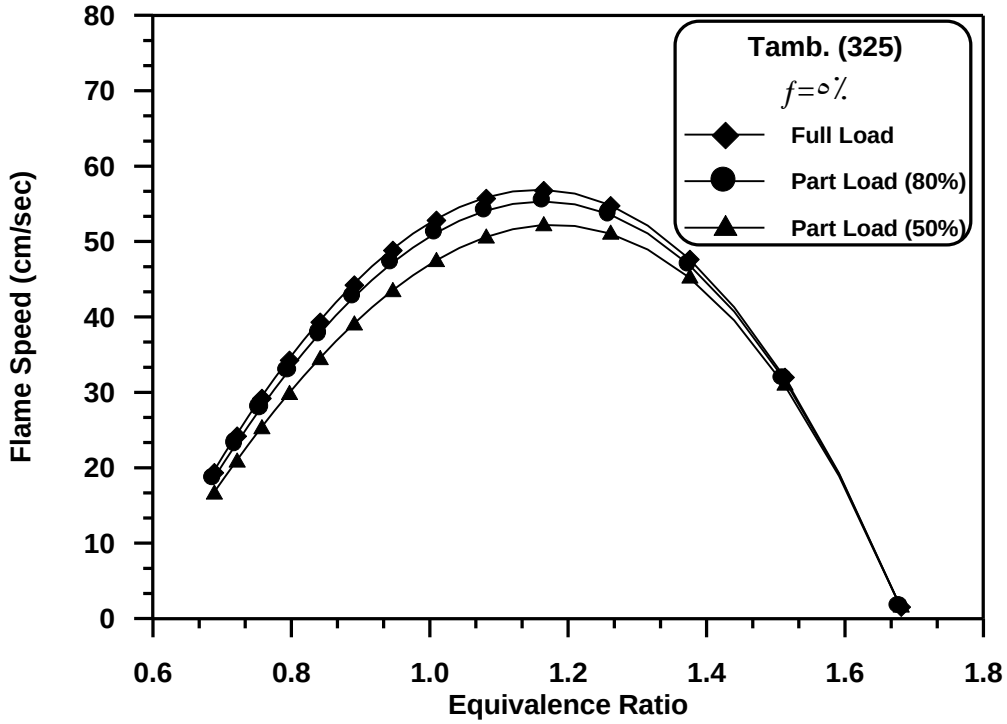


Fig. (٥.٥٥): Variation of Flame Speed with Equivalence Ratio for Different Loads.

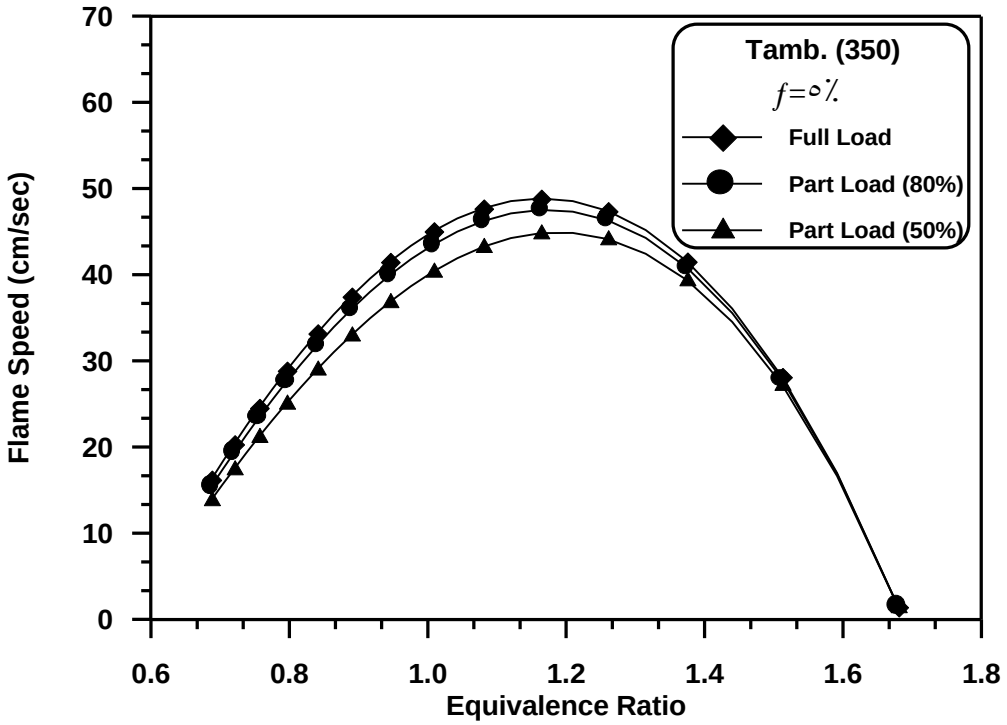


Fig. (٥.٥٦): Variation of Flame Speed with Equivalence Ratio for Different Loads.

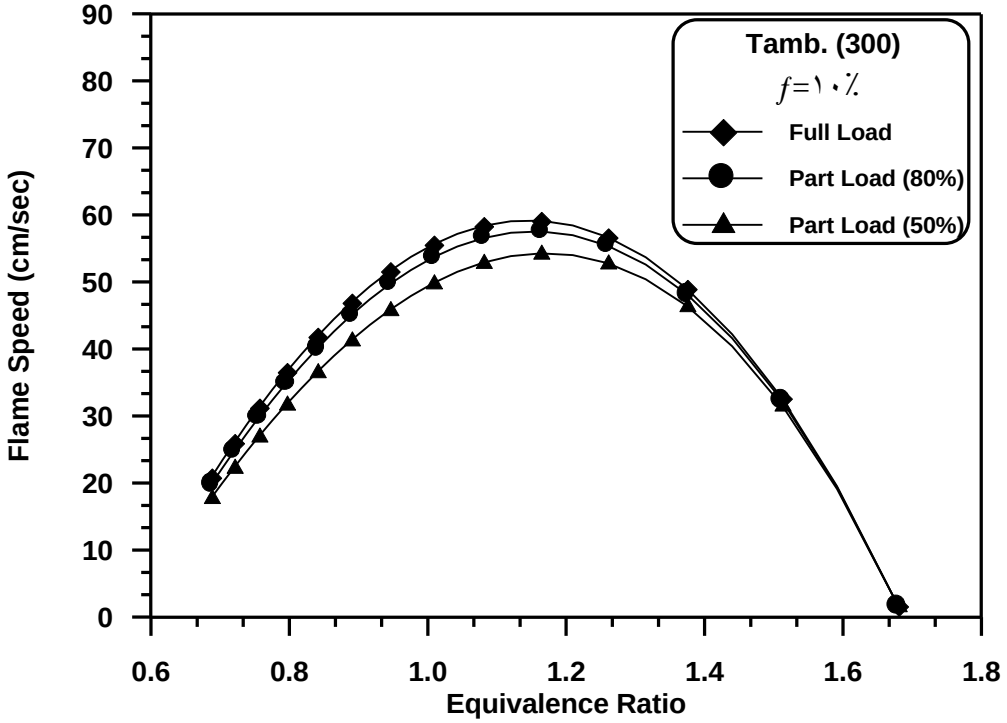


Fig. (٥.٥٧): Variation of Flame Speed with Equivalence Ratio for Different Loads.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

٦. ١: *Conclusions*

Conclusions Based on the results obtained in the present study, several conclusions may be draw. They are summarized : -

١- The detailed analysis requires more effort than the simplified one. even though the simplified analysis is useful in some general areas, the detailed analysis must be used to provide detailed design criteria.

٢- The present level of modeling is capable of predicting thermal efficiency, indicated mean effective pressure, brake mean effective pressure, peak values of gas pressure and temperature, spark advance and concentrations of dissociation.

٣- A thermodynamic model for S.I.engine cycle simulation has been presented to used for the optimal design of engine control strategies. The pressure cycle is simulated with a high level of accuracy over a large set of data after the identification of the burning time duration.

٤- These results characterize the engine behavior as it could be curves of indicated mean effective pressure, brake mean effective pressure, indicated torque, brake torque, indicated power, brake power, specific fuel consumption, thermal efficiency and volumetric efficiency at different engine speeds.

٥- The optimum spark advance which limited the optimum engine design and peak pressure after top dead center, but more spark

advance it is undesirable because of more rises in pressure (stress limit) before top dead center and this impractical. The cylinder maximum pressure decreases with the decrease the spark advance.

٦- The optimum operating engine parameters for equivalence ratio is $\phi = ١$ and then decreases for richer and leaner mixture, but, thermal efficiency is optimum, when the mixture is lean.

٧- The effect of ambient pressure and temperature very important, when the ambient pressure decrease leads to a decrease in mean effective pressure, torque, power and thermal efficiency, also; ambient temperature. The increase in power output leads to a decrease in mechanical efficiency due to friction.

٨- The effect of dissociation very important in which this leads to reduced the operating parameters, pollution and emissions due to energy loss by product of combustion, compared with without dissociation leads to increased the power, thermal efficiency etc.

٩- The maximum flame speed is at the mixture strength slightly rich and full load. And flame speed increase with increase unburned as temperature.

٦. ٢: Recommendations for Future Work

The following recommendations are suggested for future work:

١- Investigating the performance of a turbocharged spark ignition engine.

٢- Investigating the performance of a supercharger spark ignition engine.

٣- Making a comparison between a supercharger and a turbocharged .

٤- Study the effect of emission and pollution.

٥- Investigating the performance for different fuels.

٦- Investigating the performance experimentally.

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الخلاصة

النموذج يقدم مساعدة بشكل كبير لمصممي الماكينة بحيث هذا النموذج يعطي المعلومات الكاملة للماكينة للدراسة والتطوير خلال زمن وكلفة قليلين. علاوة على ذلك تعطي فهم افضل لمتغيرات الماكينة المحتواة وتأثيراتها لاداء المحرك وتعتبر أداة نافعة للحصول على افضل تصميم للمحرك لتطبيقات خاصة.

تحليل الدورة تعطي علاقات نافعة بين متغيرات الأداء مثلا نسبة الانظغاط ، نسبة الوقود إلى الهواء والوقود المستخدم.

وضع النموذج الرياضي لدراسة أداء المحرك. وهذا يحتوي على دراسة اعظم ضغط، درجة الحرارة وتوقيت القدح.النموذج الرياضي يتعامل مع محرك رباعي الشوط يعمل بالقدح غير مشحون. الطرق العددية استخدمت لحل بعض المعادلات مثل نيوتن رافسون (Newton Raphson) و رنج كوته (Runge Kutta) . برنامج حاسبي بلغة (Quick Basic) تم إعداده لإنجاز الحل.

تم دراسة أداء المحرك لظروف عمل مختلفة مثل، متوسط الضغط ، العزم ،القدرة ، معدل استهلاك الوقود والكفاءة الحرارية ووجدت افضل النتائج لهذه المتغيرات عند نسبة الوقود إلى الهواء تساوي واحد. التغير في الضغط الابتدائي ومقاومة الخليط يؤثران بشكل اكبر على الكفاءة الحرارية وأيضا على متوسط الضغط الفعال اللذان يعتمدان بصورة مباشر على كثافة الشحنة الداخلة.

تم دراسة تأثير تحليل المركبات على أداء المحرك والتي تعتبر ظاهرة مهمة للانبعاث والتلوث لدراسات لاحقة.

تم دراسة نسبة الوقود إلى الهواء على سرعة اللهب ووجدت أعلى سرعة للهبة إلى جانب الخليط الغني قليلا.