



Certification Course on SIMULATION OF IC ENGINE
PROCESSES

Resource Person Dr D. Babu

Co-ordinator: Dr V. Sathiya Murthy

Date(s) of Event: 22-03-2021 to 14-04-2021

Organizing department: Mechanical Engineering



K.S.R.M.COLLEGE OF ENGINEERING
(UGC-AUTONOMOUS)

Kadapa, Andhra Pradesh, India-516 005

Approved by AICTE, New Delhi & Affiliated to JNTUA, Ananthapuramu.

Cr./KSRMCE/(Department of ME)/2020-2021

Date: 18/03/2021

To

The Principal,

KSRM College of Engineering,

Kadapa.

Respected Sir

Sub: KSRMCE-(Department of ME) permission to conduct certification course on "SIMULATION OF IC ENGINE PROCESSES"-Request-Reg.

It is brought to your kind notice that, with reference to the cited, the ME department is planning to conduct Certification Course on "SIMULATION OF IC ENGINE PROCESSES" for B.Tech, IV Sem Students from **23 Mar, 2021 to 14 April, 2021**. In this regard I kindly request you to grant permission to conduct the certification course. This is submitted for your kind perusal.

Thanking you sir,

Yours Faithfully

[Signature]
Dr. V Sathya Murthy

Asst.Prof, Dept.ME

KSRMCE, Kadapa.

To the Director for Information
To All Deans/HoD's/IQAC

Permitted
V. S. S. Murthy

Forwarded to principal
[Signature]



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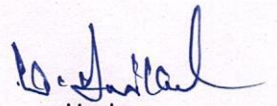
Date: 18/03/2021

Circular

All the B.Tech IV Sem ME students are hereby informed that department of MECHANICAL is going to conduct certificate course on **SIMULATION OF IC ENGINE PROCESSES** interested students may register their names on or before 21-03-2021, 5 PM

For any queries contact faculty coordinator:

Dr V. Sathiya Murthy, Asso.Prof, Dept. ME, KSRMCE, Kadapa.


Hod

Professor & Head
Department of Mechanical Engineering
K.S.R.M. College of Engineering
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Department of Mechanical Engineering
Certification Course on **SIMULATION OF IC ENGINE PROCESSES**

List of Participants

S.NO	ROLL NO	NAME OF THE STUDENT	EMAIL ID'S
1	199Y1A0301	AKULA SREEDHAR	199Y1A0301@ksrmce.ac.in
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Coordinator

HOD

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Syllabus

SIMULATION OF I.C. ENGINE PROCESSES

Course Objectives

To impart knowledge on simulation of various engine processes used in prime movers and power
To learn the simulation of engine combustion based on first and second law thermodynamics.

UNIT I

SIMULATION PRINCIPLES First and second laws of thermodynamics - Estimation of properties of gas - Structure of engine models Open and closed cycle models - Cycle studies. Chew:,,, kcactit■ns, First law ;55:5,1! ion to combustion, Heat of combustion - Adiabatic flank: I le, l'.echatlier pu:: mpG I leat transfer in engines - Heal t ran•acr mginc,. Simulation model, ior LC. I ,iices. (Ideal and actual cycle simulation, Cheinie'al ihri um and calculation of equilibrium composition.

UNIT II

SIMULATION OF COMBUSTION IN SI ENGINES

Combustion in SI engines. Flame propagation and Velocity. single /one models - Multi /one N lass hurtling rate. Turbulence models - One dimensional models- Chemical kinetics

UNIT III

COMBUSTION IN CI ENGINES

0,1111,1,lit,11 ill CI engines Single zone models - Premixed-Diffusi, models - Wiebc' model - 1110del. I /011C - Multizone models- Meguerdichian and Watson's I liro■asti.N I model Introduction to Multidimensional and spray modeling, How chart preparation

UNIT IV

STROKE ENGINES

Thermod■ Itam ot g.:, e\cliange process - Flows in engine manifolds - One dimensional and multidimensional model, I],,, around valves and through ports - Models for scavenging in two smoke engine, isothermal and non-isothermal models. Heat Transfer and Friction.

UNIT V

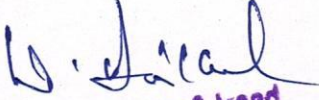
COMBUSTORS, Gas Turbine Power plant, - flame stahi I . Combustion models for Steady Flow Simulation – isothermal models..

OUTCOMES

- Able to understand the program with the heat transfer model and insert the correct modifications to perform a simulation of a non-adiabatic engine;
- Add a new combustion model, replacing the existing one used in the initial program. The new combustion model would take into account the turbulence in the cylinder and would then allow the variation of burn duration (which is fixed in the simple model used) to vary with engine speed.

Text Books

1. Domkundwar V, A Course in Internal Combustion Engines, Dhanpat Rai & Co. (P) Ltd,, 2002
2. Gancsan. V. Internal Combustion Engines, Tata McGraw I-fill Book Co., 2003,
3. John B. Heywood. Internal Combustion Engine Fundamentals. McGraw Ifill Boo . 1998.
4. Mathur. Ml.. and Sharma. R.P.. A Course in Internal Combustion Engines, Dhanpat Rai Publications Pvt.New Delhi-2, 1993.


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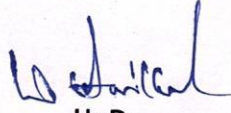
Department of Electrical and Electronics Engineering
Certification Course on Simulation of IC Engine Processes

Schedule

Timing: 4:00pm – 6:00pm

S.No	Date	Course Coordinator	Topic Covered
1	22-03-2021	Dr V. Sathiya Murthy	To learn the simulation of engine combustion based on first and second law thermodynamics.
2	23-03-2021	Dr D. Babu	First and second laws of thermodynamics
3	24-03-2021	Dr V. Sathiya Murthy	-Estimation of properties of gas - Structure of engine models Open and closed cycle models - Cycle studies
4	25-03-2021	Dr D. Babu	Ideal and actual cycle simulation,
5	26-03-2021	Dr V. Sathiya Murthy	Combustion in SI engines. Flame propagation and Velocity.
6	27-03-2021	Dr D. Babu	single /one models - Multi /one N loss hurtling rate. Turbulence models
7	30-03-2021	Dr V. Sathiya Murthy	Chemical Equilibrium and calculation of equilibrium composition
8	31-03-2021	Dr D. Babu	One dimensional and multidimensional mode
9	01-04-2021	Dr V. Sathiya Murthy	One dimensional models- Chemical kinetics
10	03-04-2021	Dr D. Babu	Flows in engine manifolds -
11	06-04-2021	Dr V. Sathiya Murthy	Models for scavenging in two stroke engine
12	08-04-2021	Dr D. Babu	Isothermal and non-isothermal models. Heat Transfer and Friction.
13	09-04-2021	Dr V. Sathiya Murthy	Gas Turbine Power plant
14	12-04-2021	Dr D. Babu	Combustion models for Steady Flow Simulation
15	15-04-2021	Dr V. Sathiya Murthy	Isothermal models.


Coordinator


HoD

Professor & Head
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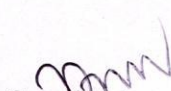
Department of Mechanical Engineering

Activity Report

Name of the Event	: Certification Course on SIMULATION OF IC ENGINE PROCESSES
Duration of the Event	: 22-03-2021 to 14-04-2021
Scheduled Time	: 4.00 to 6.00PM
Target Audience	: B.Tech IV Sem Students
Course Coordinator	: Dr V. Sathiya Moorthy

Activity Description:

SIMULATION OF IC ENGINE PROCESSES is one of the latest tool for Mechanical students. Department of ME organized a Certificate Course on "SIMULATION OF IC ENGINE PROCESSES". Head of the Department, faculty & participants of the Course inaugurated with all good spirit. Resource persons began the first day first session To learn the simulation of engine combustion based on first and second law thermodynamics. Finally valedictory. Students were issued participation certificates by Head of the Department.


Coordinator


HOD

Professor & Head
Department of Mechanical Engineering
K.S.R.M. College of Engineering
KADAPA - 516 003.


Principal

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Kadapa, Andhra Pradesh, India- 516 003

Certificate Course on Simulation of IC Engine Processes

22/03/2021 to 14/04/2021

Organized by
**DEPARTMENT
OF
MECHANICAL ENGINEERING**

17	KONERU SAI KARTHIK	199Y1A0318	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
18	KOTHAPALLE VAMSIDHAR REDDY	199Y1A0319	✓	A	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
19	KUMMARI MANJUNATH	199Y1A0320	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
20	KUMMETHA SAI KUMAR REDDY	199Y1A0321	✓	✓	✓	A	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A
21	L M VINAY KUMAR	199Y1A0322	✓	✓	A	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
22	MALEPATI SIVA SAI REDDY	199Y1A0324	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	✓
23	MANJUNATHA DINESH KUMAR	199Y1A0325	A	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓
24	MARKAPURAM MYSORA REDDY	199Y1A0326	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
25	MEDIMALA KIRAN KUMAR	199Y1A0327	✓	✓	✓	✓	✓	A	✓	✓	A	✓	✓	✓	✓	✓	✓	✓
26	MOGHAL JUNAID BAIG	199Y1A0328	✓	✓	✓	✓	✓	A	✓	✓	A	✓	✓	✓	✓	✓	✓	✓
27	ACHUKATLA NUMAIR	209Y5A0301	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
28	BHOJANAPALLE NAGA SIVA	209Y5A0302	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	A	✓	✓	✓	✓	✓
29	BIJJE PURUSHOTHAM	209Y5A0303	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	A	✓	✓
30	BUCHUPALLI SIVA PRASAD REDDY	209Y5A0304	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	✓
31	CHINNI GURU PRASAD	209Y5A0305	A	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	✓	✓	✓
32	CHINTHAGINJALA VENKATA SUBBARAYUDU	209Y5A0306	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	A	✓	✓	✓	✓
33	CHITRALA VENKATA SWAMY SETTY	209Y5A0307	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
34	DAKALA SRINIVASULU	209Y5A0308	✓	✓	✓	✓	✓	A	✓	✓	✓	A	✓	✓	✓	✓	✓	✓
35	DHARA SUNIL KUMAR	209Y5A0309	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
36	ESLAVATH RAVI NAIK	209Y5A0310	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
37	GORLA CHARAN KUMAR REDDY	209Y5A0311	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	✓	✓	✓	✓	✓
38	GUDISHA DILIP KUMAR	209Y5A0312	✓	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	✓	A	✓
39	GUTTURU GIRISHKUMAR REDDY	209Y5A0313	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	✓	A	✓	✓	✓
40	JAMPANGI OBULESU	209Y5A0314	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	A	✓	✓	✓
41	JONNADULA SATISH	209Y5A0315	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
42	KOTA UPENDRA REDDY	209Y5A0316	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	A	✓	✓	✓
43	KUNU SIVA BABJI	209Y5A0317	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A
44	KURUVA AJAY KUMAR	209Y5A0318	A	✓	A	A	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓
45	LAKKINENI SUDHARSHAN	209Y5A0319	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
46	LANKAMSETTY VENKATA LOKESH	209Y5A0320	✓	A	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	✓	✓	✓	✓
47	LINGAMBOTI BHUSHAN	209Y5A0321	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
48	MEALLA GOWTHAMSAI	209Y5A0322	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓
49	MEDIREDDY BHARATH REDDY	209Y5A0323	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	A	✓	✓	✓

50	MOKA VEERABHADRA	209Y5A0324	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
51	MOOLI CHENNAKESAVA REDDY	209Y5A0325	✓	A	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
52	NEELAM PAVAN KUMAR	209Y5A0326	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
53	PATTU MONESH	209Y5A0327	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
54	PINNAPURAM MADHU SUDHAN	209Y5A0330	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
55	POREDDY HARI VARDHAN REDDY	209Y5A0331	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	✓	✓	✓
56	PRODDATURU NAZEER BASHA	209Y5A0332	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓	✓	✓	✓	✓
57	YATAGIRI HEMANTH KUMAR	209Y5A0343	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	✓	✓
58	YEDDULADODDI ASHOK	209Y5A0344	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	A	✓	✓	✓	✓

Coordinator

HOD

Professor & head
Department of Mechanical Engineering
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Introduction

This report presents the Thermodynamics theory describing the main physical phenomena occurring inside a spark ignition four stroke (4S) internal combustion engine (ICE) while it is running at steady speed (constant revolutions per minute, rpm). The mathematical form of the Thermodynamic theory is developed and implemented numerically by way of a "Scilab" computer program. The result is an ICE computer simulation. This computer simulation may be used to obtain some fairly good estimates of engine performance in which the main effects of compression ratio, sparks timing, some aspects of valve timing, valve sizing, and fuel types, over a range of engine speeds.

Of course not every detail of ICE performance can be accounted for, but depending on the physical details incorporated and their relative importance, many of the most important performance characteristics can be determined to a reasonable degree of accuracy. This report does not deal with any structural or mechanical aspects of an ICE beyond those of the basic geometric features relevant to the containment and external manifestations of the Thermodynamics processes occurring in the engine. These thermodynamic processes are idealized to a certain degree in order to reduce the complexity at this stage of development of the engine simulation.

The simulation is based on the standard configuration of a reciprocating piston in a cylinder closed at one end, the cylinder 'head'. The piston is connected to a crank by way of a connecting rod that protrudes out the opposite open end of the cylinder and connects to a crank. Figure 1 is a schematic diagram of one cylinder of an ICE. The resulting reciprocating motion of the piston imparts a rotation to the crank. This basic slider-crank mechanism (the piston being the slider) transmits power generated by a working fluid, or gas, in the space enclosed by the piston, cylinder and cylinder head, to whatever is connected to crank. The crank is also geared to a camshaft that operates the valves in the cylinder head that periodically open and close to expel or inhale the working gases. Most ICE's have multiple cylinders operating in unison on a common crankshaft. The processes that occur are essentially identical for each cylinder so that the analysis need be done for only one cylinder. The ICE performance is then simply the number of cylinders times the input/output for a single cylinder.

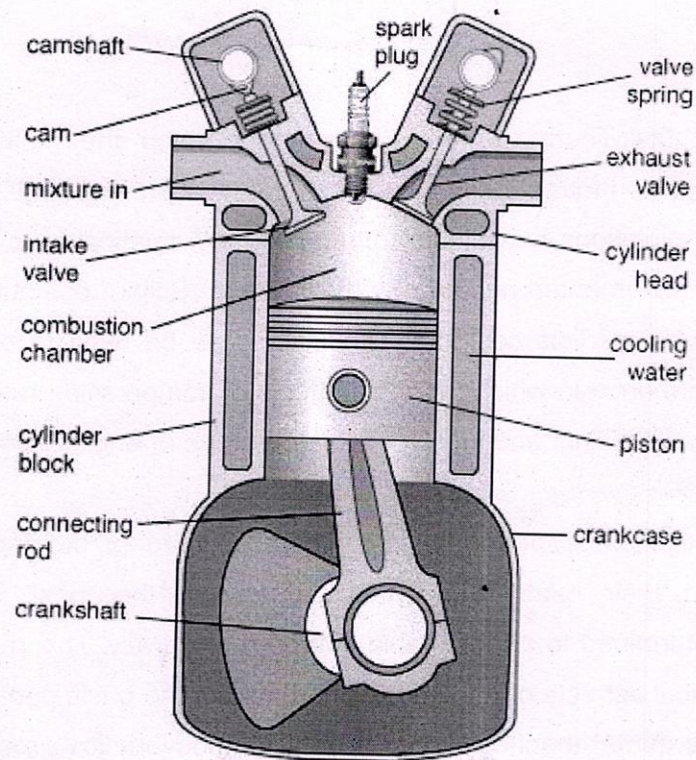


Figure 1 – Four stroke internal combustion engine. [5]

The ICE Thermodynamics analysis is based on the following primary assumptions. All thermodynamics processes are assumed to be internally reversible. The working medium (fuel and air mixtures) is assumed to be an ideal gas with constant specific heats. The equations of state for the burned and unburned media are derived on the basis of equilibrium chemistry. The gas exchange process is based on quasi-steady compressible flow through an orifice.

Further secondary assumptions and idealizations are discussed in the formulation of the thermodynamics model in the next chapters.

Chapter 1- Internal Combustion Engine

1.1.The Basic ICE Mechanism

The piston cylinder-crank mechanism (the slider-crank) is shown schematically in Figure 2. This figure indicates how the up and down motion of the piston turns the crank. The space enclosed by the piston and the cylinder is the main concern here. This is where the latent energy of the fuel-air mixture is released by combustion (oxidized) to produce the sensible energy, which drives the piston. The top of the cylinder enclosure contains an intake and exhaust valve which open and close at appropriate moments of the engine cycle to allow escape of burned gases and ingestion of fresh fuel-air mixture.

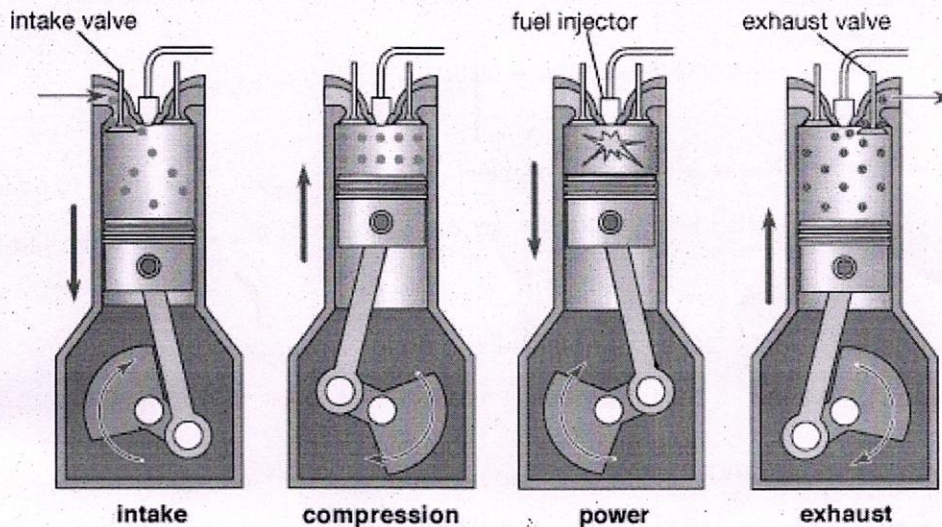


Figure 2 - A four-stroke spark ignition cycle. [5]

The basic engine performance cycles are controlled by the crank rotation. The crank rotation in turn moves the piston up and down, thus varying the volume V of the space enclosed by the piston and cylinder. This varying volume is the primary controlling factor of the sequence of thermodynamic events occurring in the piston-cylinder space. Henceforth this space will be referred to simply as the cylinder.

The crank rotation is measured in terms of the rotation angle θ shown in Figure 3.

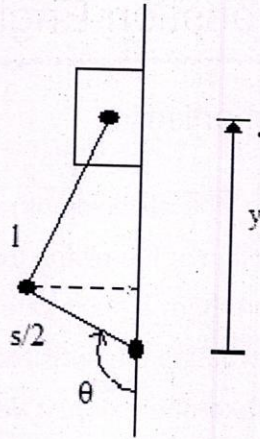


Figure 3 - Sketch of the slider crank model of piston-cylinder geometry

At $\theta=0$ ($2*n*\pi$) the piston is at the bottom-most point in its travel. This point is called bottom center, BC. The cylinder volume $V(\theta)$ can be shown, by an analysis of the slider-crank mechanism to be,

$$v(\theta) = V_m * \left(\frac{1}{r_c} + \frac{1}{2} * \left(1 - \frac{1}{r_c} \right) * (1 + \cos(\theta) + R_c - \sqrt{(R_c - \sin(\theta))^2}) \right) \quad (1.1)$$

In formula (1.1), V_m is the (maximum) volume in the cylinder at BC, R_c is the ratio of connecting rod length to s , where $s = \text{stroke}$, and r_c is the compression ratio $\frac{V_m}{V_c}$, where V_c is the (minimum) volume of the cylinder at top center (TC) or $\theta = \pi$ ($2*(n-1)*\pi$). V_c is called the clearance volume and $V_d = V_m - V_c$ is the "displacement" volume, the usual measure of engine capacity or, more commonly, engine size.

The calculation of the instantaneous volume equation (1.1) is discussed in "Appendix A.1".

Using V_m as an input variable, some other variables such as the V_c , V_d , bore (bo) and stroke (s) have to be calculated in order to proceed. It was assumed that the bore was equal to the stroke in order to simplify some equations and to use the minimum ones possible,

$$V_d = \frac{\pi}{4} * b^2 * s \quad (1.2)$$

$$\frac{V_d}{V_c} = rc - 1 \quad (1.3)$$

$$V_c = \frac{V_m}{rc} \quad (1.4)$$

$$h_c = \frac{4 * V_c}{\pi * b_o} \quad (1.5)$$

assuming that the shape of the clearance volume is a cylinder with diameter b_o and height, h_c (Figure 4).

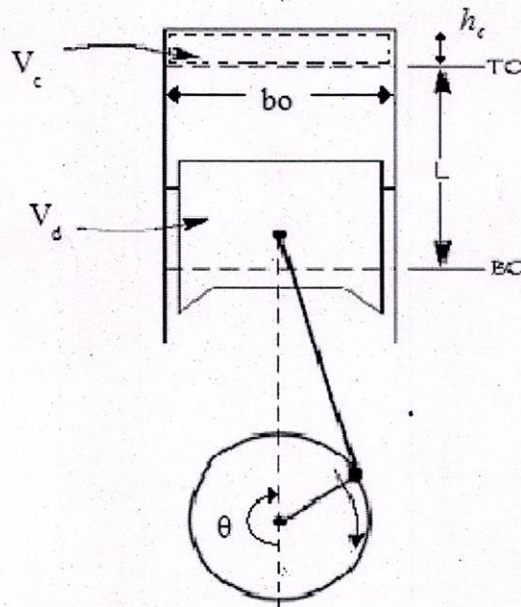


Figure 4 – Basic geometry of the internal combustion engine

The exposed total cylinder area $A_w(\theta)$ is the sum of the cylinder clearance area and the displacement area,

$$A_w = A_d + A_c \quad (1.6)$$

$$A_w = \frac{v - V_c}{bo/8} + A_c \quad (1.7)$$

and

$$A_c = \pi * bo * h_c + \frac{\pi}{2} bo^2 \quad (1.8)$$

The operation of the valves is synchronized to the motion of the piston by way of gear or chain drives from the crankshaft. This is not shown in Figure 2. There will be no need for a description of this mechanism here since it will not be made use of. For the present it is only necessary to describe the valve configuration and actual motion of the valves as a function of the crank angle θ . This will be reserved for the chapter on the gas exchange process in order to keep the exposition simple at this stage. The function, of θ , describing the motion of the valves is given as part of the basic engine specifications utilized in this study. The camshaft and valve actuation mechanism must then be designed to realize this valve motion function. For this the reader is referred to books on engine and mechanism design given in the bibliography.

1.2. The Equations of State of the Working Gases

This gaseous mixture is assumed to be an ideal gas, albeit with different equations of state in the unburned and burned states. The equations of state of the unburned (subscript u) and burned (subscript b) are derived on the basis of combustion equilibrium chemistry and the coefficients of the thermodynamic properties are given in Reference 1. These two thermodynamic laws apply to the gaseous fuel-air mixture in the cylinder. The linearized versions of the equations are used here. For illustrative purpose the fuel used here is CH_4 with an equivalence ratio of 1 and whose properties are very similar to gasoline. During the process of combustion the cylinder contains a mixture of the

unburned and burned fuel-air. The mixture is quantified by the burned to total mass ratio $x = \frac{m_b}{m}$,

where $m = m_b + m_u$. The equations of state for each of the gases is

$$PV_u = m_u * R_u * T \quad (1.9)$$

and

$$u_u = C_{v_u} * T + hf_u \quad (1.10)$$

$$PV_b = m_b * R_b * T \quad (1.11)$$

and

$$u_b = C_{v_b} * T + hf_b \quad (1.12)$$

then, if the gases are homogeneously mixed,

$$PV = m * (xR_b + (1-x)R_u) * T = mRT \quad (1.13)$$

and

$$U = m * u = m * (x * C_{v_b} + (1-x) * C_{v_u}) * T + x * hf_b + (1-x) * hf_u \quad (1.14)$$

CVK utilized the homogeneously mixed charge described by equations (1.13) and (1.14), where

$\frac{dx(t)}{dt}$ is an empirically determined burning rate.

One of the main aims of this study is to implement a more realistic combustion model in which unburned and burned gases remain separated, ie., a "two zone" model.

1.3. Thermodynamics and Mathematical Model of the Engine

The engine operates in a two-cycle (4 stroke) mode. Each cycle consists of a complete rotation, through an angle of 2π , of the crank. The first cycle is called the power cycle in which both valves are closed and the power of the engine is produced. This cycle consists of the compressions stroke roughly from $\theta=0$ to $\theta=\pi$, in which the fuel-air mixture in the cylinder is compressed, followed by the expansion stroke, roughly from $\theta=\pi$ to $\theta=2\pi$, in which the main positive engine work is done. The second cycle is called the gas exchange cycle in which the burned gases from the expansion stroke are expelled (exhaust stroke, $\theta=2\pi$ to $\theta=3\pi$) and fresh fuel-air is ingested (intake stroke, $\theta=3\pi$ to $\theta=4\pi$).

These two cycles are completely described by the main two laws of thermodynamics governing the unburned and burned fuel-air mixture in the cylinder. The two laws are the Conservation of Mass,

$$\frac{dm}{dt} = \dot{m}_i - \dot{m}_e \quad (1.15)$$

and the Conservation of Energy (1st Law of Thermodynamics),

$$\frac{dU}{dt} = \dot{Q} - \dot{W} + \dot{H}_i - \dot{H}_e \quad (1.16)$$

Here m is the mass of burned plus unburned fuel-air mixture in the cylinder at any instant of time t , \dot{m}_i and \dot{m}_e are the mass flow rates into and out the cylinder, respectively \dot{Q} , \dot{W} , \dot{H}_i and \dot{H}_e is the heat rate into or out of the cylinder, the work rate into or out of the cylinder, the enthalpy flow rate into the cylinder, and the enthalpy flow rate out the cylinder respectively.

In the application of these equations to the analysis of the engine cycles at constant speed (constant crank angular velocity, ω), it is the most convenient to transform time t to angle θ by the equation $\omega * dt = d\theta$. Then the time derivatives in equations (1.15) and (1.16) are replaced by angle derivatives and the \sim over symbols is replaced by \wedge over the symbols to signify that rates are with respect to angle instead of time. Then,

$$\frac{dm}{d\theta} = \hat{m}_i - \hat{m}_e \quad (1.17)$$

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} + \hat{H}_i - \hat{H}_e \quad (1.18)$$

Chapter 2 - Power Cycle

This chapter presents the thermodynamics theory describing the main physical phenomena occurring inside an ICE.

The thermodynamic models of the four movements, or strokes, of the piston before the entire engine firing sequence is repeated, are described in this chapter and chapter 3.

2.1. Introduction

In this cycle, the valves are closed so there is no mass exchange and it is where the main power of the engine is produced.

This cycle consists in a complete rotation which is characterized by two stages:

(a) Compression stage - roughly from $\theta=0$ to just before the spark plug goes off,

$\theta_s = 0.88 * \pi$, in which the fuel-air mixture in the cylinder is compressed. The value of θ_s given is typical. All engines start combustion before TC ($\theta=\pi$). This called spark advance.

(b) Combustion stage - roughly from θ_s to just before exhaust valve opens,

$\theta_{evo} = 2 * \pi - \delta$, where $\delta > 0$ depending on some factors, such as the flame speed and piston speed. Combustion begins during compression and most expansion.

In the beginning of this cycle, the cylinder and the combustion chamber are full of the low pressure fresh fuel/air mixture and residual (exhaust gas), as the piston begins to move, the intake valve closes. With both valves closed, the combination of the cylinder and combustion chamber form a completely closed vessel containing the fuel/air mixture. As the piston is pushed to the TC, the volume is reduced and the fuel/air mixture is compressed during the compression stroke.

As the volume is decreased because of the piston's motion, the pressure in the gas is increased, as described by the laws of thermodynamics.

Sometime before the piston reaches TC of the compression stroke, the electrical contact is opened. The sudden opening of the contact produces a spark in the combustion chamber which ignites the fuel/air mixture. Rapid combustion of the fuel releases heat, and produces exhaust gases in the combustion chamber. Because the intake and exhaust valves are closed, the combustion of the

fuel takes place in a totally enclosed (and nearly constant volume) vessel. The combustion increases the temperature of the exhaust gases, any residual air in the combustion chamber, and the combustion chamber itself. From the ideal gas law, the increased temperature of the gases also produces an increased pressure in the combustion chamber. The high pressure of the gases acting on the face of the piston cause the piston to move to the BC which produces work.

Unlike the compression stroke, the hot gas does work on the piston during the expansion stroke. The force on the piston is transmitted by the piston rod to the crankshaft, where the linear motion of the piston is converted to angular motion of the crankshaft. The work done on the piston is then used to turn the shaft, and to compress the gases in the neighboring cylinder's compression stroke.

As the volume increase during the expansion, the pressure and temperature of the gas tends to decrease once the combustion is completed.

2.2.Compression stage

2.2.1 Thermodynamic Model of the compression stage

During this stage, the energy balance on the in-cylinder gas is,

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} \quad (2.1)$$

As both valves are closed there is no mass exchange so

$$\frac{dm}{d\theta} = \hat{m}_i = \hat{m}_e = 0 \quad (2.2)$$

After the algebraic manipulation shown in "Appendix A.2" equation (2.1) becomes,

$$\frac{dT}{d\theta} = \frac{\hat{Q}}{m \cdot A} - (xR_b + (1-x)R_u) \frac{T}{AV} \frac{dV}{d\theta} - \frac{\Delta C_v T + \Delta h_f}{A} \frac{dx}{d\theta} \quad (2.3)$$

where the quantity $A = x \cdot \Delta C_v + C_{v_u}$

At this stage, the mass of gases in the cylinder is primarily unburned. However, there is a small amount of burned gas, called residual gases (m_r) which remains after the exhaust stroke. These

gases are homogeneously mixed, where $x = \frac{m_r}{m}$. Thus equations (1.13) and (1.14) do apply for this part of the cycle.

2.2.2 Heat transfer

Heat transfer plays an important role inside an ICE because it affects the engine performance, efficiency, and emissions.

"The peak burned gas temperature in the cylinder of an internal combustion engine is of order 2500K. Maximum metal temperatures for the inside of the combustion chamber space are limited to much lower values by a number of considerations, and cooling for the cylinder head, cylinder, and piston must therefore be provided. These conditions lead to heat fluxes to the chamber walls that can reach as high as 10 MW/m² during the combustion period." [1]

In regions of high heat transfer, it is necessary to estimate it in order to avoid thermal stresses that would cause fatigue cracking in the engine's materials ("temperatures must be less than about 400°C for cast iron and 300°C for aluminum alloys" [1])

The critical areas due to the heat transfer inside an ICE are the engine's piston which is exposed to the gases at the combustion chamber and exhaust system that contains the exhaust valve which is exposed to the exhaust gases that flow past it at high velocities (making for good heat transfer).

For a given mass of fuel within the cylinder, higher heat transfer to the combustion chamber walls will lower the average combustion gas temperature and pressure, and reduce the work per cycle transferred to the piston. Thus specific power and efficiency are affected by the magnitude of engine heat transfer.

The source of the heat flux is not only the hot combustion gases, but also the engine friction that occurs between the piston rings and the cylinder wall which will not be contained in this work. Heat transfer due to the friction is negligible.

"The maximum heat flux through the engine components occurs at fully open throttle and at maximum speed. Peak heat fluxes are on the order of 1 to 10 MW/m². The heat flux increases with increasing engine load and speed. The heat flux is largest in the center of the cylinder head, the exhaust valve seat and the center of the piston. About 50% of the heat flow to the engine coolant is through the engine head and valve seats, 30% through the cylinder sleeve or walls, and the remaining 20% through the exhaust port area." [3]

Therefore, heat transfer is a very important parameter in an engine because it is required for a number of important reasons, including engine's performance and efficiency, material temperature limits, lubricant performance limits, emissions, and knock (see appendix B.1).

a) Heat transfer modeling

In the previous equations, the differential heat transfer is represented by \hat{Q} .

The differential heat transfer \hat{Q} to the cylinder walls can be calculated if the instantaneous average cylinder heat transfer coefficient $h_g(\theta)$ and engine speed N (=rpm) are known.

The average heat transfer rate at any crank angle θ to the exposed cylinder wall at an engine speed N is determined with a Newtonian convection equation:

$$\hat{Q} = h_g(\theta) A_w(\theta) (T(\theta) - T_w) / N \quad (2.4)$$

The cylinder wall temperature T_w is the area-weighted mean of the temperatures of the exposed cylinder wall, the head, and the piston crown. The heat transfer coefficient $h_g(\theta)$ is the instantaneous averaged heat transfer coefficient. At this stage, the exposed cylinder area $A_w(\theta)$ is the sum of the cylinder bore area, the cylinder head area and the piston crown area, assuming a flat cylinder head.

b) Heat transfer coefficient

The instantaneous heat transfer coefficient, $h_g(\theta)$ during the power cycle depends on the gas speed and cylinder pressure, which change significantly during the combustion process.

There are two correlations that are used to get the heat transfer coefficient, the Annand and the Woschni correlation. However, to compute the heat transfer coefficient it was used an empirical formula for a spark ignition engine given in Han et al.(1997),

$$h_g = 687 * P^{0.75} * U^{0.75} * b_o^{-0.25} * T^{-0.465} \quad (2.5)$$

with some slightly modifications.

The units of h_g , P , U , b and T are W/m^2K , kPa , m/s , m and K , respectively.

The heat transfer coefficient can also be obtained using the averaged heat transfer coefficient correlation of C. F. Taylor ("The Internal Combustion Engine in Theory and Practice", MIT Press, 1985),

$$\frac{h*b}{k} = 10.4 * m^{(3/4)} \left(\frac{U*b}{V*\mu} \right)^{(3/4)} \quad (2.6)$$

where k is gas thermal conductivity and μ the gas kinematic conductivity.

However, this formula can be manipulated into the form,

$$h_g = C' * P^{0.75} * U^{0.75} * b o^{-0.25} * T^{-0.75} \quad (2.7)$$

Formula (2.5) differs from this only by the coefficient C' and the power of T . We have found that the value of $C'=300$ along with the power of T in formula (2.6) seems to be more reasonable.

$$h_g = 300 * P^{0.75} * U^{0.75} * b o^{-0.25} * T^{-0.465} / 1000 \quad (2.8)$$

Note that the unit used for energy in the "Scilab" simulation was kJ , so, equation (2.8) was divided by 1000 to get everything in the same units.

In equation (2.8), U is an empirical piston speed and calculated using the equation,

$$U = 0.494 * \overline{U_p} + 0.73 * 10^{-6} * (PdV + VdP) \quad (2.9)$$

where

$$\overline{U_p} = \frac{(2 * N * S)}{60} \quad (2.10)$$

is the actual piston speed. It is very unclear what the last term, $(PdV + VdP)$, in equation (2.9)

means. Since the coefficient of this term is very small, we left it out.

2.3. Combustion stage

In the power cycle chemical combustion commences with spark ignition at the point θ_s of crank angle just before BC on the compression stroke, $\theta_s = 0.88 * \pi$.

The combustion processes that occur in an ICE are very complex and there are many types of models which can describe it. The combustion model used by CVK was based on a single zone homogeneously mixed burning mass with a constant burning rate. This is a model with empirical burning rates that does fairly well, but does not capture true combustion rates.

We now introduce a somewhat more sophisticated combustion model. In the one zone model firstly used, the rate of combustion is assumed to be proportional to crank angle. This is the main assumption we would like to remove. It is well known that the combustion rate has its own dynamics and does not follow crank angle. The combustion rate depends on the dynamics of the gas motion inside the cylinder, especially the turbulence level. As engine speed increases, the turbulence level increases and thus combustion rate increase, but it does not increase at the same rate as engine speed. In order to capture this effect and assess its impact on engine performance we now develop a two zone combustion model. The model used consists in a two zone analysis of the combustion chamber which contains an unburned and burned gas region separated by a turbulent flame front. The flame front progresses at a turbulent flame speed. The turbulence model used that determines the flame speed is given in [2].

2.3.1 Combustion modeling

In an ICE the fuel and air are mixed together in the intake system, inducted through the intake valve into the cylinder, where mixing with residual gas take place, and then compressed. Under normal operating conditions, combustion is initiated towards the end of the compression stroke at the spark plug by an electric discharge. Following inflammation, a turbulent flame develops, propagates through this premixed fuel, air, burned gas mixture until it reaches the combustion chambers walls, and then extinguishes.

From this description it is plausible to divide the combustion process into three distinct phases:

- (1) Spark ignition
- (2) Combustion development

(3) Combustion termination

The understanding of each of these phases will be developed next.

(1) Spark Ignition

Ignition is treated as an abrupt discontinuity between the compression and burn stages with the instantaneous conversion of a specified mass fraction f of the reactants to products. This produces the unburned and burned zone, each assumed to be homogeneous and hence characterized by its own single state.

Using the 1st law equation for burned gas developed later in this section

$$M_b * C_{v_b} \frac{dT_b}{dt} = \dot{Q}_b - P \frac{dV_b}{dt} + \dot{M}_b * (C_{p_u} * T_u - C_{v_b} * T_b + hf_u - hf_b) \quad (2.11)$$

integrate it over a small time interval Δt (or $\Delta \theta$). Then by use of the mean value theorem, one obtains (see "Appendix A.3"),

$$\ddot{T}_b = \frac{1}{C_{v_b}} (C_{p_u} * T_u + |(\Delta h_f)| - \frac{\ddot{P} * \ddot{V}_b}{\ddot{M}_b}) \quad (2.12)$$

where the "double dots" represent the moment right after the spark goes on, $\theta_s + \epsilon$. \ddot{T}_b is the initial value of the temperature in the burned zone right after spark.

(2) Combustion development

After the ignition, useful combustion chamber design information can be generated with simple geometric models of the flame. Usually, the surface which defines the leading edge of the flame can be approximated by a portion of the surface of a sphere. Thus the mean burned gas front can also be approximated by a sphere. However, in this study we are mainly interested in overall performance and not in detailed combustion chamber design. Hence we will use a simplified model of the combustion chamber and flame front.

a) Engine combustion Chamber Design

There are a large number of options for the ICE chamber design which includes cylinder head and piston crown shape, spark plug location, size and number of valves, and intake port design. The design of these important parts of the ICE revolves around issues such as chamber compactness, surface/volume ratio, flame travel length, the fuel mixture motion and more important the burning velocity.

It is known, that the combustion chamber design which increases the burning velocity, favors the engine performance. When the fuel burning process takes place faster, occupies a shorter crank angle interval at a given engine speed, produces less heat transfer (due to lower burned gas temperatures) and increases efficiency.

Illustrations of each of the most commons examples ICE chamber shapes which produces a "fast burn" will be given next (Figure 5),

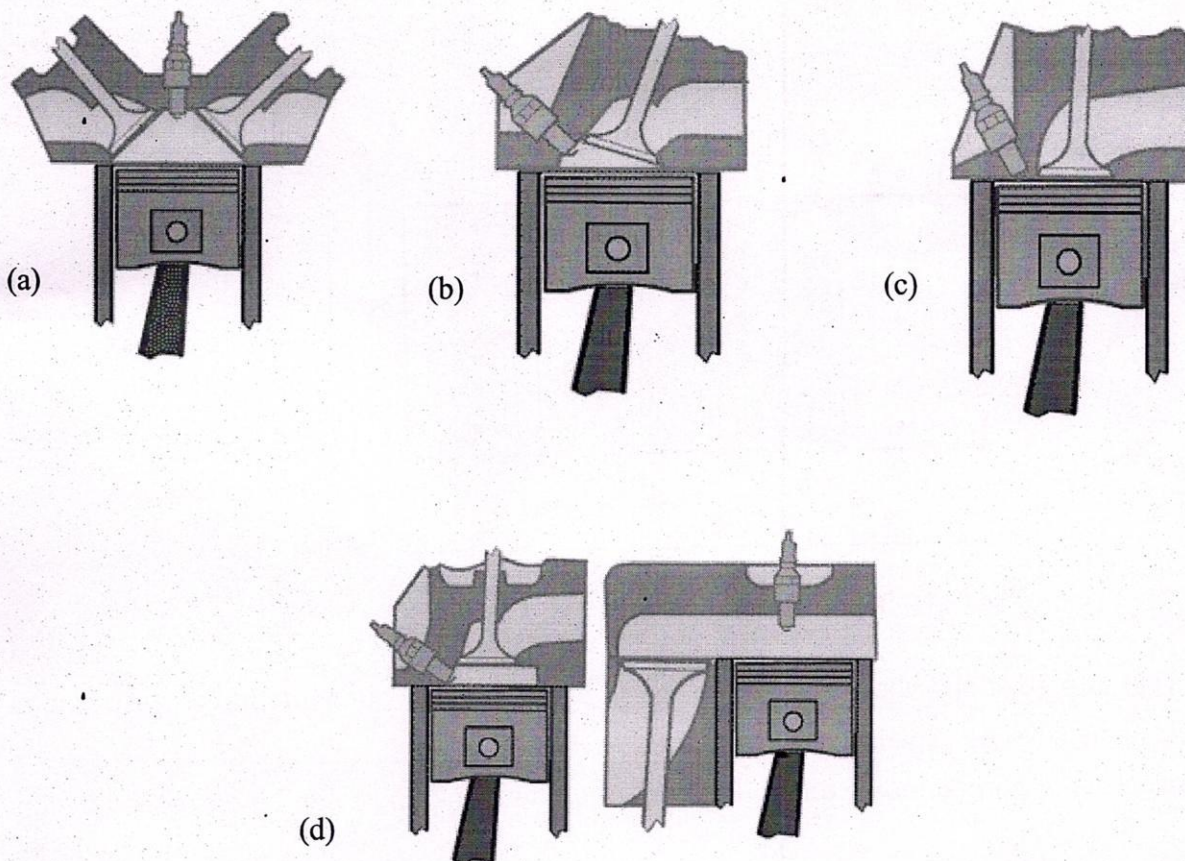


Figure 5 – Example of common internal combustion engine chambers: (a) hemispherical chamber; (b) wedge shaped chamber; (c) bathtub chamber; (d) bowl & piston with flathead on the right. [6]

In the scilab's program it was assumed that the combustion chamber was the simplest possible, so the piston is flat on top, the location of the spark plug is in the middle of the cylinder between the valves and the combustion chamber has a cylindrical geometry. Using this shape and knowing that the combustion reaction is so quick, it is possible to assume that the mean burned gas front can also be approximated by a cylinder instead by a sphere without committing significant errors as regards overall performance

b) Combustion chamber considerations

Assuming that the burned zone is a cylinder of height, h and radius, r at any instant of θ and using the model which consists in a two zone analysis of the combustion chamber which contains an unburned and burned gas region separated by a turbulent flame front (Figure 6),

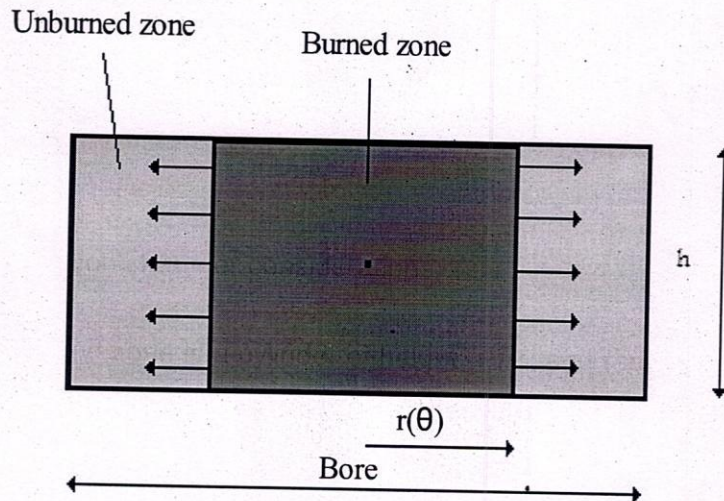


Figure 6 – Sketch of the front shape of the combustion chamber

It is possible to predict how the radius, r is going to change during the flame travel, assuming a linear distribution as it will be explained next.

The variation of the burned radius due to the change of the crank angle can be expressed by the approximation (Figure 7),

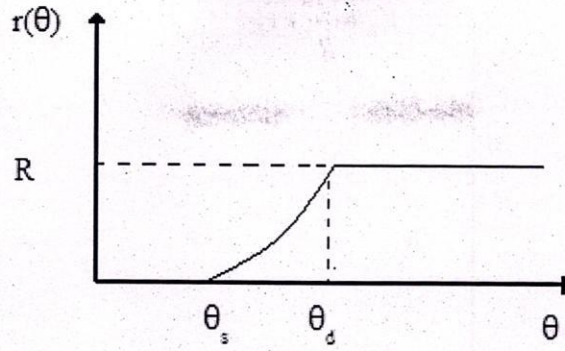


Figure 7 - Schematic of burned radius as a function of the crank angle

where $R = \frac{bo}{2}$, and θ_d is the angle θ at which the burned gas cylinder reaches the engine cylinder wall. θ_d is determined by the flame speed \hat{V}_f (in terms of angular rates),

Then,

$$\left\{ \begin{array}{l} r = \frac{dr}{d\theta} = 0 \text{ for } \theta < \theta_s \\ r = \hat{V}_f(\theta - \theta_s) \text{ and } \frac{dr}{d\theta} = \hat{V}_f \text{ for } \theta_s < \theta < \theta_d \\ r = R \text{ and } \frac{dr}{d\theta} = 0 \text{ for } \theta > \theta_d \end{array} \right. \quad (2.13)$$

The flame speed \hat{V}_f is given by the turbulence model. See "Appendix A.4" for r mathematical development.

Being $h = h(\theta)$ the height from piston, at any instant of θ , to the top of the cylinder, comes

$$h = \frac{v}{(\pi * R^2)} \quad (2.14)$$

where v is the total volume of the cylinder.

The burned and unburned volume and mass are easy to predict, now that r and h are known.

The burned volume is,

$$V_b = \pi * h * r^2 \quad (2.15)$$

so

$$V_b = \frac{1}{R^2} v^* r^2 \quad (2.16)$$

and its derivative

$$\frac{dV_b}{d\theta} = \frac{1}{R^2} r^2 \frac{dv}{d\theta} + \frac{2*v*r}{R^2} \frac{dr}{d\theta} \quad (2.17)$$

the unburned volume comes,

$$V_u = v - V_b \quad (2.18)$$

Finally, assuming that m_b is proportional to V_b , it is possible to say,

$$\frac{m_b}{m} = \frac{V_b}{v} \quad (2.19)$$

then, the burned mass is

$$m_b = \frac{m}{v} V_b \quad (2.20)$$

its derivative

$$\frac{dm_b}{d\theta} = m * \left(\frac{1}{v} \frac{dV_b}{d\theta} - \frac{V_b}{v^2} \frac{dv}{d\theta} \right) \quad (2.21)$$

and the unburned mass is

$$m_u = m - m_b \quad (2.22)$$

and its derivative

$$\frac{dm_u}{d\theta} = -\frac{dm_b}{d\theta} \quad (2.23)$$

c) Thermodynamic model

The energy balance on the unburned zone is

$$\frac{dU_u}{dt} = \dot{Q}_u - P \cdot \frac{dV_u}{dt} - \dot{M}_b * h_u \quad (2.24)$$

where $\frac{dm_u}{dt} = \dot{M}_u = -\dot{M}_b$, \dot{M}_u is the mass transfer rate to the unburned zone and \dot{M}_b is the mass transfer rate to the burned zone; \dot{Q}_u is the heat transfer rate from the unburned zone to the walls; P is the pressure in the cylinder; V_u the unburned volume; and U_u is the total internal energy in the unburned zone.

After the algebraic manipulation explained in "Appendix A.5" this equation becomes,

$$M_u * C_{v_u} \frac{dT_u}{dt} = \dot{Q}_u - P \frac{dV_u}{dt} + \dot{M}_b * (C_{v_u} - C_{p_u}) * T_u \quad (2.25)$$

The energy balance for the burned zone is

$$\frac{dU_b}{dt} = \dot{Q}_b - P \frac{dV_b}{dt} + \dot{M}_b * h_u \quad (2.26)$$

where $\frac{dm_b}{dt} = \dot{M}_b$; where \dot{Q}_b is the heat transfer rate from the burned zone to the walls; V_b and U_b are the volume and internal energy of the burned zone. The mass balances are

$$\dot{M}_b = \frac{dM_b}{dt} \quad (2.27)$$

and

$$\dot{M}_u = -\frac{dM_b}{dt} \quad (2.28)$$

where M_b and M_u are the zone masses.

After the algebraic manipulation explained in "Appendix A.6" equation (2.26) becomes,,

$$M_b * C_{v_b} \frac{dT_b}{dt} = \dot{Q}_b - P \frac{dV_b}{dt} + \dot{M}_b * (C_{p_u} * T_u - C_{v_b} * T_b + hf_u - hf_b) \quad (2.29)$$

(3) Combustion termination

The end of the combustion occurs when the flame front reaches the walls of the cylinder,

$$r=R$$

If the end of the combustion process is progressively delayed by retarding the spark timing, or decreasing the flame speed by decreasing the piston speed, the peak cylinder pressure occurs later in the expansion stroke and is reduced in magnitude. These change reduce the expansion stroke work transfer from the cylinder gases to the piston.

2.3.2 Turbulence characteristics

The flow processes in the engine cylinder are turbulent. In turbulent flows, the rates of transfer and mixing are several times greater than the rates due to molecular diffusion. This turbulent flow is produced by the high shear flow set up during the intake process and modified during compression. It leads to increased rates of heat, mass transfer and flame propagation, so, is essential to the satisfactory operation of the ICE.

This section explains the structure of the turbulent engine flame in the combustion process, as it develops from the spark discharge and the speed at which it propagates across the combustion chamber.

The turbulence model is based on equations that describe the evolution of turbulence in a fluid where the density is a function only of time. For details see [2].

The turbulence kinetic energy per unit mass, k is described by,

$$\frac{dk}{dt} = P - D + \frac{1}{M_c} \left(I - E - k \frac{dM_c}{dt} \right) \quad (2.30)$$

During the burn stage the turbulence energies of the burned and unburned gases are tracked together using the above equation with just the P and D terms,

$$\frac{dk}{dt} = P - D \quad (2.31)$$

because during the combustion there is no mass transfer, so $\frac{dM_c}{dt} = 0$ and I and E are

functions of $\frac{dM_c}{dt}$.

The term P is the turbulence energy production rate per unit mass and is modeled by,

$$P = F_p \frac{A_w}{v} |(U)| - \frac{2}{3} k \frac{1}{v} \frac{dv}{dt} \quad (2.32)$$

where the first term accounts for turbulence production due to the strain in the shear flow on the walls and the second the effects of compression.

D is the turbulence energy dissipation rate per unit mass and is modeled by,

$$D = F_d \frac{kV_t}{v^{(1/3)}} \quad (2.33)$$

The coefficients F_p and F_d are parameters that can be set by reference to experiments. To get the respective values for each term, the Engine Simulation Program (ESP) which is available in the Internet at the site "<http://esp.stanford.edu/>" was referenced.

Using the turbulence equation and the next two equations, it is possible to estimate V_f which is the velocity of the flame front relative to the unburned gases,

$$k = \frac{V_t^2}{2} \quad (2.34)$$

and

$$V_f = V_L + C_f * V_t \quad (2.35)$$

where V_t is the turbulence velocity in the unburned zone, V_L is the specified laminar flame speed and C_f is a specified coefficient, approximately unity. The value for V_L was obtained from the ESP.

After having the V_f estimated, it is possible to make a realistic approximation to the real variation of the burned radius. Knowing that,

$$\frac{dr}{dt} = V_f \quad (2.36)$$

and $dt = \frac{1}{w} d\theta$ comes,

$$\frac{dr}{d\theta} = \frac{V_f}{w} \quad (2.37)$$

Finally, the real variation of the burned radius due to the change of the crank angle when $\theta_s < \theta < \theta_d$ can be expressed by the next equation,

$$r = \frac{V_f}{w} (\theta - \theta_s) \quad (2.38)$$

and

$$\frac{dr}{d\theta} = \frac{V_f}{w} \quad \text{for } \theta_s < \theta < \theta_d \quad (2.39)$$

2.3.3 Heat Transfer

The heat transfer model used in this report is the same as before.

However, since we are using the two zone combustion model, during the development of the combustion, different quantities of heat are released from the burned and unburned zone.

Applying the Newtonian convection equation,

$$\hat{Q} = h_g(\theta) A_w(\theta) (T(\theta) - T_w) / N \quad (2.40)$$

The only thing that will change in this two zone combustion model are the exposed burned and unburned areas as shown in figure 8.

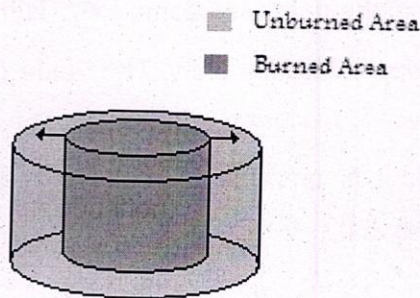


Figure 8 – Areas evolution during the combustion

The exposed burned area varies in function of the burned radius,

$$A_{wb} = \pi * r^2 \quad (2.41)$$

the exposed unburned area comes,

$$A_{wu} = A_w - A_{wb} \quad (2.42)$$

Finally, the heat transfer equation for the the burned zone is,

$$\hat{Q}_{wb} = -hg * A_{wb} * (T_b - T_w) / N \quad (2.43)$$

where,

$$h_g = 300 * P^{0.75} * U^{0.75} * bo^{-0.25} * T_b^{-0.465} / 1000 \quad (2.44)$$

and the heat transfer equation for the unburned zone is,

$$\hat{Q}_{wu} = -hg * A_{wu} * (T_u - T_w) / N \quad (2.45)$$

where,

$$h_g = 300 * P^{0.75} * U^{0.75} * bo^{-0.25} * T_u^{-0.465} / 1000 \quad (2.46)$$

where the units of h_g , P , U , b and T are $\text{kW/m}^2\text{K}$, kPa , m/s , m and K , respectively. The piston speed, U is given by equations (2.9) and (2.10).

2.4. Completely burned gas expansion

The burning stage lasts until the piston has progressed beyond TC, but ends before the piston has descended to the point of $\theta = \theta_{evo}$ near BC. The expansion from the end of combustion to θ_{evo} involves completely burned gas.

2.4.1 Thermodynamic equation of burned gas expansion

During this stage, the energy balance on the in-cylinder gas is,

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} \quad (2.47)$$

As both valves are closed there is no mass exchange so

$$\frac{dm}{d\theta} = \hat{m}_i = \hat{m}_e = 0 \quad (2.48)$$

After the algebraic manipulation shown in "Appendix A.7" comes,

$$\frac{dT}{d\theta} = \frac{\hat{Q}}{\hat{m} * \Delta C_v} - R_b \frac{T}{\Delta C_v V} \frac{dV}{d\theta} \quad (2.49)$$

2.4.2 Heat Transfer

At this stage, the heat release rate at any crank angle θ to the exposed cylinder wall at an engine speed N is determined with a Newtonian convection equation,

$$\hat{Q} = h_g(\theta) A_w(\theta) (T(\theta) - T_w) / N \quad (2.50)$$

The heat transfer coefficient $h_g(\theta)$ is the instantaneous averaged heat transfer coefficient and A_w is the exposed cylinder area.

The instantaneous heat transfer coefficient during the expansion stage is estimated in the same way as the heat transfer coefficient in the compression stage.

$$h_g = 300 * P^{0.75} * U^{0.75} * b_o^{-0.25} * T_b^{-0.465} / 1000 \quad (2.51)$$

where the units of h_g , P , U , b and T are $\text{kW/m}^2\text{K}$, kPa , m/s , m and K , respectively. The piston speed, U is given by equations (2.9) and (2.10).

Chapter – Gas exchange cycle

This cycle deals with the fundamentals of the gas exchange process, intake and exhaust and the valves mechanism in a four stroke internal combustion engine, called the gas exchange cycle . Only a brief explanation about the thermodynamics state and gas flow rate will be given.

This cycle is called the gas exchange cycle because it is where the burned gases from the expansion stroke are expelled (exhaust stroke, $\theta = \theta_{evo} \approx 2\pi$ to $\theta = 3\pi$) and fresh fuel-air is ingested (ingested stroke, θ_{ivo} to $\theta = 4\pi$).

This cycle is also described by the main two laws of thermodynamics governing the unburned and burned fuel-air mixture in the cylinder.

The gas exchange process is based on a "perfect valve timing". In perfect valve timing the exhaust valve closes exactly when the cylinder pressure drops below exhaust system pressure, but after TC at $\theta = 3\pi$. Then both valves remain closed while the piston descends (expanding the cylinder volume) until the cylinder pressure reaches the intake system pressure, which is less than the exhaust pressure for unsupercharged engines.

3.1. Valve action

Valves allow the gas exchange to occur. Valve opening and closing control is called valve timing. The valves action occurs for the following values of θ ,

- exhaust valve opens (evo) at $\theta = \theta_{evo} = 2\pi - \pi/9$;
- exhaust valve closes (evc) at $\theta = \theta_{evc} = 3\pi$;
- intake valve opens (ivo) at $\theta = \theta_{ivo}$ determined by intake pressure;
- intake valve closes (ivc) at $\theta = \theta_{ivc} = 4\pi$.

3.1.1 Geometry

The valves are driven by a cam that rotates at half the speed of the crankshaft.

The valve flow area depends on valve lift and the geometric details of the valve head, seat and stem.

$$\text{Curtain area} = \pi * D * l \quad (3.1)$$

and

$$\text{Port area} = \frac{1}{4} \pi (D^2 - d^2) \quad (3.2)$$

where $l = l(\theta)$ is the valve lift, D the port diameter and d the stem diameter.

It can be written mathematically as,

$$A_v = \min(\pi D l(\theta), \frac{1}{4} \pi (D^2 - d^2)) \quad (3.3)$$

The valve lift, $l(\theta)$ is usually a curve that looks like figure 9,

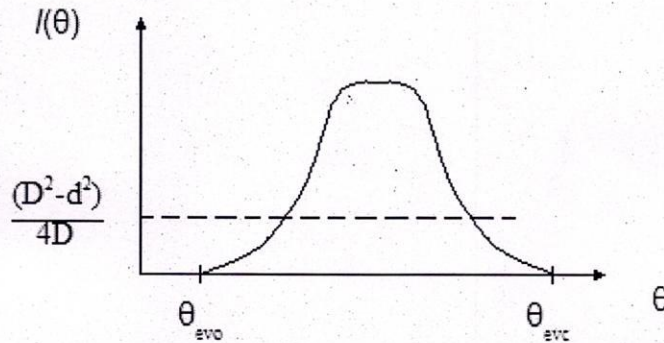


Figure 9 - Typical exhaust valve timing diagram

Approximating this curve by the formula,

$$l(\theta) = \frac{l_m}{2} (1 - \cos(\theta)) \quad (3.4)$$

where l_m is the maximum valve lift and usually

$$l_m > \frac{(D^2 - d^2)}{4D} \quad (3.5)$$

3.1.2 Isentropic flow through an orifice

From the gas dynamics, the isentropic flow through an orifice is given by,

$$\dot{m} = \rho_o * C_f * A_v * c_o * \left(\frac{2}{k-1} \left(\left(\frac{P_v}{P_o} \right)^{\frac{2}{k}} - \left(\frac{P_v}{P_o} \right)^{\frac{k+1}{k}} \right) \right)^{\frac{1}{2}} \quad (3.6)$$

as long as,

$$\frac{P_o}{P_v} < \left(\frac{P_o}{P_{v_{cr}}} \right) = \left(\frac{k+1}{2} \right)^{\frac{k}{k+1}} \quad (3.7)$$

if

$$\frac{P_o}{P_v} \geq \left(\frac{P_o}{P_{v_{cr}}} \right) \quad (3.8)$$

then

$$\dot{m} = \dot{m}_{cr} = \rho_o * C_f * A_v * c_o * \left(\frac{2}{k+1} \right)^{\frac{k+1}{2(k+1)}} \quad (3.9)$$

When

$$\frac{P_o}{P_v} \leq \left(\frac{P_o}{P_{v_{cr}}} \right)$$

the flow is "choked". This means that the flow right at the orifice (the valve passage) is "sonic". At this condition only \dot{m}_{cr} (=constant) can pass through the valve passage regardless of how small we

make $\frac{P_v}{P_o}$ (see figure 10). Note that P_o is the upstream pressure (stagnation pressure) and

P_v is the downstream (static pressure – just "behind" valve).

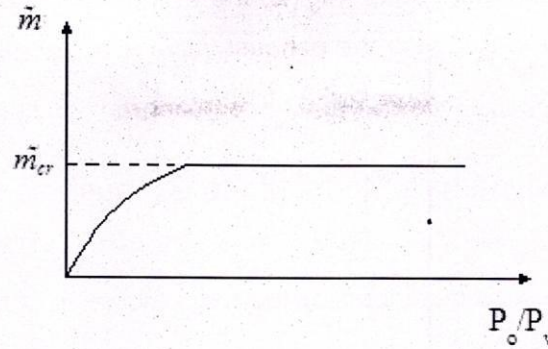


Figure 10 – Isentropic flow through an orifice

For an ideal gas,

$$\rho_o = \frac{P_o}{R * T_o} \quad (3.10)$$

and c_o is the speed of sound,

$$c_o = \sqrt{kRT_o} \quad (3.11)$$

Note that the speed of sound goes up with the upstream temperature. And note that the upstream for exhaust is the cylinder temperature and the upstream for intake is the manifold temperature which is much less than cylinder temperature. That is why exhaust valve is smaller than intake valve.

The non-ideal effects are accounted by introducing a flow coefficient C_f , which is here assumed to be $C_f \approx 0.40$.

3.2. Exhaust stage

The configuration of the exhaust system plays an important role for a good engine performance.

The exhaust system typically consists of an exhaust manifold, exhaust pipe, often a catalytic converter for emission control, and a muffler or silencer. My study concerns only with the exhaust stroke which proceeds the power stroke cycle.

At the end of the power stroke, the piston is located at the bottom center, $\theta=2\pi$. Heat that is left over from the power stroke is now transferred to the water in the water jacket until the pressure approaches atmospheric pressure. The exhaust valve is then opened by the cam on the rocker arm to begin the exhaust stroke.

The purpose of the exhaust stroke is to clear the cylinder of the spent exhaust in preparation for

another ignition cycle. As the exhaust stroke begins, the cylinder and combustion chamber are full of exhaust products at low pressure. Because the exhaust valve is open, the exhaust gas is pushed past the valve and exits the engine.

3.2.1 Thermodynamics Model of the Exhaust stage

As it was said, this stroke is also described by the main two laws of thermodynamics. The energy balance for the exhaust stroke is,

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} - \hat{H}_e \quad (3.12)$$

As the intake valve is closed there is no mass exchange through it so $\hat{m}_i = \hat{H}_i = 0$ and as the exhaust valve is opened comes $\frac{dm}{d\theta} = -\hat{m}_e$.

After the algebraic manipulation shown in "Appendix A.8" comes,

$$\frac{dT}{(d\theta)} = \frac{\hat{Q}}{(m * C_{vb})} - \frac{R_b * T}{(V * C_{vb})} \frac{dV}{(d\theta)} - \frac{\hat{m}_e R_b * T}{m C_{vb}} \quad (3.13)$$

3.2.2 Heat transfer

At this stage, the heat release rate at any crank angle θ to the exposed cylinder wall at an engine speed N is determined with a Newtonian convection equation,

$$\hat{Q} = h_g(\theta) A_w(\theta) (T(\theta) - T_w) / N \quad (3.14)$$

The heat transfer coefficient $h_g(\theta)$ is the instantaneous averaged heat transfer coefficient and is the exposed cylinder area.

The instantaneous heat transfer coefficient during the expansion stage is estimated in the same way as the heat transfer coefficient in the compression stage.

$$h_g = 300 * P^{0.75} * U^{0.75} * b_o^{-0.25} * T_b^{-0.465} / 1000 \quad (3.15)$$

where the units of h_g , P , U , b and T are kW/m²K, kPa, m/s, m and K, respectively. The piston speed, U is given by equations (2.9) and (2.10).

2.3. Intake stage

The exchange cycle ends with the intake stroke as the piston is pulled towards the crankshaft (to the bottom center position, BC) as the intake valves opens and fuel (unburned gas) and air are drawn past it and into the combustion chamber and cylinder from the intake manifold located on top of the combustion chamber. The exhaust valve is closed and the electrical contact switch is open. The fuel/air mixture is at a relatively low pressure (near atmospheric). At the end of the intake stroke, the piston is located at the bottom center position, ready to begin the power cycle.

Intake manifolds consisting of plenums (separated spaces containing air at a pressure greater than atmospheric pressure) and pipes are usually required to deliver the inlet air charge from some preparation device such as an air cleaner or compressor.

3.3.1 Thermodynamics Model of the Intake stage

Using the main two laws of thermodynamics, the energy balance for the intake stroke is,

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} + \hat{H}_i \quad (3.18)$$

As the exhaust valve is closed there is no mass exchange through it so $\hat{m}_e = \hat{H}_e = 0$ and as the intake valve is opened comes $\frac{dm}{d\theta} = \hat{m}_i$.

After the algebraic manipulation shown in "Appendix A.9" comes,

$$\frac{dT}{d\theta} = \frac{\hat{Q}}{\hat{m} * C_{vu}} - \frac{R_u * T}{V * C_{vu}} \frac{dV}{d\theta} + \frac{\hat{m}_i}{\hat{m} * C_{vu}} (C_{pu} * T_u - C_{vu} * T) \quad (3.19)$$

3.3.2 Heat transfer

The heat transfer model for the intake is the same as it was explained for the exhaust process.

Note that in the intake process the heat transfer can be neglected, when comparing it to the heat released in the exhaust and combustion stages.

As it was said, the computer simulation was implemented by way of "Scilab" computer program.

"Scilab" is a scientific software for numerical computations, and it is currently used in educational and industrial environments around the world. This program can be found freely in the following website: <http://www.scilab.org/>

The ICE computer simulation developed is called CycleComQC (see appendix C.1) and the original one developed by CVK is called CycleCom (see appendix C.2).

CycleComQC Inputs

The values of the input parameters used in this simulation can be found in the tables below.

Engine data:

Connecting rod/stroke length ratio	Rc	2	/
Compression ratio	rc	11	/
Maximum cylinder volume	Vm	0.00055	m ³
Ignition onset	ths	0.88 π	rad
Burn duration	thb	0.33 π	rad
Burning end	thd	1.21 π	rad
Engine rpm	rpm	6000	rpm
Cylinder wall temperature	Tw	400	K

Exhaust valve data:

Exhaust valve opens	thevo	2 π - π /9	rad
Exhaust valve closes	thevc	3 π	rad
Exhaust port diameter	eportd	0.040	m
Exhaust valve stem diameter	estem	0.015	m
Exhaust valve max lift	elm	0.035	m

Intake valve data:

Intake valve opens	thivo	3π	rad
Intake valve closes	thivc	4π	rad
Intake port diameter	ipord	0.040	m
Intake valve stem diameter	istem	0.015	m
Intake valve max lift	ilm	0.035	m

Intake-exhaust state

Intake pressure	Pi	100	K
Exhaust pressure	Pe	150	K
Intake temperature	Ti	320	K

Unburned fuel-air equation of state, CH₄:

Uf-air gas constant	Ru	0.2968	kJ/kgK
Cp for uf-air	Cpu	1.022	kJ/kgK
Specific heat ration for uf-air	ku	1.409	/
Zero degree enthalpy of uf-air	hfu	-692.0	kJ/kg
Cv for uf-air	Cvu	0.725	kJ/kgK

Burned fuel-air equation of state, CH₄:

Bf-air gas constant	Rb	0.2959	kJ/kgK
Cp for bf-air	Cpb	1.096	kJ/kgK
Specific heat ration for bf-air	kb	1.370	/
Zero degree enthalpy of bf-air	hfb	-3471.0	kJ/kg
Cv for bf-air	Cvb	0.800	kJ/kgK

Turbulence coefficients:

Parameters set by reference to experiments	F_d	0.03	/
Parameters set by reference to experiments	F_p	0.05	/
Coefficient set by reference to experiments	C_f	0.4	/
Laminar flame speed	V_L	0.04	m/s

Results/Discussion

The single zone with homogeneously mixed burned and unburned gases developed by CVK is called the single zone adiabatic model. I have added heat transfer to this model and it will be called the single zone heat transfer model so first the results obtained from the single zone heat transfer model simulation program (see appendix C.3.) are discussed. After that, it is discussed the main simulation program and how the power, efficiency and heat transfer vary with the engine speed and how this have influence on the engine performance. Another point explained is the turbulence model and how it affects the combustion stage.

As illustration an engine with a compression ratio of 11, total volume of 55 cm^3 , using methane (CH_4) which properties are very similar to gasoline and setting the engine to run at 6000rpm was obtained the following results for the heat transfer model,

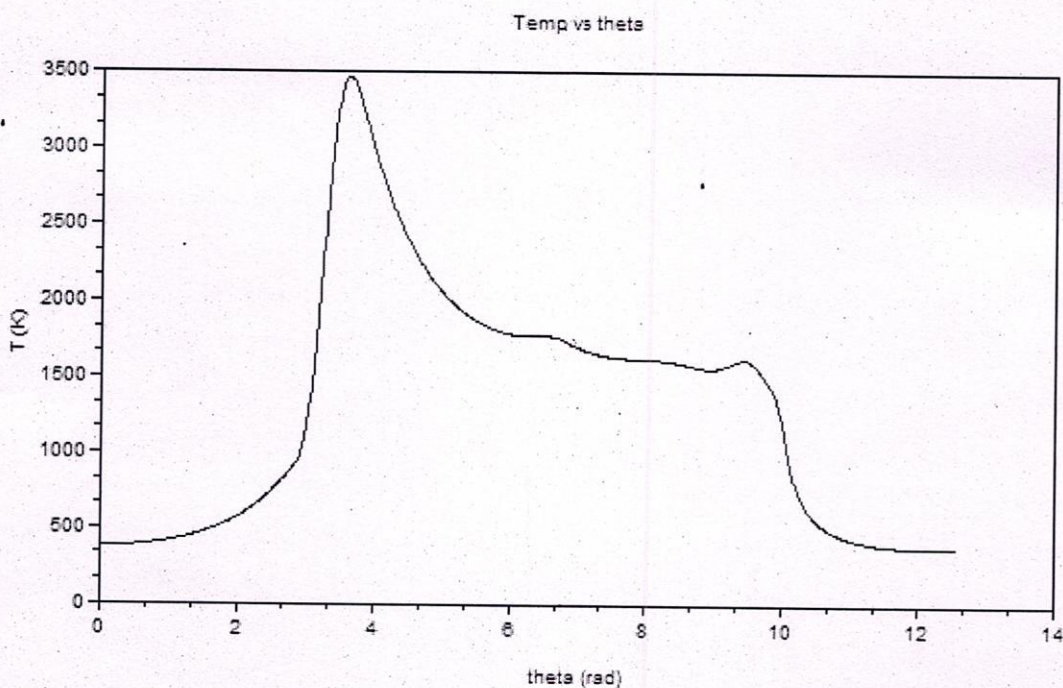


Figure 11 - Temperature vs crank angle diagram for a four stroke engine running at 6000rpm

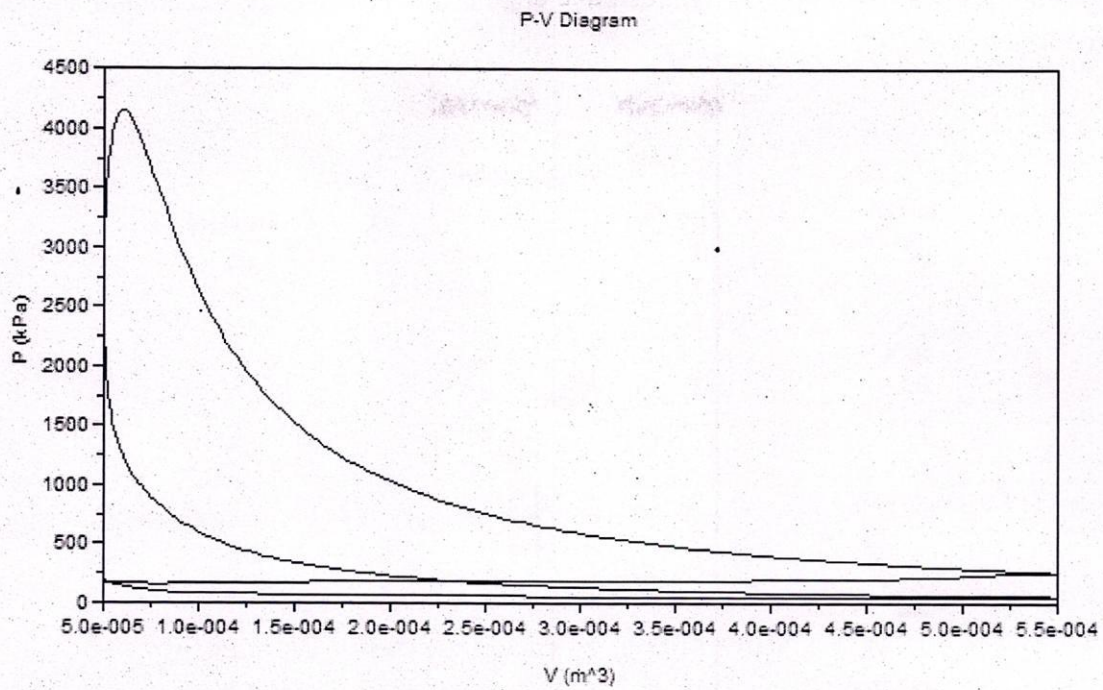


Figure 12 - P-V diagram for a four stroke engine running at 6000rpm

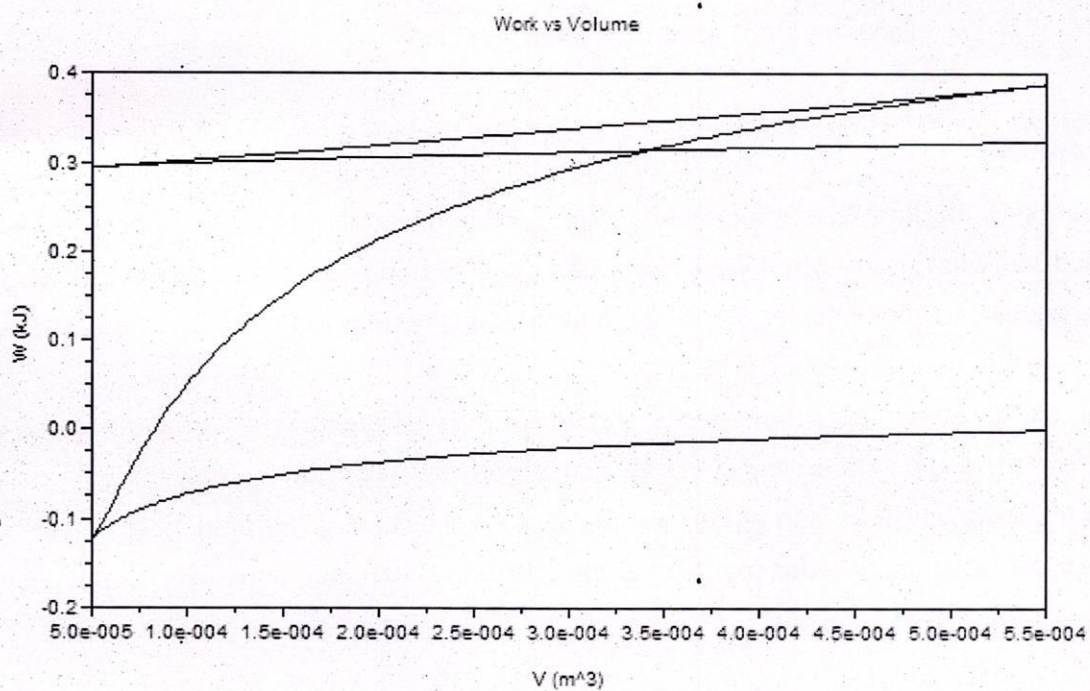


Figure 13 - Work vs Volume diagram for a four stroke engine running at 6000rpm

In order to compare the performance of an adiabatic engine with a non-adiabatic engine, the following results were obtained,

For an adiabatic engine			
RPM	Power (kW)	Efficiency (%)	Work (kJ)
3000	14.81	0.49	0.59
4000	16.32	0.47	0.49
5000	16.60	0.46	0.40
6000	16.22	0.43	0.32
7000	15.57	0.41	0.27
8000	14.84	0.39	0.22

Figure 14 – Variation of the power, efficiency and work with the engine speed (rpm) for an adiabatic engine

For a non-adiabatic engine			
RPM	Power (kW)	Efficiency (%)	Work (kJ)
3000	12.40	0.40	0.50
4000	13.79	0.38	0.41
5000	14.05	0.37	0.34
6000	13.66	0.35	0.27
7000	13.00	0.32	0.22
8000	13.58	0.34	0.20

Figure 15 – Variation of the power, efficiency and work with the engine speed (rpm) for a non-adiabatic engine

As we can see, the heat loss affects the engine performance. As we were expecting, the power, efficiency and the work drops when we have a non-adiabatic engine. In a non-adiabatic engine the temperature and pressure is smaller than in an adiabatic engine due to the greater heat losses which represents work that cannot be done.

Note that as the engine speed increases, the work drops off because the amount of gas burned goes down. At high engine speed the burn gas exchange is restricted by the flow through the valves. In particular the exhaust gases can not be completely expelled. Hence the amount of intake gases is reduced. Also the work required for pumping of the intake and exhaust, especially exhaust is greatly increased.

The results from the last version of the CycleComQC are the followings,

RPM	Power (kW)	Efficiency (%)	Qt (kW)	Work (kJ)	Qw (kW/rad)
2000	8.90	40.0%	-10.60	0.534	-0.636
3000	13.52	43.1%	-14.98	0.541	-0.599
4000	17.89	47.2%	-19.03	0.537	-0.571
5000	21.85	52.1%	-22.79	0.524	-0.547
6000	25.24	57.4%	-26.21	0.505	-0.524
7000	27.55	62.8%	-28.91	0.472	-0.496
8000	27.93	67.5%	-30.05	0.419	-0.451
9000	26.71	69.7%	-29.75	0.356	-0.397
10000	24.38	68.8%	-28.43	0.293	-0.341
11000	21.38	65.1%	-26.52	0.233	-0.289
12000	18.04	59.1%	-24.29	0.180	-0.243

Figure 16 – Variation of the power, efficiency, work and heat loss with the engine speed (rpm) for a non-adiabatic engine

Comparing this values for the final program with the ones obtained in heat transfer model, we conclude that the values for the efficiency are to high. For 6000rpm, the efficiency result obtained for this final simulation doing $Q=0$ is 77%. This cannot be correct because it exceeds Carnot efficiency which is only approximately 60%. This confirms that something is wrong with the basic two zone model we have developed. However, we will show the results obtained by this model.

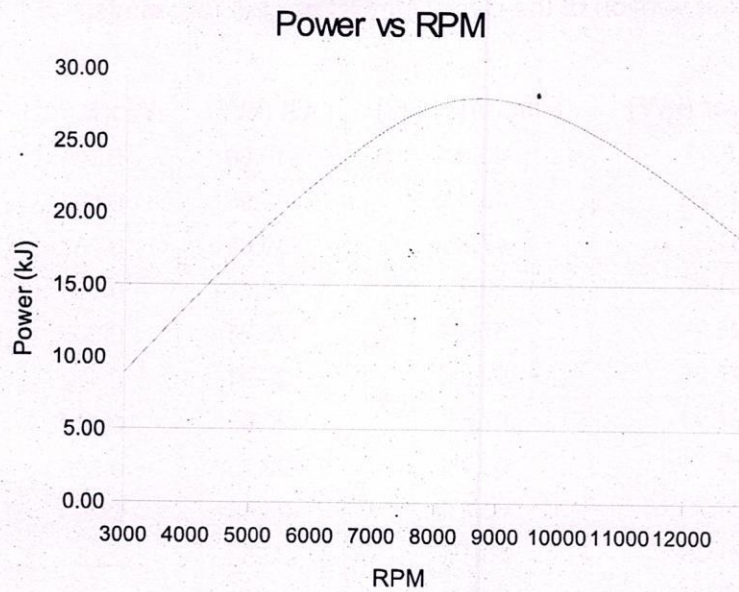


Figure 17 – Power as a function of engine speed (rpm)

As expected, the power rises and falls with rpm, but the peak occurs for 8500rpm which is much too high. Also the peak power is much too large.

When we have heat loss in an engine, occurs a reduction in its temperature and pressure which represents work that cannot be done leading to lower values of power.

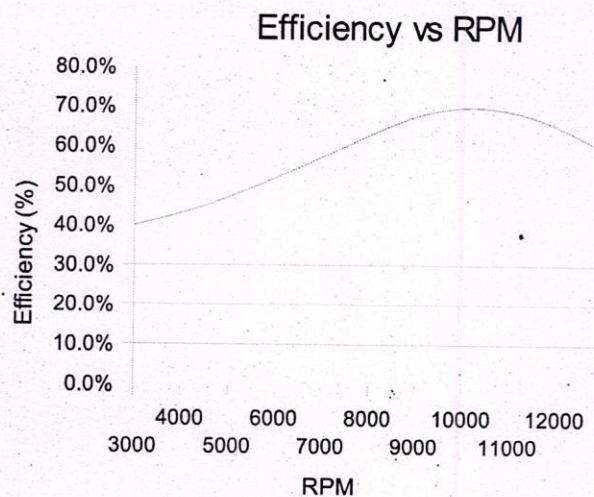


Figure 18 – Efficiency as a function of engine speed (rpm)

Analysing the figure 18 and the efficiency values obtained for an non-adiabatic engine (figure 16), we see that the combustion model is not correct in the simulation program. In fact, as expected the efficiency drops for a non-adiabatic engine. However, and as it was said, for an adiabatic engine running at 6000rpm the efficiency is 77% which is impossible to occur because even the Carnot Cycle does not have so high efficiency.

After some experiences with the program, CVK and I conclude that something is wrong with the pressure calculation which is leading to high values of work and efficiency.

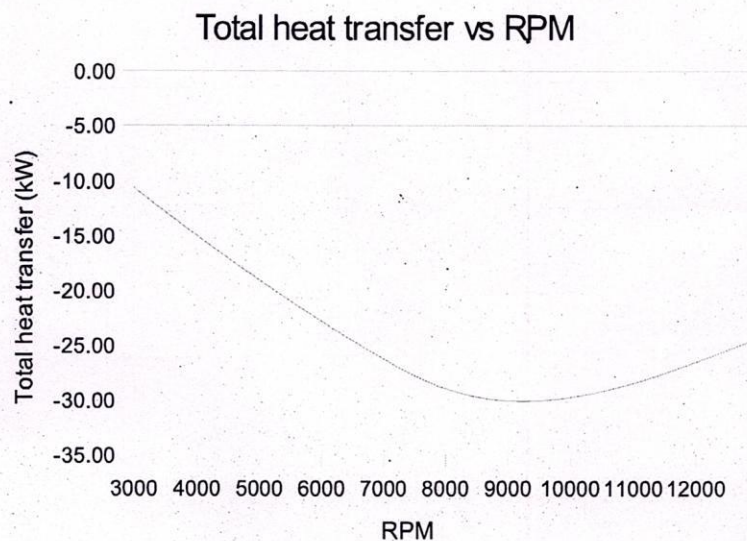


Figure 19 – Total heat transfer as a function of engine speed (rpm)

From figure 19 we can say that the peak of total heat transfer occur for 9000rpm. As the engine speed is increasing the heat transfer also increase because the temperature and pressure in the chamber are also increasing. However, after 9000rpm, the heat transfer start to decrease due to the decrease of the temperature in the chamber because the amount of fresh gas decreases due to restricted gas exchange process as explained before.

Note as the engine speed increases, the heat loss per cycle and the work drops off because the amount of gas burned goes down. As it was said, at high engine speed the gas exchange process is more and more restricted.

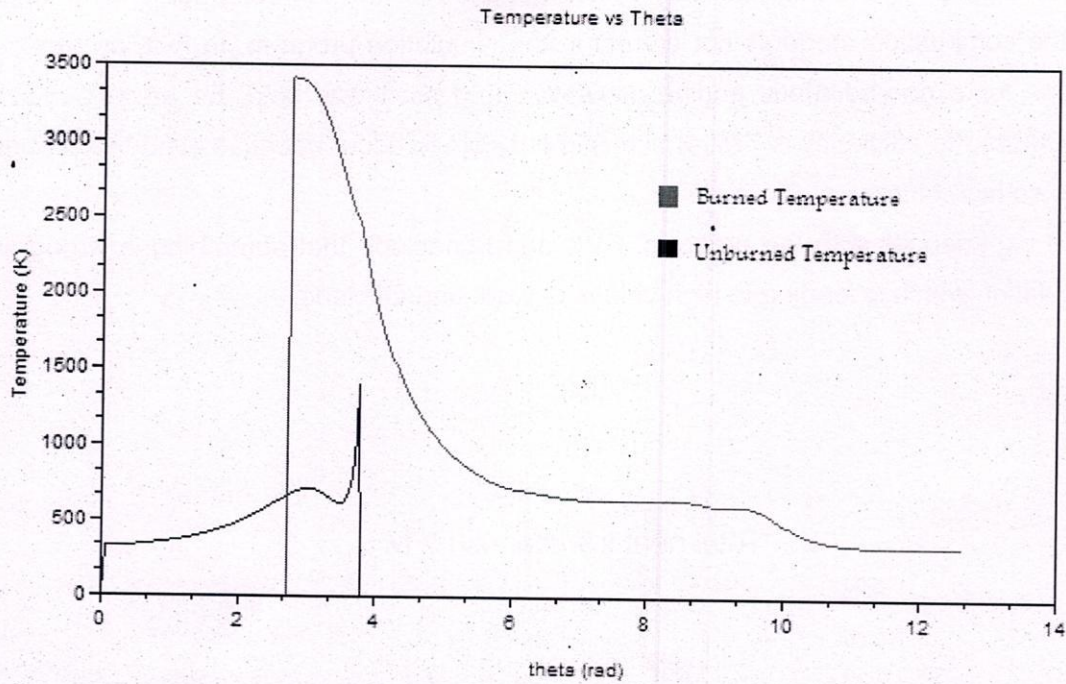


Figure 20 - Temperature vs crank angle diagram for a four stroke engine running at 6000rpm

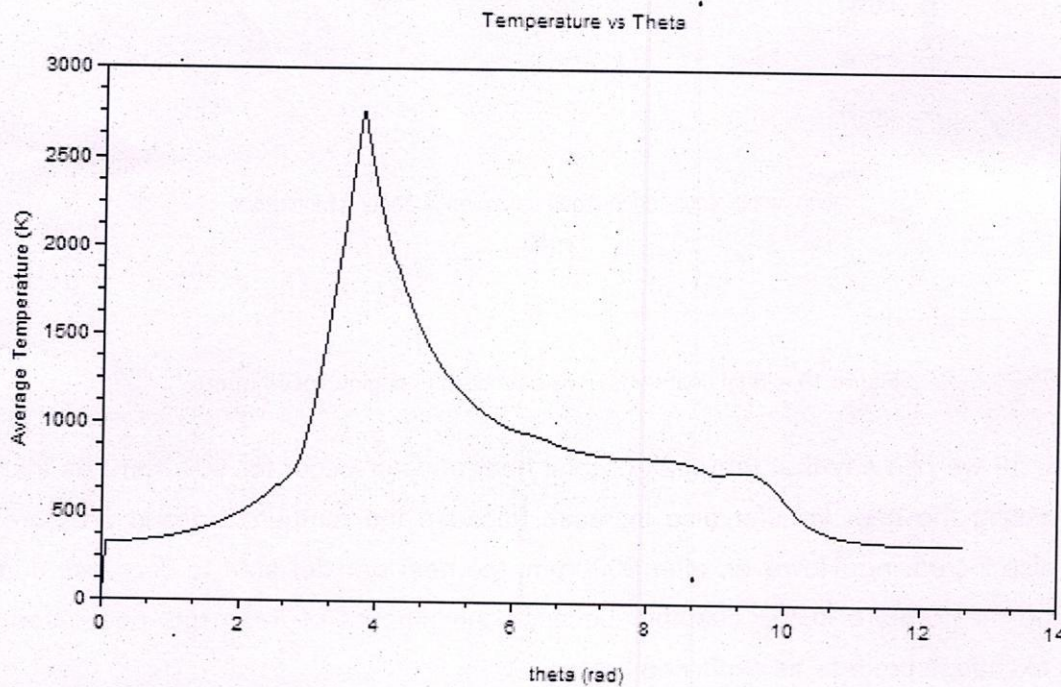


Figure 21 - Average temperature vs crank angle diagram for a four stroke engine running at 6000rpm

As it was expected, the temperature peak occurs for the combustion stage, when the spark goes off. Looking to the average temperature curve, the temperature peak occurs for 4 rad, but this is due

to the fact that in the unburned curve there is a small glitch when the unburned mass goes to zero (the temperature increase which in a real engine is wrong). This problem is occurring due to some restrictions that were implemented for the combustion stage.

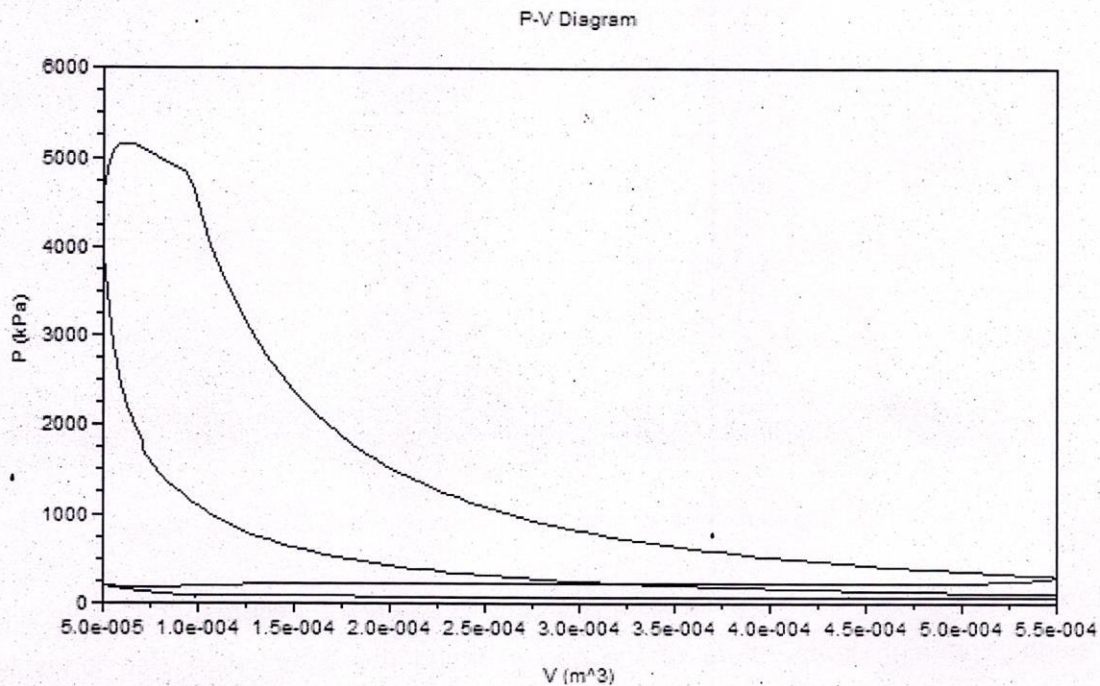


Figure 22 – P-V diagram for a four stroke engine running at 6000rpm

The pressure increases with the movement of the piston from the BC position to the TC position. The peak pressure occurs when the piston is in the TC position. In this position, the spark already went off and we are in the middle of the combustion stage. The pressure starts to drop when the piston start the descending movement to the BC position, where the exhaust valve will open. Note that with the increase of the piston speed, the peak pressure gets smaller.

We can see that the pressure peak here is considerably higher (5200kPa) than the 4200kPa of the single zone model with heat transfer (figure 12). The single zone model is much more realistic for typical engines. This shows that the approximation we have made for the pressure calculation is not accurate.

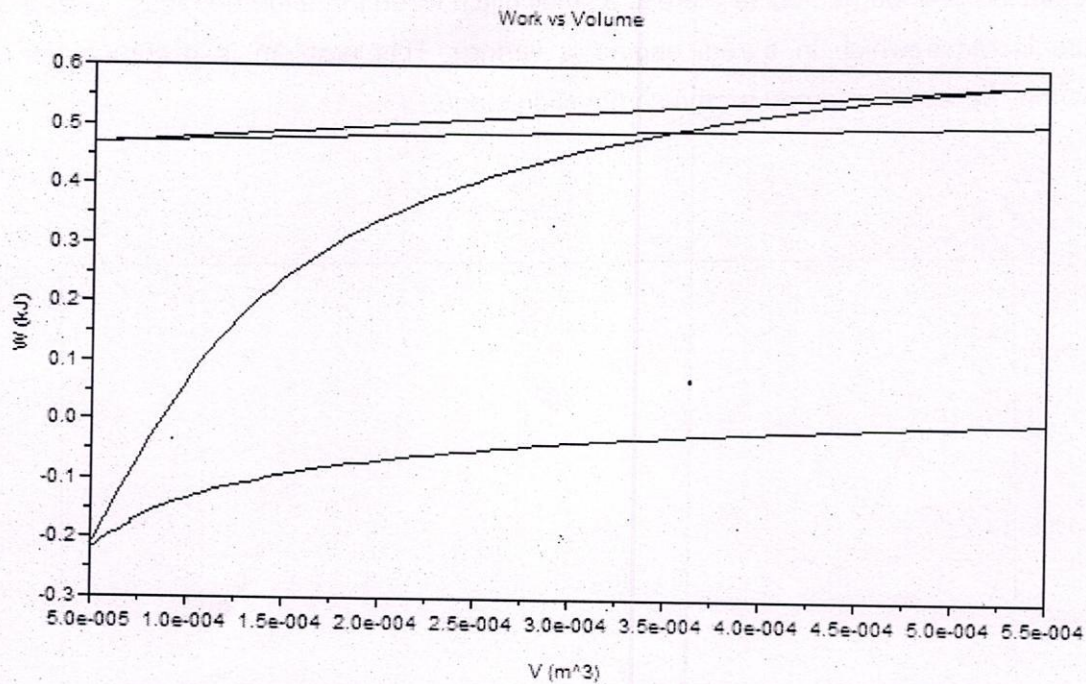


Figure 23 - Work vs crank angle diagram for a four stroke engine running at 6000rpm

As we can see from figure 23, the main work produced in an engine occurs in the combustion and expansion stage. The high temperature of the gases during the combustion also leads to high pressure. This high pressure of the gases will act on the face of the piston causing it to move from T.C (minimum volume) to the B.C (maximum volume) in which produces the main work. In the intake stage, the work is also positive, however it can be neglected.

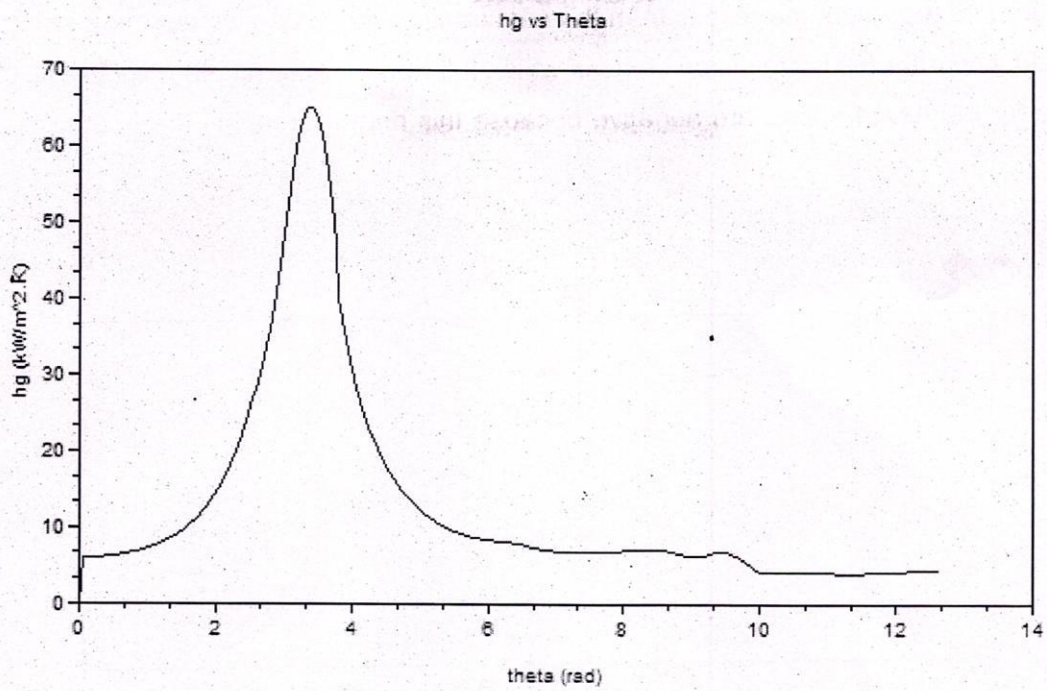


Figure 24 - Heat transfer coefficient vs crank angle diagram for a four stroke engine running at 6000rpm

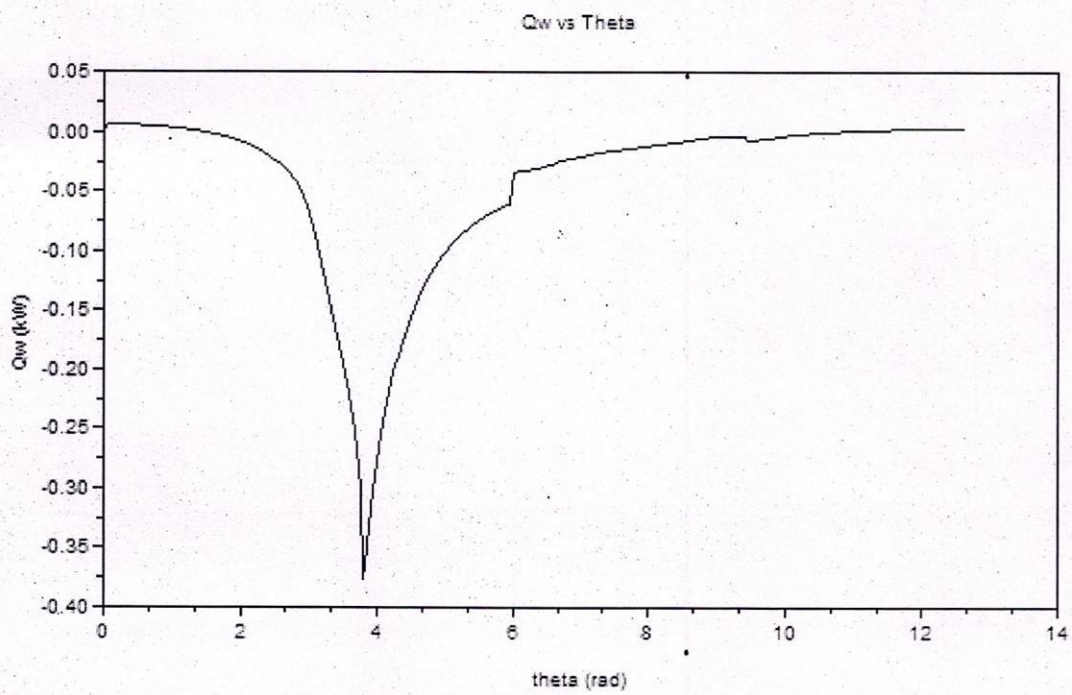


Figure 25 – Heat transfer vs crank angle diagram for a four stroke engine running at 6000rpm

Figures 24 and 25 represent the results from the heat transfer model. As we were expecting in both curves the peaks occur for the combustion stage in which the temperature hits the higher values. Note that the values for the heat transfer are negative because this heat represents the heat losses in the engine.

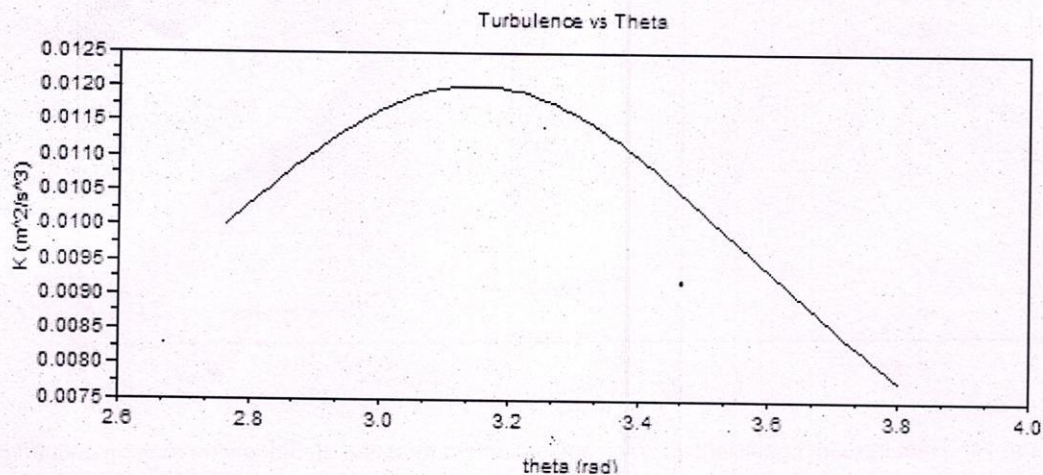


Figure 26 – Turbulence vs crank angle diagram for a four stroke engine running at 6000rpm

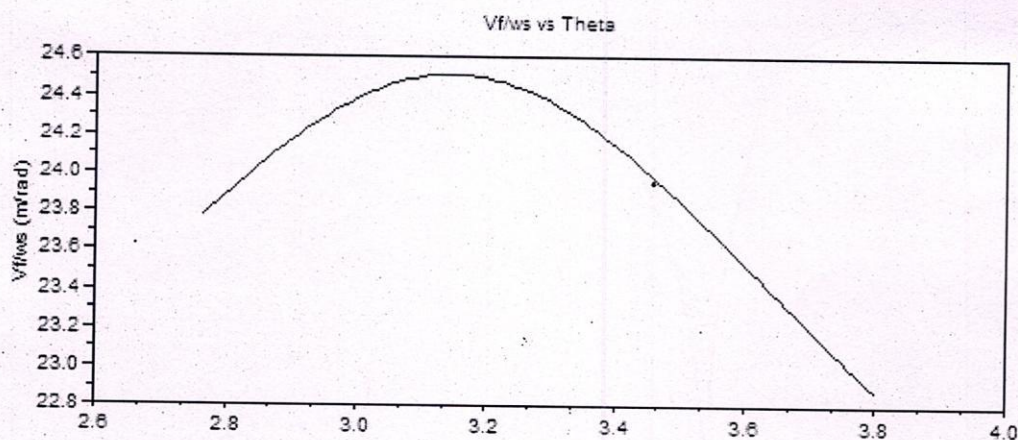


Figure 27 – Flame speed vs crank angle diagram for a four stroke engine running at 6000rpm

Figures 26 and 27 represent the results from the combustion model in which is taken into account the turbulence in the cylinder.

The main objective of this model was to watch the variation of burn duration with the engine speed. In real engines, the mixture burning and the flame speed are strongly influenced by engine speed.

When the engine speed increases, the flame speed also increases. However, the burning rate throughout the combustion process increases, though not quite, as rapidly as engine speed leading to higher crank angle intervals.

In the CycleComQC, the objective was to implement one turbulence model that represent this variation, however the results were not quite the ones expected. As we can see from the figures 26 and 27 the variation of the turbulence is proportional to the piston speed. From this we can conclude, that the turbulence model used were not the most accurate. The increase in the engine speed leads to a proportional increase in the flame speed and the combustion duration stays constant. The net effect is not much different from CvK's essentially constant burning rate. The turbulence model used was approximated from Lumley's model. This should be corrected in a future effort.

Concluding remarks

The fundamentals principles which govern internal combustion engine design and operations were well developed and implemented using the "Scilab" computer program. All the objectives proposed were achieved. For the heat transfer model the results obtained were quite good. However, the results obtained from the last simulation where it was added the heat transfer model plus the combustion model were not the ones expected. As mentioned previously, the model used was oversimplified, thus leading to quantitatively erroneous results although the results were qualitatively correct.

We can conclude that treating the combustion model CVK developed is simpler than the one developed during this project and leads to the good results. However, the CycleComQC is more realistic because it presents a non-adiabatic engine, and the two zone model inside the combustion chamber which takes into account the turbulence inside it. However, some of the details need to be improved in order to give quantitatively more accurate results.

We have concluded that the main difficulty is the pressure approximation. It is felt that with a little more time the pressure calculation can be improved by a new model that we have developed.

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Appendix A

Mathematical and thermodynamic manipulations

A.1. Equation 1.1

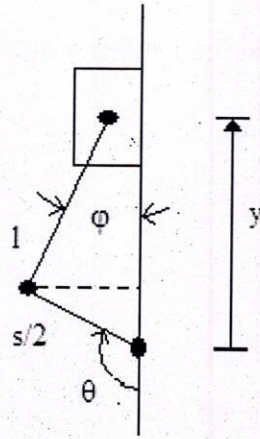


Figure 23 - Sketch of the slider crank model of piston-cylinder geometry

When

$$\left\{ \begin{array}{l} \theta=0: y(\theta)=l-\frac{s}{2} \\ \theta=\pi: y(\pi)=l+\frac{s}{2} \end{array} \right.$$

$$y(\theta)=l*\cos(\varphi)-\frac{s}{2}\cos(\varphi)$$

$$l*\sin(\varphi)-\frac{s}{2}\sin(\varphi) = \sin(\varphi)=\frac{s}{2l}\sin(\varphi)$$

and

$$\cos(\varphi)=\sqrt{(1-\sin^2(\varphi))}$$

comes,

$$\cos(\varphi)=\sqrt{(1-(\frac{s}{2l})^2 \sin^2(\varphi))}$$

so

$$y(\theta)=l\sqrt{(1-(\frac{s}{2l})^2 \sin^2(\theta))}-\frac{s}{2}\cos(\theta)$$

Finally,

$$V(\theta) = \frac{\pi}{4} b o^2 (y(\theta) - l + \frac{s}{2})$$

A.2. Equation 2.3

Energy Balance for the compression stage:

$$\frac{dU}{\theta} = \hat{Q} - \hat{W}$$

$$\frac{dm}{\theta} = \hat{H}_i = \hat{H}_e = 0$$

- Total internal energy (U):

$$U = m * [x * (Cv_b * T + hf_b) + (1-x) * (Cv_u * T + hf_u)]$$

and its derivative

$$\begin{aligned} \frac{dU}{d\theta} &= m * \left[\frac{dx}{d\theta} (Cv_b * T + hf_b) + x * (Cv_b \frac{dT}{d\theta}) - \frac{dx}{d\theta} (Cv_u * T + hf_u) + (1-x) * (Cv_u \frac{dT}{d\theta}) \right] \\ &= \frac{dU}{d\theta} = m * \left[((Cv_b - Cv_u) * T + hf_b - hf_u) \frac{dx}{d\theta} + (x * (Cv_b - Cv_u) + Cv_u) \frac{dT}{d\theta} \right] \end{aligned}$$

where $\Delta hf = hf_b - hf_u$ and $\Delta Cv = Cv_b - Cv_u$

- Power output to the piston (W):

$$\delta W = PdV$$

so

$$\delta \hat{W} = P \frac{dV}{d\theta}$$

assuming ideal gas behavior,

$$PV = mRT \text{ then,}$$

$$P = \frac{mRT}{V} \quad \text{where} \quad R = xR_b + (1-x)R_u$$

Finally,

$$\delta \hat{W} = m(xR_b + (1-x)R_u) \frac{T}{V} \frac{dV}{d\theta}$$

- Energy balance equation:

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} \quad \text{comes}$$

$$m*(x*\Delta C_v + C_{v_u}) \frac{dT}{(d\theta)} = \hat{Q} - m(xR_b + (1-x)R_u) \frac{T}{V} \frac{dV}{(d\theta)} - m*(\Delta C_v * T + (hf_b - hf_u) \frac{dx}{(d\theta)})$$

$$\text{where} \quad A = x*\Delta C_v + C_{v_u}$$

Finally,

$$\frac{dT}{d\theta} = \frac{\hat{Q}}{m*A} - (xR_b + (1-x)R_u) \frac{T}{AV} \frac{dV}{d\theta} - \frac{\Delta C_v T + \Delta hf}{A} \frac{dx}{d\theta}$$

A.3. Equation 2.12

Calculation of the spark ignition equation using the "Mean Value theorem of Calculus":

Using the following general equation which it will be explained in this appendix,

$$M_b * C_{v_b} \frac{dT_b}{dt} = \dot{Q}_b - P \frac{dV_b}{dt} + \dot{M}_b * (C_{p_u} * T_u - C_{v_b} * T_b + hf_u - hf_b)$$

and applying the "Mean Value theorem of calculus" comes,

$$\int_{\epsilon} M_b * C_{v_b} \frac{dT_b}{d\theta} d\theta = \int_{\epsilon} \dot{Q}_b d\theta - \int_{\epsilon} P \frac{dV_b}{d\theta} d\theta + \int_{\epsilon} \dot{M}_b * (C_{p_u} * T_u - C_{v_b} * T_b + hf_u - hf_b) d\theta$$

where ϵ represents $[\theta, \theta + \epsilon]$ and

$$\int_{\epsilon} \dot{Q}_b d\theta = 0 ;$$

$$\int_{\epsilon} P \frac{dV_b}{d\theta} d\theta = P(\theta_s)(V_b(\theta_s + \epsilon) - V_b(\theta_s)) = P(\theta_s)V_b(\theta_s + \epsilon)$$

$$\begin{aligned} \int_{\epsilon} \frac{dM_b}{d\theta} (Cp_u * T_u - Cv_b * T_b + hf_u - hf_b) d\theta = \\ (Cp_u * T_u(\theta_s) - Cv_b * T_b(\theta_s))(M_b(\theta_s + \epsilon) - M_b(\theta_s)) + (hf_u - hf_b)(M_b(\theta_s + \epsilon) - M_b(\theta_s)) \end{aligned}$$

Knowing that

$$\begin{aligned} M_b(\theta_s) &= 0 \\ \ddot{M}_b &= M_b(\theta_s + \epsilon) > 0 , \end{aligned}$$

$$\int_{\epsilon} M_b * Cv_b \frac{dT_b}{d\theta} d\theta = M_b(\epsilon) * Cv_b (T_b(\theta_s + \epsilon) - T_b(\theta_s))$$

Replacing the integrals in equation comes,

$$\begin{aligned} M_b(\epsilon) * Cv_b (T_b(\theta_s + \epsilon) - T_b(\theta_s)) = -P(\theta_s)V_b(\theta_s + \epsilon) + \\ (Cp_u * T_u(\theta_s) - Cv_b * T_b(\theta_s))(M_b(\theta_s + \epsilon) - M_b(\theta_s)) + (hf_u - hf_b)(M_b(\theta_s + \epsilon) - M_b(\theta_s)) \end{aligned}$$

and knowing that

$$T_b(\theta_s) = T_u , \quad T_b(\theta_s + \epsilon) = \ddot{T}_b \quad \text{and} \quad V_b(\theta_s + \epsilon) = \ddot{V}_b$$

$$Cv_b(\ddot{T}_b - T_u) = (Cp_u * T_u(\theta_s) - Cv_b * T_u) + (hf_u - hf_b) - P(\theta_s) * \frac{\ddot{V}_b}{\ddot{M}_b}$$

$$Cv_b * \ddot{T}_b = Cv_b * T_u + Cp_u * T_u - Cv_b * T_u + (hf_u - hf_b) - P(\theta_s) \frac{\ddot{V}_b}{\ddot{M}_b}$$

Finally,

$$\ddot{T}_b = \frac{1}{C_{vb}} (C_{pu} * T_u + |(hf_u - hf_b)| - \frac{\ddot{P} * \ddot{V}_b}{\ddot{M}_b})$$

A.4. Equation 2.13

Calculation of the burned radius, r assuming a "linear distribution",

$$r(\theta) = a\theta + b$$

Boundary conditions:

i. $r(\theta_s) = 0$

ii. $r(\theta_d) = R$ where $R = \frac{b\phi}{2}$

it comes,

i. $r(\theta_s) = 0 \Rightarrow 0 = a\theta_s + b \Rightarrow b = -a\theta_s$

ii. $r(\theta_d) = R \Rightarrow R = a\theta_d + b \Rightarrow R = a\theta_d - a\theta_s \Rightarrow a = \frac{R}{\theta_d - \theta_s}$

Replacing a and b in equation comes,

$$r(\theta) = \frac{R}{\theta_d - \theta_s} \theta - \frac{R}{\theta_d - \theta_s} \theta_s$$

$$r(\theta) = \frac{R}{\theta_d - \theta_s} (\theta - \theta_s)$$

A.5. Equation 2.25

The energy balance on the unburned zone is described by

$$\frac{dU_u}{dt} = \dot{Q}_u - P \frac{dV_u}{dt} - \dot{M}_b * h_u$$

applying to it some thermodynamic concepts and algebraic manipulation comes:

- Unburned Internal Energy and its derivative

$$U_u = M_u * (C_{v_u} * T_u + hf_u)$$

and its derivative is

$$\frac{dU_u}{dt} = \frac{dM_u}{dt} (C_{v_u} * T_u + hf_u) + M_u * C_{v_u} \frac{dT_u}{dt}$$

- Expressing the unburned enthalpy as

$$h_u = C_{p_u} * T_u + hf_u$$

Replacing in the main energy balance equation comes

$$M_u * C_{v_u} \frac{dT_u}{dt} = \dot{Q}_u - \frac{P \cdot dV_u}{dt} + \dot{M}_b * (C_{p_u} * T_u + hf_u) + \dot{M}_b * (C_{v_u} * T_u + hf_u)$$

and finally

$$M_u * C_{v_u} \frac{dT_u}{dt} = \dot{Q}_u - P \frac{dV_u}{dt} + \dot{M}_b * (C_{v_u} - C_{p_u}) * T_u$$

A.6. Equation 2.29

The energy balance on the burned zone is described by

$$\frac{dU_b}{dt} = \dot{Q}_b - P \frac{dV_b}{dt} + \dot{M}_b * h_u$$

applying to it some thermodynamic concepts and algebraic manipulation comes:

- Burned internal energy and its derivative

$$U_b = M_b * (C_{v_b} * T_b + hf_b)$$

and its derivative is

$$\frac{dU_b}{dt} = \frac{dM_b}{dt} (C_{v_b} * T_b + hf_b) + M_b * C_{v_b} \frac{dT_b}{dt}$$

- Expressing the burned enthalpy as

$$h_b = h_u$$

Substituting in the main energy balance equation on the burned zone comes

$$M_b * C_{v_b} \frac{dT_b}{dt} = \dot{Q}_b - \frac{P \cdot dV_b}{dt} + \dot{M}_b * (C_{p_u} * T_u + hf_u) - \dot{M}_b * (C_{v_b} * T_b + hf_b)$$

and finally

$$M_b * C_{v_b} \frac{dT_b}{dt} = \dot{Q}_b - P \frac{dV_b}{dt} + \dot{M}_b * (C_{p_u} * T_u - C_{v_b} * T_b + hf_u - hf_b)$$

A.7. Equation 2.49

Energy Balance for completely burned gas expansion:

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W}$$

$$\frac{dm}{d\theta} = \hat{H}_i = \hat{H}_e = 0$$

- Total internal energy (U):

$$U = \hat{m} * [x * (C_{v_b} * T + hf_b) + (1-x) * (C_{v_u} * T + hf_u)]$$

and its derivative

$$\begin{aligned} \frac{dU}{d\theta} &= \hat{m} * \left[\frac{dx}{d\theta} (C_{v_b} * T + hf_b) + x * (C_{v_b} \frac{dT}{d\theta}) - \frac{dx}{d\theta} (C_{v_u} * T + hf_u) + (1-x) * (C_{v_u} \frac{dT}{d\theta}) \right] \\ &= \frac{dU}{d\theta} = \hat{m} * \left[((C_{v_b} - C_{v_u}) * T + hf_b - hf_u) \frac{dx}{d\theta} + (x * (C_{v_b} - C_{v_u}) + C_{v_u}) \frac{dT}{d\theta} \right] \end{aligned}$$

where $\Delta hf = hf_b - hf_u$ and $\Delta C_v = C_{v_b} - C_{v_u}$

- Power output to the piston (W):

$$\delta W = P dV$$

so

$$\delta \hat{W} = P \frac{dV}{d\theta}$$

assuming ideal gas behavior,

$$PV = mRT \text{ then,}$$

$$P = \frac{mRT}{V} \text{ where } R = xR_b + (1-x)R_u$$

Finally,

$$\delta \hat{W} = \hat{m} (xR_b + (1-x)R_u) \frac{T}{V} \frac{dV}{d\theta}$$

- Energy balance equation:

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} \text{ comes}$$

$$\hat{m} * (x * \Delta C_v + C_{v_u}) \frac{dT}{d\theta} = \hat{Q} - \hat{m} (xR_b + (1-x)R_u) \frac{T}{V} \frac{dV}{d\theta} - \hat{m} * (\Delta C_v * T + (hf_b - hf_u) \frac{dx}{d\theta})$$

$$\text{where } A = x * \Delta C_v + C_{v_u}$$

Finally and since $x=1$,

$$\frac{dT}{d\theta} = \frac{\hat{Q}}{\hat{m} * \Delta C_v} - R_b \frac{T}{\Delta C_v V} \frac{dV}{d\theta}$$

A.8. Equation 3.13

Energy balance for the exhaust stage:

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} - \hat{H}_e$$

$$\hat{m}_i = \hat{H}_i = 0 \text{ and } \frac{dm}{d\theta} = -\hat{m}_e$$

- Total internal energy (U):

$$u = C_{vb} * T + h_{fb}$$

and

$$\frac{dU}{dt} = \frac{dm * u}{dt} = m \frac{du}{dt} + u \frac{dm}{dt}$$

- Power output to the piston (W):

$$\delta W = PdV \text{ so } \delta \hat{W} = P \frac{dV}{d\theta}$$

assuming ideal gas behavior,

$$PV = mRT \text{ then,}$$

$$P = \frac{mRT}{V} \text{ where } R = xR_b + (1-x)R_u$$

As long as we only have burned gas comes,

$$\delta \hat{W} = \frac{\hat{m} * R_b * T}{V} \frac{dV}{d\theta}$$

Energy balance equation comes,

$$\hat{m} \frac{du}{d\theta} - u * \hat{m}_b = \hat{Q} - \hat{W} - \hat{m}_b * (Cp_b * T + hf_b) =$$

$$\hat{m} \frac{du}{d\theta} - (Cv_b * T + hf_b) * \hat{m}_b = \hat{Q} - \hat{W} - \hat{m}_b * (Cp_b * T + hf_b) =$$

$$\hat{m} \frac{du}{d\theta} = \hat{Q} - \hat{W} - \hat{m}_b (Cp_b * T + hf_b - Cv_b * T - hf_b) = \hat{m} \frac{du}{d\theta} = \hat{Q} - \hat{W} - \hat{m}_b (Cp_b - Cv_b) T =$$

$$\hat{m} \frac{du}{d\theta} = \hat{Q} - \hat{W} - \hat{m}_b * R_b * T$$

Finally,

$$\hat{m} \frac{du}{d\theta} = \frac{-\hat{m} * d(Cv_b * T + hf_b)}{d\theta} = \hat{m} * Cv_b \frac{dT}{d\theta}$$

so

$$\frac{dT}{d\theta} = \frac{\hat{Q}}{\hat{m} * Cv_b} - \frac{R_b * T}{V * Cv_b} \frac{dV}{d\theta} - \frac{\hat{m}_b R_b * T}{\hat{m} Cv_b}$$

A.9. Equation 3.19

Energy balance for the intake stage :

$$\frac{dU}{d\theta} = \hat{Q} - \hat{W} + \hat{H}_i$$

$$\hat{m}_e = \hat{H}_e = 0 \quad \text{and} \quad \frac{dm}{(d\theta)} = \hat{m}_i$$

- Total internal energy (U):

$$u = C_{v_u} * T + hf_u$$

and

$$\frac{dU}{dt} = \frac{dm * u}{dt} = m \frac{du}{dt} + u \frac{dm}{dt}$$

- Power output to the piston (W):

$$\delta W = PdV \quad \text{so} \quad \delta \hat{W} = P \frac{dV}{d\theta}$$

assuming ideal gas behavior,

$$PV = mRT \quad \text{then,}$$

$$P = \frac{mRT}{V} \quad \text{where} \quad R = xR_b + (1-x)R_u$$

As long as we only have unburned gas comes,

$$\delta \hat{W} = \frac{\hat{m} * R_u * T}{V} \frac{dV}{d\theta}$$

Energy balance equation comes,

$$\hat{m} \frac{du}{d\theta} + u * \hat{m}_u = \hat{Q} - \hat{W} + \hat{m}_u * (C_{p_u} * T_u + hf_u) =$$

$$\hat{m} \frac{du}{d\theta} - (C_{v_u} * T + hf_u) * \hat{m}_u = \hat{Q} - \hat{W} + \hat{m}_b * (C_{p_u} * T_u + hf_u) =$$

$$\hat{m} \frac{du}{d\theta} = \hat{Q} - \hat{W} + \hat{m}_u (C_{p_u} * T_u + hf_u - C_{v_u} * T - hf_u) = \hat{m} \frac{du}{d\theta} = \hat{Q} - \hat{W} + \hat{m}_u (C_{p_u} * T_u - C_{v_u} * T)$$

Finally,

$$\hat{m} \frac{du}{d\theta} = \frac{-\hat{m} * d(C_{v_u} * T + hf_u)}{d\theta} = \hat{m} * C_{v_u} \frac{dT}{d\theta}$$

so

$$\frac{dT}{d\theta} = \frac{\hat{Q}}{\hat{m} * C_{v_u}} - \frac{R_u * T}{V * C_{v_u}} \frac{dV}{d\theta} + \frac{\hat{m}_u}{\hat{m} * C_{v_u}} (C_{p_u} * T_u - C_{v_u} * T)$$



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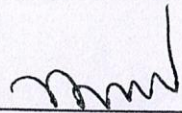
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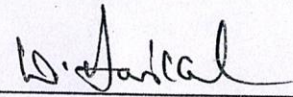
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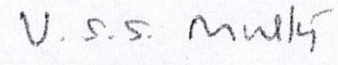
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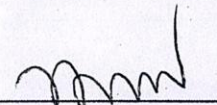
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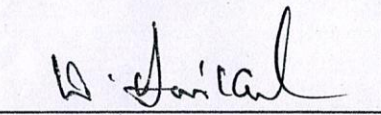
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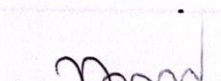
Approved by AICTE, New Delhi & Affiliated to JNTUA, Ananthapuramu.

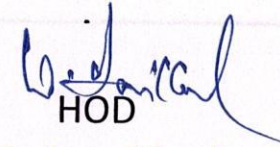
Department of Mechanical Engineering
Certification Course on Simulation of IC Engine Process

S.No	Name of the Student	Roll List	Is the Course content meet your expectation	Is the lecture sequence well planned	Is the level of course high	Is the course exposed you to the new knowledge and practices	Rate the Knowledge of the Speaker	Rate the value of Course in increasing your skills	Any Issues
1	AKULA SREEDHAR	199Y1A0301	yes	Agree	Strongly Ag	Yes	4	5	Nil
2	B SRINIDHI SAI	199Y1A0302	yes	Agree	Agree	Yes	5	5	Need extra Explanation
3	BANDI SHIVA REDDY	199Y1A0303	yes	Agree	Agree	Yes	4	5	Nil
4	CHAGANTI SUNIL KUMAR REDDY	199Y1A0304	yes	Agree	Agree	Yes	5	5	Nil
5	CHEPPALI AMATHYA	199Y1A0305	yes	Agree	Agree	Yes	5	5	Provide PPT
6	CHIRUCHAPALA ABDUL SUBAHAN	199Y1A0306	yes	Agree	Agree	Yes	5	4	Nil
7	DEVAPATLA BHARATH SIMHA REDDY	199Y1A0307	yes	Agree	Agree	Yes	5	4	Nil
8	DUDIMANI SAI SRUJAN KUMAR	199Y1A0308	yes	Agree	Agree	Yes	4	5	Nil
9	GADIKOTA MURALIDHAR REDDY	199Y1A0309	yes	Agree	Agree	Yes	5	5	Nil
10	GANGALA VENKATA PRATHAP	199Y1A0310	yes	Agree	Agree	Yes	5	5	Nil
11	GANUGAPENTA BHARATH	199Y1A0311	yes	Agree	Agree	Yes	5	5	Nil
12	GODDENDLA ASHOK KUMAR	199Y1A0312	yes	Agree	Agree	Yes	5	5	Nil
13	GUDURU SUBHAN	199Y1A0313	yes	Agree	Agree	Yes	5	5	Nil
14	KAPURAM VAMSINATH REDDY	199Y1A0315	yes	Agree	Agree	Yes	5	5	Nil
15	KETHIREDDY NAVEEN KUMAR REDDY	199Y1A0316	yes	Agree	Agree	Yes	5	5	Nil

16	KONANGI SUBBANNA	199Y1A0317	yes	Agree	Agree	Yes	5	5	Nil
17	KONERU SAI KARTHIK	199Y1A0318	yes	Agree	Agree	Yes	5	5	Nil
18	KOTHAPALLE VAMSIDHAR REDDY	199Y1A0319	yes	Agree	Agree	Yes	4	5	Nil
19	KUMMARI MANJUNATH	199Y1A0320	yes	Agree	Agree	Yes	5	5	Nil
20	KUMMETHA SAI KUMAR REDDY	199Y1A0321	yes	Agree	Agree	Yes	5	5	Nil
21	L M VINAY KUMAR	199Y1A0322	yes	Agree	Agree	Yes	5	5	Nil
22	MALEPATI SIVA SAI REDDY	199Y1A0324	yes	Agree	Agree	Yes	5	5	Nil
23	MANJUNATHA DINESH KUMAR	199Y1A0325	yes	Agree	Agree	Yes		5	Nil
24	MARKAPURAM MYSORA REDDY	199Y1A0326	yes	Agree	Agree	Yes	5	5	Nil
25	MEDIMALA KIRAN KUMAR	199Y1A0327	yes	Agree	Agree	Yes	5	5	Nil
26	MOGHAL JUNAID BAIG	199Y1A0328	yes	Agree	Agree	Yes	5	5	Nil
27	ACHUKATLA NUMAIR	209Y5A0301	yes	Agree	Agree	Yes	5	5	Nil
28	BHOJANAPALLE NAGA SIVA	209Y5A0302	yes	Agree	Agree	Yes		5	Nil
29	BIJJE PURUSHOTHAM	209Y5A0303	yes	Agree	Agree	Yes	5	5	Nil
30	BUCHUPALLI SIVA PRASAD REDDY	209Y5A0304	yes	Agree	Agree	Yes	5	5	Nil
31	CHINNI GURU PRASAD	209Y5A0305	yes	Agree	Agree	Yes	5	5	Nil
32	CHINTHAGINJALA VENKATA SUBBARAYUDU	209Y5A0306	yes	Agree	Agree	Yes	5	5	Nil
33	CHITRALA VENKATA SWAMY SETTY	209Y5A0307	yes	Agree	Agree	Yes	4	5	Nil
34	DAKALA SRINIVASULU	209Y5A0308	yes	Agree	Agree	Yes	5	5	Nil
35	DHARA SUNIL KUMAR	209Y5A0309	yes	Agree	Agree	Yes	5	5	Nil
36	ESLAVATH RAVI NAIK	209Y5A0310	yes	Agree	Agree	Yes	5	5	Nil
37	GORLA CHARAN KUMAR REDDY	209Y5A0311	yes	Agree	Agree	Yes	5	5	Nil
38	GUDISHA DILIP KUMAR	209Y5A0312	yes	Agree	Agree	Yes	4	5	Nil
39	GUTTURU GIRISHKUMAR REDDY	209Y5A0313	yes	Agree	Agree	Yes	5	5	Nil
40	JAMPANGI OBULESU	209Y5A0314	yes	Agree	Agree	Yes	5	5	Nil
41	JONNADULA SATISH	209Y5A0315	yes	Agree	Agree	Yes	5	5	Nil
42	KOTA UPENDRA REDDY	209Y5A0316	yes	Agree	Agree	Yes	5	5	Nil
43	KUNU SIVA BABJI	209Y5A0317	yes	Agree	Agree	Yes	4	5	Nil
44	KURUVA AJAY KUMAR	209Y5A0318	yes	Agree	Agree	Yes	5	5	Nil

45	LAKKINENI SUDHARSHAN	209Y5A0319	yes	Agree	Agree	Yes	5	5	Nil
46	LANKAMSETTY VENKATA LOKESH	209Y5A0320	yes	Agree	Agree	Yes	5	5	Nil
47	LINGAMBOTI BHUSHAN	209Y5A0321	yes	Agree	Agree	Yes	5	5	Nil
48	MEALLA GOWTHAMSAI	209Y5A0322	yes	Agree	Agree	Yes	4	5	Nil
48	MEDIREDDY BHARATH REDDY	209Y5A0323	yes	Agree	Agree	Yes	5	5	Nil
50	MOKA VEERABHADRA	209Y5A0324	yes	Agree	Agree	Yes	5	5	Nil
51	MOOLI CHENNAKESAVA REDDY	209Y5A0325	yes	Agree	Agree	Yes	5	5	Nil
52	NEELAM PAVAN KUMAR	209Y5A0326	yes	Agree	Agree	Yes	5	5	Nil
53	PATTU MONESH	209Y5A0327	yes	Agree	Agree	Yes	5	5	Nil
54	PINNAPURAM MADHU SUDHAN	209Y5A0330	yes	Agree	Agree	Yes	4	5	Nil
55	POREDDY HARI VARDHAN REDDY	209Y5A0331	yes	Agree	Agree	Yes	5	5	Nil
56	PRODDATURU NAZEER BASHA	209Y5A0332	yes	Agree	Agree	Yes	5	5	Nil
57	YATAGIRI HEMANTH KUMAR	209Y5A0343	yes	Agree	Agree	Yes	5	5	Nil
58	YEDDULADODDI ASHOK	209Y5A0344	yes	Agree	Agree	Yes	5	5	Nil


Coordinator


HOD

Professor & head
Department of Mechanical Engineering
K.S.R.M. College of Engineering
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