Ministry of Higher Education and Scientific Research - Iraq

Northern Technical University

College of Oil and Gas Techniques Engineering

Department of Fuel and Energy Engineering

MODULE DESCRIPTION FORM

نموذج وصف المادة الدراسية

Module Information معلومات المادة الدراسية				
Module Title	**Internal Combustion Engine**		**Module Delivery**	
Module Type	**core**		☒Theory ☒Lab ☐Tutorial ☐Practical ☒Seminar	
Module Code	**FEK304**			
ECTS Credits	**5**			
SWL (hr/sem)	**125**			
Module Level	Three		**Semester of Delivery**	1
Administering Department	FEK		**College**	Type College Code
Module Leader	Isam Ezzulddin Yousif		**e-mail**	isamizz@ntu.edu.iq
Module Leader's Acad. Title	Lecturer		**Module Leader's Qualification**	M.Sc. Eng.
Module Tutor	Name (if available)		**e-mail**	E-mail
Peer Reviewer Name	Name		**e-mail**	E-mail
Scientific Committee Approval Date	01/06/2023		**Version Number**	1.0

Relation with other Modules العلاقة مع المواد الدراسية الأخرى			
Prerequisite module	Thermodynamics	**Semester**	1
Co-requisites module	None	**Semester**	

Module Aims, Learning Outcomes and Indicative Contents أهداف المادة الدراسية ونتائج التعلم والمحتويات الإرشادية	
Module Aims أهداف المادة الدراسية	This course elaborates on the fundamentals of internal combustion engines and what affects their performance, operation, fuel requirements and environmental impact. The course considers thermodynamics, combustion, heat transfer and friction phenomena, and fuel properties, relevant to engine power, efficiency and emissions, and examines design features and operating characteristics of different types of internal combustion engines; including spark-ignition, diesel, stratified charge, and mixed-cycle engines.
Module Learning Outcomes مخرجات التعلم للمادة الدراسية	**Upon completion of this course, students will be able to:** 1. describe the basic operation of different IC engines including the method of combustion, engine speed control and all of the major components; 2. conduct performance analysis of IC engines based on engineering parameters such as mean-effective pressure and volumetric efficiency; 3. conduct complete thermodynamic analyses of IC engines including the effects of residual mass fraction, finite heat release, valve timing, and heat losses; 4. carry out calculations of combustion reactions for hydrocarbon fuels using simple stoichiometric analysis, and general chemical equilibrium modeling; 5. characterize properties of fuels in terms of octane number, cetane number and volatility; 6. quantify the impact of air/fuel equivalence ratio, mixing and combustion temperature on emissions; emission control; 7. characterize engine enhancements in terms of their impact on combustion efficiency, overall engine performance and emissions; 8. conduct laboratory tests on IC engines.
Indicative Contents المحتويات الإرشادية	Indicative content includes the following. 1-understand the fundamentals which govern internal combustion (IC) engine design and operation, the methods to improve the efficiency and limitations. 2-evaluate the performance of IC engines such as engine power, torque, fuel consumption and gas emissions. 3-analyse combustion processes, including the determination of air/fuel ratio and exhaust gas emissions. 4-perform basic analysis for designing a turbocharge system. 5-identify engine knock and understand the basic protection methods. 6-demonstrate the advancement of using renewable fuels. 7-process and analyses engine experimental data including generating p-V diagram and calculating indicated engine power.

Learning and Teaching Strategies استراتيجيات التعلم والتعليم	
Strategies	**Independent learning**Student learning outside the classroom is a key learning strategy in this subject. Students will do a large part of the learning in this subject by studying the materials such as the learning guide and sample questions. **Interactive learning** Interactive class discussion will be organized between students and their peers to enhance their understanding of the theories by applying the theories to solving problems. Students will work in groups under supervision and with assistance provided by the lecturer. This learning activity may incorporate a guest lecture from industry practitioners to improve students' knowledge of the recent technological developments and awareness of the real world. **Laboratory** Students, working in groups of 2-5, will undertake practical tasks related to engine testing. The laboratory work will provide students with the opportunity to become familiar with the instruments and equipment used for engine testing, and acquire experience in conducting engine experiments, processing experimental data and deriving useful results. **Feedback** Feedback, including common problems and detailed marking criteria, will be provided to the students after each of the assessment items are marked (except the final examination). Two one-hour consultations will be arranged weekly to assist students' study outside the class time.

Student Workload (SWL) الحمل الدراسي للطالب محسوب لـ ١٥ اسبوعا			
Structured SWL (h/sem) الحمل الدراسي المنتظم للطالب خلال الفصل	63	**Structured SWL (h/w)** الحمل الدراسي المنتظم للطالب أسبوعيا	4
Unstructured SWL (h/sem) الحمل الدراسي غير المنتظم للطالب خلال الفصل	62	**Unstructured SWL (h/w)** الحمل الدراسي غير المنتظم للطالب أسبوعيا	4
Total SWL (h/sem) الحمل الدراسي الكلي للطالب خلال الفصل	125		

Module Evaluation تقييم المادة الدراسية						
		Time/Number	Weight (Marks)	Week Due	Relevant Learning Outcome
Formative assessment	Quizzes	2	10% (10)	5, 10	LO #1, 2, 10 and 11
	Assignments	2	10% (10)	2, 12	LO # 3, 4, 6 and 7
	Projects / Lab.	1	10% (10)	Continuous	All
	Report	1	10% (10)	13	LO # 5, 8 and 10
Summative assessment	Midterm Exam	2 hr	10% (10)	7	LO # 1-7
	Final Exam	2hr	50% (50)	16	All
Total assessment			100% (100 Marks)		

Delivery Plan (Weekly Syllabus) المنهاج الاسبوعي النظري	
	Material Covered
Week 1	**Introduction** Introduction to internal combustion engine. Terminology, TDC & BDC, Stroke &swept volume, compression ratio.
Week 2	**I.C.E. Classification** Engine components and basic engine nomenclature. I.C. Engines classifications. Four stroke SI Engines. Four stroke CI Engines. Two stroke Engines. Fundamental differences between SI Engines and CI engines. Application of IC Engines. First law analysis of engine cycle-energy balance.
Week 3	**Air Standard Cycles** Introduction. Ideal or air standard cycles. Useful thermodynamic relations. The Carnot cycle. The Otto cycle. The Diesel cycle. The dual combustion cycle. Comparison of Otto, Diesel, and dual combustion cycles.
Week 4	**Operating Characteristic (Indicated and Effective values)** Engine Parameters. Work. Mean Effective Pressure. Torque and Power. Air-Fuel Ratio and Fuel-Air Ratio. Specific Fuel Consumption. Engine Efficiencies. Volumetric Efficiency. Emissions.
Week 5	**Fuels and Combustion** Hydrocarbon Fuels-Gasoline. Some Common Hydrocarbon Components. Self-Ignition and Octane Number. Diesel fuel. Chemical equilibrium. Combustion temperature. Adiabatic flame temperature. Liquid and gaseous combustion.
Week 6	**Fuel Injection**

	Heat release pattern and fuel injection. Requirements of a diesel injection system. Types of injection systems. Fuel pump. Types of fuel injectors. Injection nozzles. Quantity of fuel per cycle, size of nozzle orifice. Spray formation. Spray direction. Injection timing.
Week 7	**Ignition** Ignition system requirements. Battery ignition system. Magneto ignition system. Ignition Timing. Spark plugs. Disadvantage of conventional system. Electronic ignition system. Factors affecting spark plug operation.
Week 8	**Combustion in Spark Ignition Engines** Stages of combustion in S.I.E. Abnormal combustion. Ricardo's theory of combustion chamber. Basic types of combustion chamber in S. I. Engines.
Week 9	**Combustion in Compression Ignition Engines** Combustion stages in C.I engines. Factors effecting on ignition delay.Type of combustion chamber in C.I engines.
Week 10	**Lubrication System in I.C Engines** Lubrication principles. Function of lubrication. Properties of lubricating oil. Classification of lubricating oils. Oil Filters. Lubrication systems. Engine performance and lubrication.
Week 11	**Cooling System in IC Engines** Necessity of Engine cooling. Air Cooling. Water-cooling. Comparison of air and water-cooling systems. Radiators.
Week 12	**Supercharging** Objects of supercharging. Thermodynamic cycle with supercharging. Supercharging of spark ignition engine. Supercharging of C.I engine. Supercharging limits. Methods of supercharging. Turbo charging. Methods of Turbo charging. Limitations of Turbo charging.
Week 13	Supercharging of C.I engine. Supercharging limits. Methods of supercharging. Turbo charging. Methods of Turbo charging. Limitations of Turbo charging.
Week 14	**Rotary Engines** The working principle. Features of the rotary engines. Engine geometry. Combustion in rotary engines. Applications of rotary engines.
Week 15	**Air Pollution** Pollutants from gasoline engines. Emission control for Gasoline engine. Diesel emission. Diesel smoke and control. Comparison of diesel and gasoline emissions. Air pollution from gas turbine.
Week 16	**Preparatory week before the final Exam**

Delivery Plan (Weekly Lab. Syllabus)

المنهاج الاسبوعي للمختبر

	Material Covered
Week 1	Lab 1: Introduction to I.C. Engine essential parts and operating strokes.
Week 2	Lab 2: measuring devices ,tachometer ,dynamometer .orifice and manometer .
Week 3	Lab 3: speed ,torque and brake power measurement .

Week 4	Lab 4: Air consumption ,Fuel consumption ,air-fuel ratio measurement.
Week 5	Lab 5: brake thermal efficiency measurement.
Week 6	Lab 6: brake specific fuel consumption measurement .
Week 7	Lab 7: Engine performance parameters .

Learning and Teaching Resources مصادر التعلم والتدريس		
	Text	**Available in the Library?**
Required Texts	[1] Engineering fundamentals of the internal combustion engines by Willard P.	Yes
Recommended Texts	Internal combustion engines, by Mathur and Sharma	No
Websites	https://www.technicalbookspdf.com internal Combustion Engine by R K Rajput	

Grading Scheme مخطط الدرجات				
Group	**Grade**	التقدير	**Marks (%)**	**Definition**
Success Group (50 - 100)	**A** - Excellent	امتياز	90 - 100	Outstanding Performance
	B - Very Good	جيد جدا	80 - 89	Above average with some errors
	C - Good	جيد	70 - 79	Sound work with notable errors
	D - Satisfactory	متوسط	60 - 69	Fair but with major shortcomings
	E - Sufficient	مقبول	50 - 59	Work meets minimum criteria
Fail Group (0 – 49)	**FX –** Fail	راسب (قيد المعالجة)	(45-49)	More work required but credit awarded
	F – Fail	راسب	(0-44)	Considerable amount of work required

Note: Marks Decimal places above or below 0.5 will be rounded to the higher or lower full mark (for example a mark of 54.5 will be rounded to 55, whereas a mark of 54.4 will be rounded to 54. The University has a policy NOT to condone "near-pass fails" so the only adjustment to marks awarded by the original marker(s) will be the automatic rounding outlined above.

Northern Technical University-Iraq

Oil and Gas Technologies Engineering

College-Kirkuk

Fuel and Energy Engineering

Technologies Department

Internal Combustion Engines

Edited by/ Mr. Eng. Isam Ezzulddin Yousif

M.Sc. Pwer Engineering (2005)

2024-2025

Internal Combustion Engines

1. Introduction:

Engines which convert heat energy into mechanical energy are called heat engines. They are classified as Internal combustion engines (IC engines) and external combustion engines. An internal combustion engine is a heat engine in which the chemical energy of the combustion is released inside the engine cylinder, while in the other group of heat engines - S. eng. - energy is developed during combustion of a fuel is transmitted first to steam in a boiler. Internal combustion engines are preferred to other heat engines where the simplicity and low cost of operation are determining factors in the adoption of internal combustion engines.

2. Classification of Internal Combustion Engines:

Internal combustion engines are classified on the basis of working cycle, fuel, speed, thermodynamic cycle, method of cooling, applications, method of fuel ignition and cylinder arrangements as give:

1. Working Cycle:
   - Four - stroke engine.
   - Two - stroke engine.

2. Fuel
- Petrol or Gasoline engine.
- Diesel or Gas-oil engine.
- Gas engine.

3. Thermodynamic cycle:
- Otto cycle.
- Diesel cycle.
- Dual cycle.

4. Speed:
- Low speed engine - up to 500 r.p.m.
- Medium speed engine - up to 1000 r.p.m.
- High speed engine - above 1000 r.p.m.

5. Method of cooling:
- Air - cooled engine.
- Water - cooled engine.

6. Applications:
- Stationary engine.
- Marine engine.
- Automobile engine.
- Motor cycle engine.
- Aero engine.
- Locomotive engine.

7. Method of ignition,
- Spark ignition engine
- Compression ignition engine.

8. Arrangement of engine cylinder:

1. In-line engine.
2. V- engine.
3. X- engine.
4. Radial engine
5. H-engine.
6. Opposed cylinder and opposed piston engines.

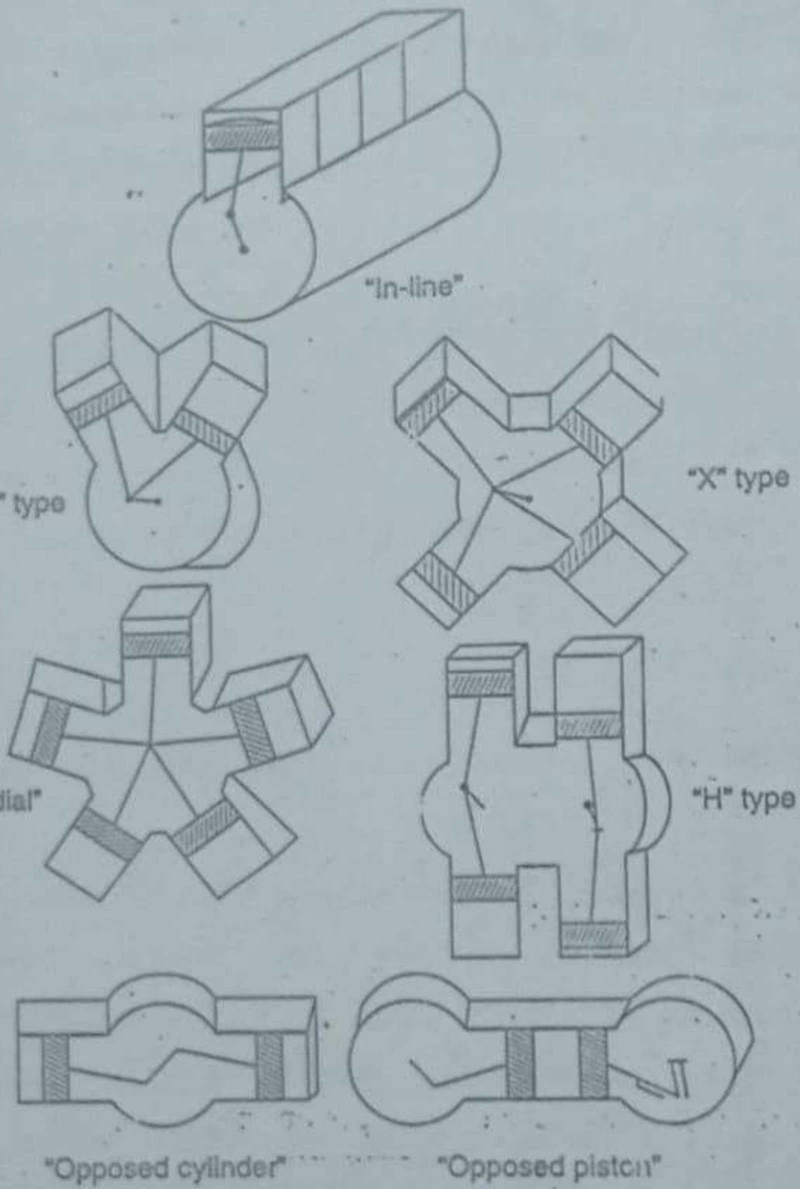


Fig.(1)- Engine Cylinder Arrangements

Internal Combustion Engines

(2-1) Working Principle of Four Stroke IC Engine

In four Stroke IC engine, one cycle of operation is completed in four strokes of the engine. One stroke is the distance which travelled by the piston from one end of the cylinder to another end during half revolution of the crankshaft. The topmost and bottommost positions up to which the piston can travel inside the cylinder are known as top dead center (TDC) and bottom dead center (BDC).

The distance between TDC and BDC is known as stroke length. One cycle of operation in a four-stroke engine consists of four strokes. They are (i) suction stroke (ii) compression stroke, (iii) power stroke and (iv) exhaust stroke.

(2-1-1) Suction Stroke

During this Stroke, the piston moves from TDC to BDC creating a vacuum inside the cylinder. The inlet valve is kept open and the exhaust valve is kept closed. The vacuum created inside the cylinder draws the charge (air and fuel vapour for SI engine and air alone in CI engine) into the cylinder through the inlet manifold and the inlet valve as shown in Fig. (2-1). This process is known as suction and is performed till the piston reaches BDC. The suction carried out during one stroke length of piston movement is called suction stroke.

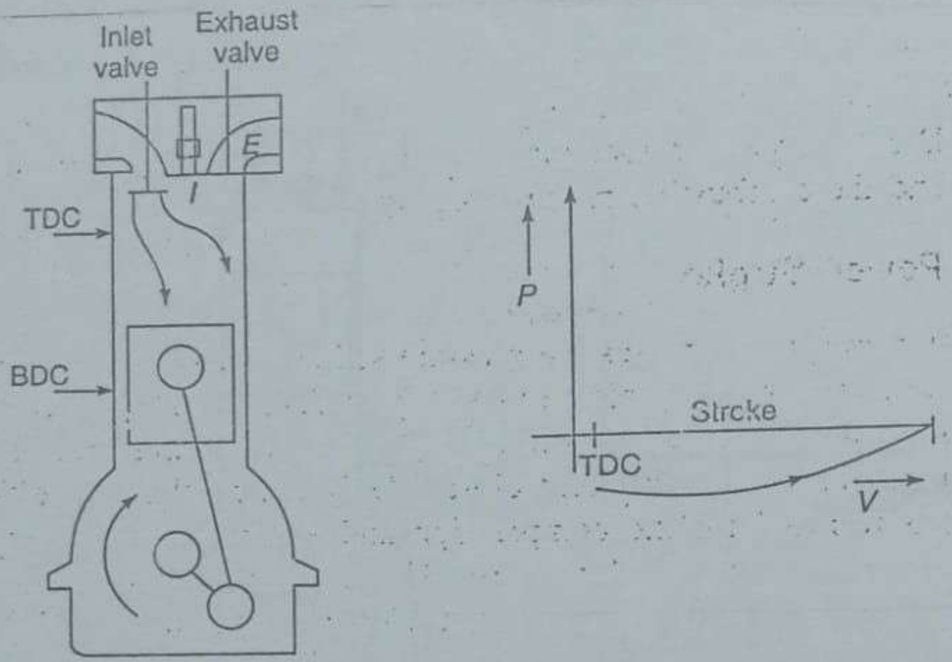


Fig. (2-1)

2.1.2 Compression Stroke

After the completion of the suction stroke, the piston moves from BDC to TDC. Both inlet and exhaust valves are kept closed during this stroke. The volume reduction of charge during this stroke increases the pressure. This process known as the compression and this stroke is called the compression stroke as shown in Fig. (2-2).

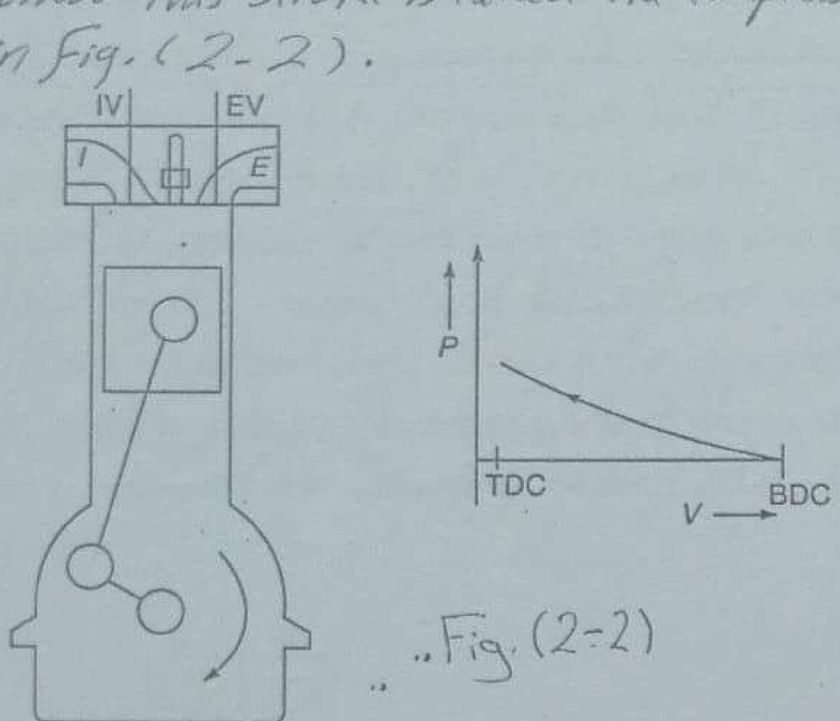
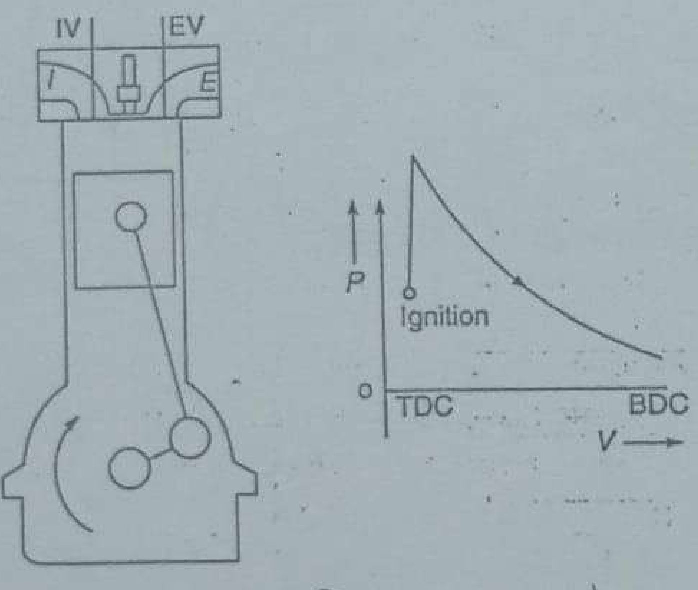


Fig. (2-2)

At the end of compression stroke, a spark is introduced by the spark plug in SI engine to initiate the combustion in which the flame produced by the charge consums the charge to increase the pressure instantaneously. In the case of CI engine, diesel fuel is injected into the compressed air at the end of the compression stroke. The compressed air at high temperature transfers heat to the diesel fuel vapour. Diesel vapour on recieving heat reaches its self-ignition temperature quickly and the combustion process begins.

(2.1.3) Power Stroke

Combustion process increases the pressure suddenly and creates an impact on the piston due to which the piston moves from TDC to BDC, generating work. This stroke is known as power stroke as shown in Figure (2-3). During this process also both inlet and exhaust valves are kept closed. Only during this stroke the engine developes power and for the compression and expansion strokes engine gets power from its crankshaft. Power required for compression and expansion strokes is only a fraction of the power developed by the engine and the balance power is transferred to the applications through the crankshaft. The engine initially needs power to crankshaft from an external force or source (manual or starting motor) till it develops its own power.



Fig(2-3) Power stroke.

(2-1-4) Exhaust Stroke

After the power stroke, the exhaust gas is pushed out of the cylinder through the exhaust valve which is kept opened during this stroke. Piston moves from BDC to TDC pushing the exhaust gas to the exhaust manifold as shown in Figure (2-4). Inlet valve is kept closed during this stroke to avoid the entery of exhaust gas into the inlet manifold. Compelete removal of exhaust gas is very important for the efficient combustion of the next cycle.

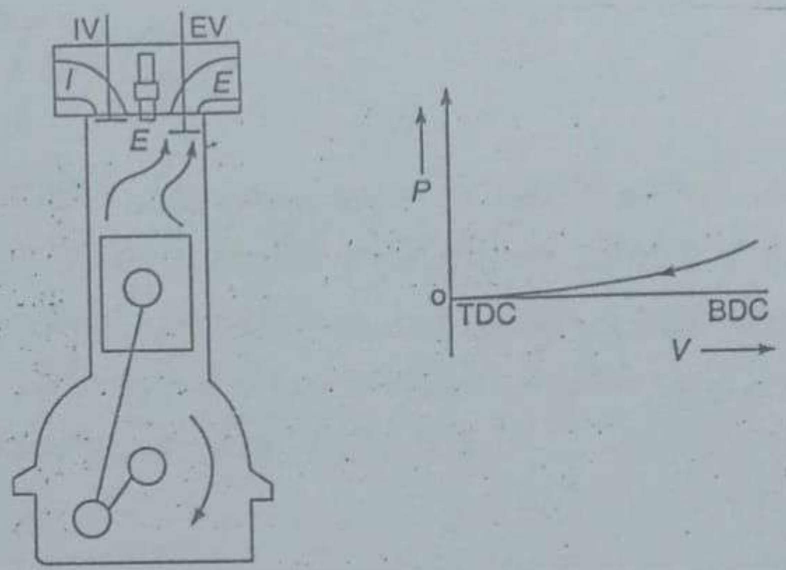


Fig (2-4) Exhaust stroke.

Internal Combustion Engines

(2-2) Thermodynamic Cycle

A cycle is formed with number of processes coupled in a particular sequence that can be repeated to convert heat into work or work into heat. Power producing cycles are formed in a particular combination to convert input heat energy - produced by the combustion of naturally available fuels - into work output.

(2-3) OTTO Cycle

Nicholas Otto has proposed a cycle with constant volume heat addition and rejection instead of isothermal heat addition and rejection as proposed by Carnot in his cycle. Compression and expansion are isentropic as shown in Fig. (2-5)

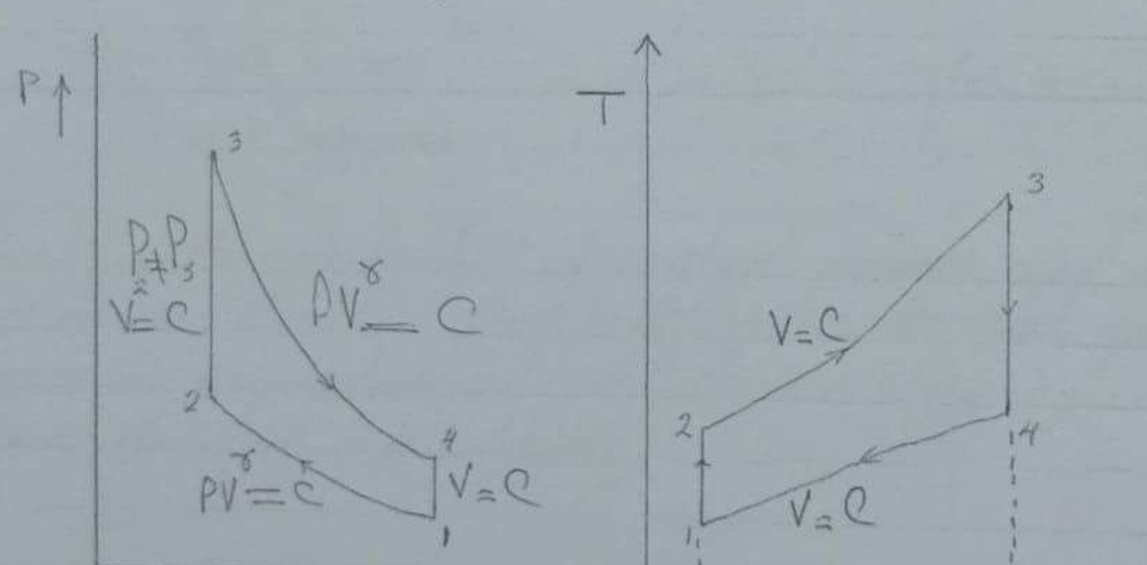


Fig. (2-5)

The Otto cycle has the following processes:
1-2: Isentropic compression, (work input).
2-3: Constant volume heat addition.
3-4: Isentropic expansion, (work output).
4-1: Constant volume heat rejection.

Air is compressed isentropically from state point 1 to 2 and then heat is added without allowing the volume to expand. This heat addition increases the pressure of the air as the volume is kept constant and this process is represented from state point 2 to 3.

The air is then allowed to expand isentropically from 3 to 4 and then further the pressure is dropped from 4 to 1 at constant volume. The thermal efficiency of this cycle is given as

$$\text{Otto cycle efficiency, } \eta_{otto} = \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}}$$

$$= \frac{\text{Net Work Done}}{\text{Heat Supplied}} = \frac{mC_v(T_3 - T_2) - mC_v(T_4 - T_1)}{mC_v(T_3 - T_2)}$$

where C_v is the specific heat at constant volume, and m is mass of air. Specific heat is defined as the heat required to raise the temperature of one kilogram of air by one degree. Its unit is kJ/kgK.

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

Also, $$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = \frac{T_3}{T_4}$$

Therefore,

$$\frac{T_3}{T_2} = \frac{T_4}{T_1}$$

$$\eta_{otto} = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{\frac{T_2}{T_1}} = 1 - \frac{1}{R^{\gamma-1}}$$

where R is compression ratio and γ is the index of compression. Also, $R = V_1/V_2$ and $\gamma = C_p/C_v$.

Otto cycle is the basis on which the petrol engine works. In the case of petrol engine, air and petrol vapour are compressed and ignited using a spark at constant volume to produce mechanical work. The petrol engine cycle is an open cycle unlike the Otto cycle. The gas produced by the combustion of petrol during the process 2-3 cannot be reused and hence discharged to atmosphere during the process 4-1 and fresh air and petrol mixture is taken for the next cycle.

Example (2-1):

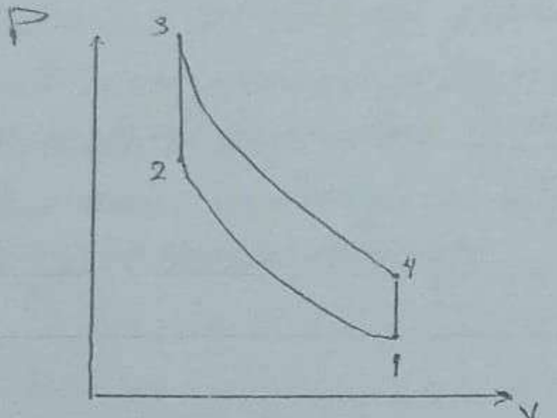
In an otto cycle the pressure and temperature at the beginning of compression are 98 kN/m² and 50°C. The compression ratio is 5:1. The heat supplied during the cycle is 930 kJ/kg. Determine;

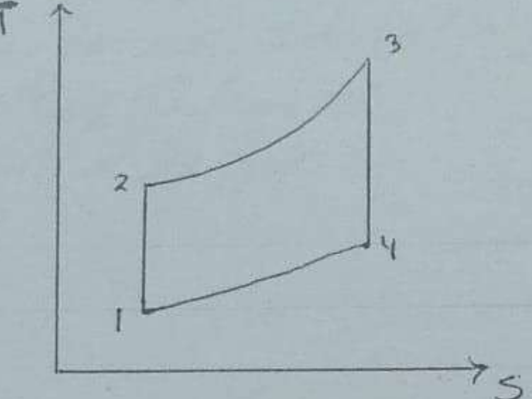
a- The maximum temperature in the cycle

b- The thermal efficiency.

c- The work done during the cycle.

Solution:-





$R = 5$, $q_{add.} = 930$ kJ/kg.

$$\frac{T_2}{T_1} = R^{\gamma-1} = 5^{0.4} = 1.904 \Rightarrow T_2 = 50 * 1.904 =$$

or; $T_2 = 323(5)^{0.4} = 615\,K$

$q_{add} = C_v(T_3 - T_2) = 0.718(T_3 - 615)$

$930 = 0.718(T_3 - 615) \Rightarrow T_3 = 1910\,K$

$$\frac{T_3}{T_4} = (R)^{0.4} \Rightarrow T_4 = \frac{1910}{5^{0.4}} = 1003\,K$$

$\therefore \eta_{otto} = 1 - \frac{1003}{1910} = 47.5\,\%$

Work done $= \eta_{otto} * q_{add.} = 0.475 * 930 = 441.75 \frac{kJ}{kg}$.

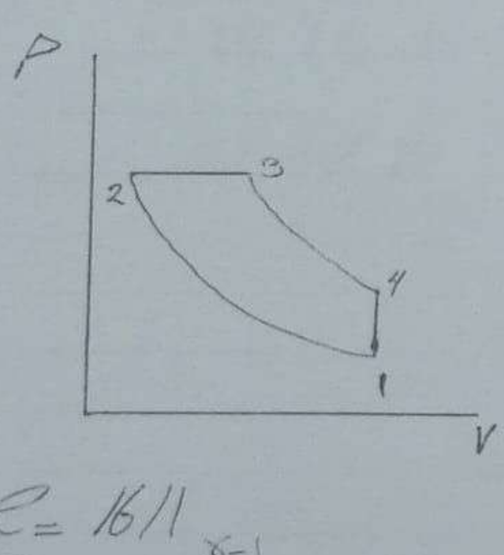
Ex(2-2):

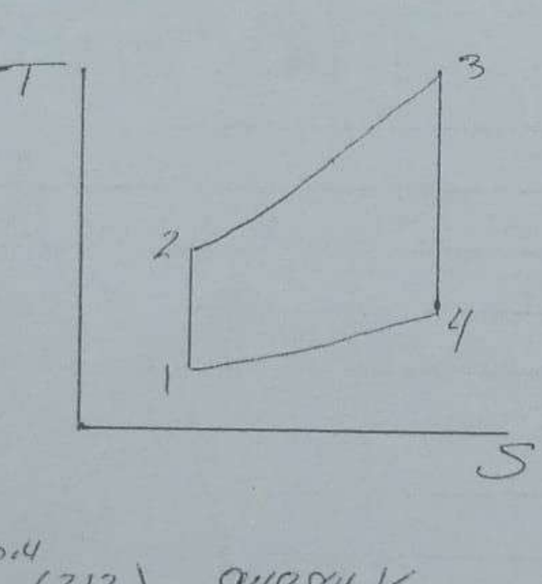
An engine works on the diesel cycle. At the beginning of compression stroke the pressure and temperature are 90 kN/m², 40°C, the compression ratio is 16:1. The heat added at constant pressure, until the temperature is 1400 °C; Calculate:

a- The pressure and temp. at all points of the cycle.

b- The cycle thermal efficiency.

Solution:-





$R = 16/1$

$\frac{T_2}{T_1} = R^{\gamma-1} = T_2 = 16^{0.4}(313) = 948.84\,K$

$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1} \Rightarrow \left(\frac{P_2}{90}\right)^{\frac{0.4}{1.4}} = \frac{948.84}{313} \Rightarrow \left(\frac{P_2}{90}\right)^{\frac{0.4}{1.4}} = 3.03$

Internal Combustion Engines

(2-4) Diesel Cycle

Rudolf Diesel proposed a cycle with constant pressure heat addition popularly known as Diesel cycle. The other processes of Diesel cycle are exactly similar to that of Otto cycle as shown in Fig. (2-6).

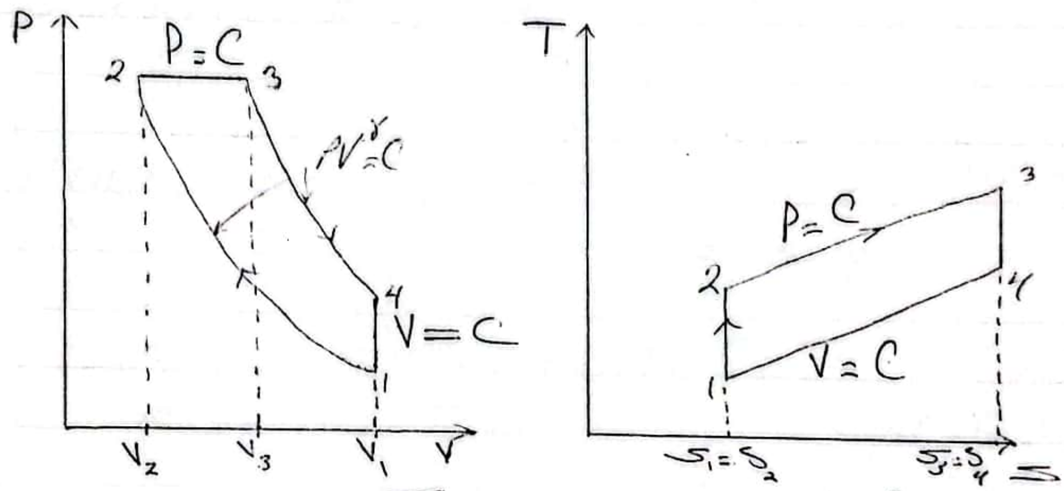


Fig. (2-6) Diesel Cycle.

The Diesel cycle has the following sequence of processes:

1-2: Isentropic compression.
2-3: Constant pressure heat addition.
3-4: Isentropic expansion.
4-1: Constant volume heat rejection.

Air is compressed from state point 1 to 2 isentropically and the heat is added during the process 2 to 3. Pressure is maintained constant by allowing the volume to expand during this heating process. Heat addition ends at state point 3 and then expansion of air follows isentropic law up to state point 4. Then the air pressure drops from 4 to 1 at constant volume.

As the heat transfer between the system and the surrounding is zero during the isentropic compression and expansion, the network done is the difference between the heat supplied and heat rejected.

$$\text{Diesel cycle efficiency, } \eta_{diesel} = \frac{\text{Heat supplied} - \text{Heat rejected}}{\text{Heat supplied}} = \frac{N.W.D.}{H.S.}$$

$$= \frac{mC_p(T_3 - T_2) - mC_v(T_4 - T_1)}{mC_p(T_3 - T_2)} = \frac{C_p(T_3 - T_2) - C_v(T_4 - T_1)}{C_p(T_3 - T_2)}$$

$$= 1 - \frac{C_v(T_4 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)} =$$

where,

$$\frac{T_2}{T_1} = R^{\gamma-1} \implies T_1 = \frac{T_2}{R^{\gamma-1}}$$

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c = \text{Cut-off ratio} = T_3 = T_2\, r_c$$

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = \left(\frac{V_3}{V_4} \cdot \frac{V_2}{V_2}\right)^{\gamma-1} = \left(r_c \cdot \frac{V_2}{V_1}\right)^{\gamma-1} = \left(r_c \frac{1}{R}\right)^{\gamma-1}$$

$$\therefore \frac{T_4}{T_3} = \left(\frac{r_c}{R}\right)^{\gamma-1}, \text{or, } T_4 = T_2\, r_c \left(\frac{r_c}{R}\right)^{\gamma-1}$$

$$\eta_{th.} = 1 - \frac{T_2\left(\frac{T_4}{T_2} - \frac{T_1}{T_2}\right)}{\gamma T_2\left(\frac{T_3}{T_2} - 1\right)} = 1 - \frac{\left(\frac{T_4}{T_2} - \frac{T_1}{T_2}\right)}{\gamma\left(\frac{T_3}{T_2} - 1\right)}$$

$$\eta_{th.} = 1 - \frac{\left(\frac{r_c^{\gamma}}{R^{\gamma-1}} - \frac{1}{R^{\gamma-1}}\right)}{\gamma\left(r_c - 1\right)}$$

$$\therefore \eta_{th.} = 1 - \frac{\left(r_c^{\gamma} - 1\right)}{\gamma R^{\gamma-1}\left(r_c - 1\right)}$$

Internal combustion engines which use diesel as fuel work on the Diesel cycle. The efficiencies of the Diesel and otto cycles are different. The normal range of compression ratio for Diesel engine is 16 to 20 and that for petrol engine which works on Otto cycle basis is 6 to 10. Otto cycle compression ratio is kept at lowers levels to avoid the self-ignition of petrol during the compression process itself.

(2-5) Dual Cycle

In the dual cycle, shown in Fig.(2-7), heat addition occurs partially during a constant volume process and the reminder during a constant pressure process.
With $R = V_1/V_2$, $r_c = V_4/V_3$ and $r_p = P_3/P_2$, the efficiency is given by,

Dual cycle efficiency, $\eta_{dual} = 1 - \frac{1}{R^{\gamma-1}}\left[\frac{r_p r_c^{\gamma} - 1}{(r_p - 1) + \gamma r_p (r_c - 1)}\right]$

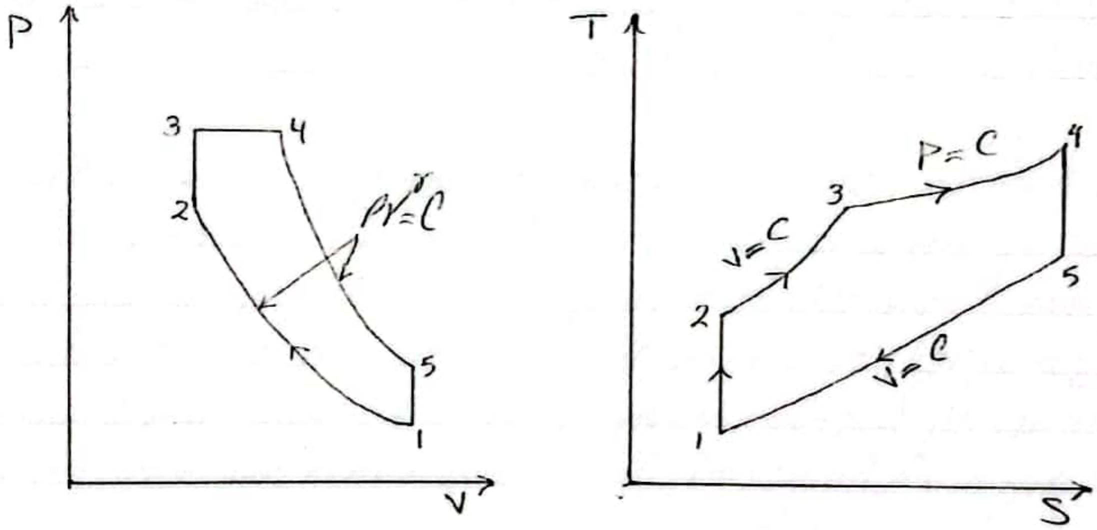


Fig.(2-7) Dual Cycle.

Air-standard cycle efficiencies decrease in the order of Otto, dual and Diesel for the same compression ratio.

(2-6) Mean Effective Pressure

Mean effective pressure is usually preferred to compare air-standard cycles of reciprocating engines. The mean effective pressure P_m is defined as the height of a rectangle on the P-V diagram having the same length and area as the cycle as shown in Fig. (2-8).

where, $P_m(V_1 - V_2) = \int P dV = W$

Mean effective pressure, $P_m = \dfrac{W}{(V_1 - V_2)} = \dfrac{\text{Work output}}{\text{Stroke volume}}$

where W is the network output per unit mass of fluid. Mean effective pressure P_m can be taken as constant pressure acting on the piston during power stroke which can produce the network of the cycle. A cycle with a higher mean effective pressure will produce a large work output per unit swept volume and the engine size will be small for a given work output.

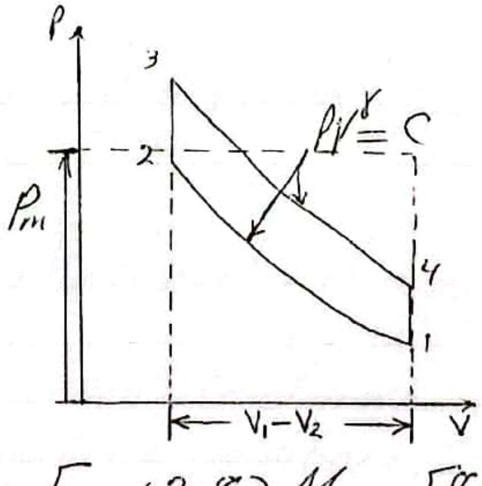


Fig. (2-8) Mean Effective Pressure (Otto Cycle).

(2-7) Work Ratio

Work ratio is the ratio between the net work done and the useful work of the engine.

$$\text{Work ratio}_{otto} = \frac{\text{network done}}{\text{useful work}} = \frac{Q_{add.} - Q_{rej.}}{W_{3-4}}$$

$$= \frac{C_v(T_3 - T_2) - C_v(T_4 - T_1)}{C_v(T_3 - T_4)}$$

$$\text{Work ratio} = 1 \mp \frac{T_1 - T_2}{T_3 - T_4}$$

(2-8) Air Standard Cycle Assumptions

In the air standard cycle the working fluid is assumed to be air. The following assumption are made in the analysis of standard cycle;

1. The working fluid is a perfect gas that mean it is follow the law; $P\vartheta = RT$.

2. The working fluid is a fixed mass of air.

3. The physical constants of working fluid are same as those of air at atmospheric conditions, [$C_p = 1.005 \frac{kJ}{kg K}$,

$C_v = 0.718$ kJ/kgK , $R = 0.287$ kJ/kgK , $\gamma = 1.4$].

4. The working fluid has a constant specific heats.

5. The working fluid does not undergo any chemical change throughout the cycle.

6. Heat supply and rejected in a reversible manner.

7. The compression and expansion processes are reversible adiabatic (S = Constant).

8. The kienetic and potential energy of working fluid are neglected.

$\therefore \eta_{Otto} = 1 - \frac{1003}{1910} = 47.5\,\%$

Work done $= \eta_{Otto} * q_{add.} = 0.475 * 930 = 441.75 \frac{kJ}{kg}$.

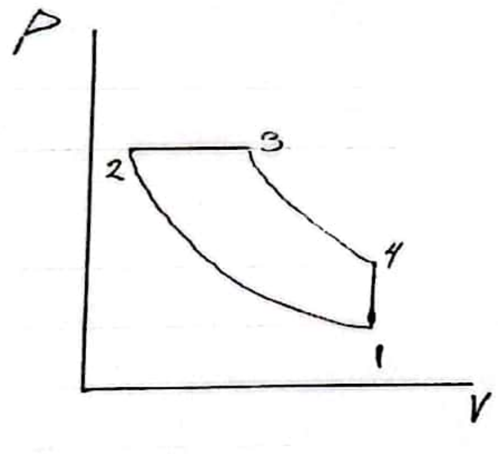
Ex(2_2):

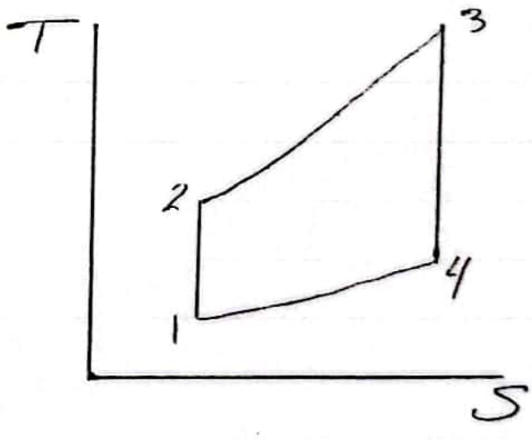
An engine works on the diesel cycle. At the beginning of compression stroke the pressure and temperature are 90 kN/m², 40°C, the compression ratio is 16:1. The heat added at constant pressure, until the temperature is 1400 °C; Calculate:

a_ The pressure and temp. at all points of the cycle.

b_ The cycle thermal efficiency.

Solution:-





$r = 16/1$

$\frac{T_2}{T_1} = r^{\gamma-1} = T_2 = 16^{0.4}(313) = 948.84\,K$

$\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1} \Rightarrow \left(\frac{P_2}{90}\right)^{\frac{0.4}{1.4}} = \frac{948.84}{313} \Rightarrow \left(\frac{P_2}{90}\right)^{\frac{0.4}{1.4}} = 3.03$

$$\ln\left(\frac{P_2}{90}\right)^{0.286} = \ln 3.03 \Rightarrow 0.286 \ln \frac{P_2}{90} = \ln 3.03$$

$$\ln P_2 - \ln 90 = 3.876 \Rightarrow \ln P_2 = 8.576 \Rightarrow P_2 = 4341.2\,kPa = P_3$$

$$\frac{T_4}{T_3} = \left(\frac{r_c}{R}\right)^{\gamma-1} \Rightarrow T_4 = 1673\left(\frac{r_c}{16}\right)^{0.4}$$

$$r_c = \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{1673}{948.84} = 1.763$$

$$T_4 = 1673\left(\frac{1.763}{16}\right)^{0.4} = 692.42\,K$$

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow 0.414 = \left(\frac{P_4}{4341.2}\right)^{0.286} \Rightarrow -3.08 = \ln P_4 - 8.38$$

$$\therefore P_4 = 199.5\,kPa.$$

$$\eta_{th} = 1 - \frac{(r_c^{\gamma}-1)}{\gamma R^{\gamma-1}(r_c-1)} = 1 - \frac{(1.763^{1.4}-1)}{1.4(16^{0.4})(1.763-1)} = 1 - \frac{1.212}{3.238}$$

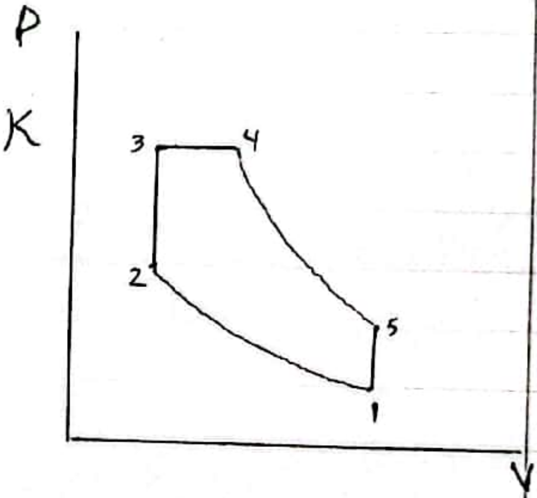
$$\eta_{diesel} = 62.57\,\%$$

Ex (2-3):

A dual combustion cycle has an adiabatic compression ratio 15:1, the conditions at beginning of compression state are 97 kN/m², 0.084 m³ and 28°C. The maximum temp. is 1320 °C and the maximum pressure is 62 bar. Calculate:

a- The work done.
b- The thermal efficiency.
c- The heat added.
d- The m.e.p..

Solution :-



$P_1 = 97\ kN/m^2,\ V_1 = 0.084 m^3,\ T_1 = 301 K$

$P_3 = P_4 = 6200\ kN/m^2,\ T_4 = 1593 K$

$$T_2 = T_1 R^{\gamma-1} = 301 * 15^{0.4} = 890 K$$

$$P_2 = P_1 R^{\gamma} = 97 * 15^{1.4} = 4282 \frac{kN}{m^2}$$

$$\frac{T_3}{T_2} = \frac{P_3}{P_2} \Rightarrow T_3 = 890 \frac{62}{42.82} = 1290 K$$

$$V_2 = \frac{V_1}{R} = 0.0056 m^3 = V_3$$

$$\frac{T_3}{T_4} = \frac{V_3}{V_4} \Rightarrow V_4 = V_3 \frac{T_4}{T_3} = 0.0056 \frac{1593}{1290} = 0.00692 m^3$$

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{\gamma-1} \Rightarrow T_5 = 1593 \left(\frac{0.00692}{0.084}\right)^{0.4} \Rightarrow T_5 = 586 K$$

$$\frac{P_3}{P_2} = r_p = \frac{62}{42.82} = 1.448$$

$$r_c = \frac{V_4}{V_3} = \frac{0.00692}{0.0056} = 1.236$$

$$\eta_{dual} = 1 - \frac{(r_p r_c^{\gamma} - 1)}{R^{\gamma-1}[(r_p - 1) + \gamma r_p(r_c - 1)]}$$

$$= 1 - \frac{(1.448(1.236)^{1.4} - 1)}{15^{0.4}[(1.448-1) + 1.4(1.448)(1.236-1)]} = 1 - \frac{0.948}{2.95(0.926)}$$

$$\eta_{dual} = 1 - 0.347 = 65.3\%$$

$$P_1 V_1 = mRT_1 \Rightarrow m = \frac{P_1 V_1}{RT_1} = \frac{(97)(0.084)}{(0.287)(301)} = 0.094\,kg$$

$$Q_{add} = m\,q_{add.}$$

$$= 0.094(0.718(1290 - 890) + 1.005(1593 - 1290))$$

$$Q_{add.} = 55.648\ kJ$$

$$W_{done} = \eta_{dual}\, Q_{add} = 36.3\ KJ$$

$$m.e.p. = \frac{W_{done}}{(V_1 - V_2)} = \frac{36.3}{(0.084 - 0.0056)} = 463\ kN/m^2$$

Practice Problems

1. The compression ratio of an Otto cycle is 7. Find the air standard efficiency. Assume that the index of compression and expansion is 1.4.

2. Air flows into a gasoline engine at 0.95 bar and 300 K. The compression ratio is 8. In the combustion process 1300 kJ/kg of heat is added. Find the temperature and pressure after combustion.

3. A spark-ignition engine works an air-standard Otto cycle that has a heat addition of 1800 kJ/kg and a compression ratio of 7. The pressure and the temperature at the beginning of the compression process are 0.9 bar and 283 K. Determine the maximum pressure and temperature of the cycle, thermal efficiency of the cycle and the mean effective pressure.

4. A diesel engine works with a compression ratio of 20. The inlet conditions of air are 0.95 bar, 290 K and 0.5 litre volume. The maximum cycle temperature is 1800 K. Find the maximum pressure, net specific work and the thermal efficiency.

5. In an ideal air-standard Diesel cycle, the state before the compression process is 0.95 bar, 290 K. The compression ratio is 20. Find the maximum temperature (by iteration) in the cycle to have a thermal efficiency of 60 %.

6. The cut-off ratio and compression ratio of an air standard Diesel cycle are 1.65 and 14. Assume $\gamma = 1.4$, find the air standard efficiency.

7. An ideal Diesel cycle has a compression ratio of 14, takes in air at 1 bar and 293 K. If cut-off takes place at 5% of the stroke, find the mean effective pressure of the cycle.

8. An ideal Diesel cycle operates on 1 kg of standard air with an initial pressure of 1 bar and a temperature of 35°C. The pressure at the end of compression is 33 bar and the cut-off 6% of the stroke. Determine:

(i) the compression ratio.
(ii) the percentage clearance.
(iii) the heat supplied.
(iv) the heat rejected.

Take; $\gamma = 1.4$, and $C_p = 1$ kJ/kg.

2006

5-1) Valve Timing Diagrams for Four-Stroke IC Engine

The diagram representing the opening and closing of the inlet and exhaust valves during one cycle of operation in a four-stroke IC engine is known as the valve timing diagram. The process of the cycle are represented with respect to movement of crankshaft in angles as shown in Figure (5-1). Inlet valve opening (IVO) is advanced before (TDC) and the inlet valve closing (IVC) is delayed after (BDC) by a few degrees to maximize the suction process. More the suction, higher the power developed by the engine. Fuel injection beginning (FIB) and spark ignition (SI) are advanced to complete the Combustion just after (TDC). Exhaust valve opening (EVO) is advanced before (BDC) at the cost of power stroke and its closing is delayed (EVC) is after (TDC) by a few degrees to increase exhaust stroke to ensure a maximum removal of exhaust gases, thus an interference state happened due to the opening of inlet valve at the exhaust stroke and closing of exhaust valve at the suction stroke in its 10° (ATDC), this case is called as the overlapping period. The diagram is shown in Figure (5-1).

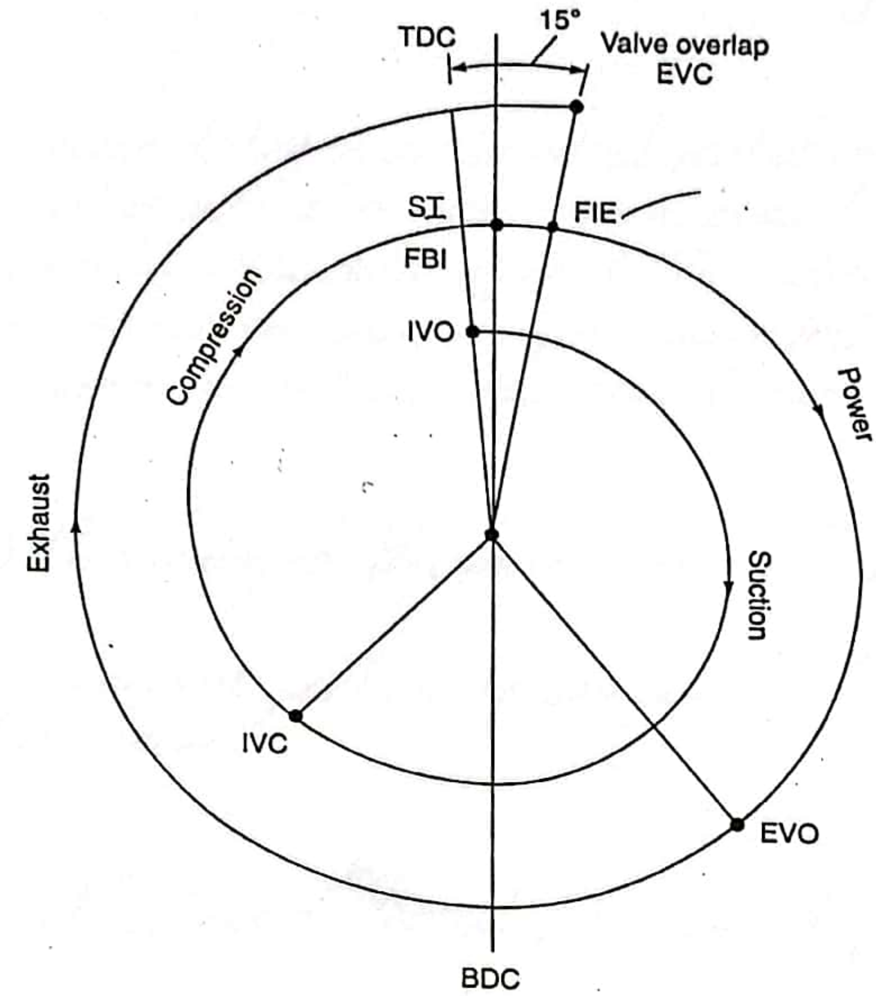


Figure (5-1): Valve Timing Diagram.

(6-1) Performance and Testing of IC Engines

Performance testing is an important part of research and development of internal combustion engines and the test facilities very widely with data required. The modern facilities used for research can have comprehensive instrumentation with computer control and data acquisition.

(6-2) Performance Parameters of IC Engine:

The important performance parameters of IC engine are defined as follow;

(6-2-1) Indicated Thermal Efficiency (η_{ith})

It is the ratio of indicated power developed on the piston to the energy content of the fuel supplied. It represents the conversion factor of chemical energy of the fuel (into) linear mechanical power.

$$\eta_{ith} = \frac{\text{Indicated Power}}{\text{Fuel Power}}$$

where

$$\text{Indicated Power} = \frac{P_{imef} L A n K}{60000} \text{ kW}$$

P_{imef} = Indicated mean effective pressure N/m^2
L = Stroke Length.
A = $\pi d^2/4$ where d = diameter of cylinder or bore
n = $N/2$ for Four-stroke engine
= N for Two-stroke engine
N = rpm of the engine
K = number of cylinders

Fuel Power (kW) = Mass flow rate of fuel (kg/s) * Calorific value of fuel (kJ/kg)

$$\text{Brake mean effective pressure, } P_{bmef} = \frac{BP * 60000}{L A n K}$$

where BP is brake power in kW.

Mean effective pressure: Mean effective pressure is the constant pressure which, acting on the piston area during the power stroke, would produce the indicated or mean effective power per cycle.

(6-2-2) Brake Thermal Efficiency (η_{bth}):

It is the ratio of brake power coming out of the engine so through the crankshaft to the fuel power. It represents the overall efficiency of the engine or the actual engine efficiency.

$$\eta_{bth} = \frac{\text{Brake Power}}{\text{Fuel Power}} = \eta_{overall} = \eta_{actual}.$$

where

Brake power $= 2\pi NT/60000$ kW

Further N = rpm
T = Torque on the output shaft, N.m,
= Net load * Effective radius, N.m.

(6-2-3) Mechanical Efficiency (η_m)

It is the ratio of brake power to indicated power. Mechanical efficiency represents the conversion factor of linear mechanical power to rotary mechanical power in the mechanical linkage of connecting rod and crankshaft. The difference between them is the friction loss. By driving a non-firing engine by an auxiliary motor, the measurement of total friction losses can be done.

where,

$$\eta_m = \frac{\text{Brake Power}}{\text{Indicated Power}}$$

(6_2_4) Volumetric Efficiency (η_v)

It is the ratio of the actual volume of air that goes to the cylinder to the stroke or swept volume of the cylinder.

$$\eta_v = \frac{\text{Actual volume of air}}{\text{Stroke (or) Swept volume}}$$

where

$$\text{Stroke volume (or) Swept volume} = \frac{\pi d^2}{4} * L * K$$

When the engine runs at N rpm then the volumetric efficiency can also be represented as the ratio of actual volume flow rate to the theoretical volume flow rate.

$$\eta_v = \frac{\text{Actual volume flow rate}, (m^3/sec)}{\frac{\pi d^2}{4} * L * \frac{N}{60} * K, (m^3/sec)}$$

(6-2-5) Piston Speed

Another variable that is more or less comparable among engines of otherwise very different natures is the average piston speed. Piston speed is proportional to engine speed and is averaged over a cycle.

The piston speed can be calculated from the rpm(N) of the crankshaft and stroke length (L) as

$$\text{Piston Speed} = 2LN/60 \;, \; m/s$$

(6-2-6) Specific Fuel Consumption (SFC)

It is the ratio of fuel consumption to the brake power produced.

$$SFC = \frac{\text{Fuel Consumption (kg/hr)}}{\text{Brake power (kW)}} \;, \; \frac{kg}{kW.hr}$$

(6_2_7) Air Fuel Ratio

It is the amount of air required for burning 1 kg of fuel completely.

$$A/F\ Ratio = \frac{\text{Air consumed}}{\text{Fuel consumed}}$$

(6_2_8) Relative Efficiency (η_R)

It is the ratio of thermal efficiency of the engine to the air standard efficiency on which it works. It represents the deviation between the cycle efficiency and engine efficiency. The relative efficieny is usually calculated based on indicated thermal efficieny and sometimes based on brake thermal efficiency.

$$\eta_{R(thi)} = \frac{\text{Indicated thermal efficiency}}{\text{Air standard cycle efficieny}}$$

$$\eta_{R(thb)} = \frac{\text{Brake thermal efficiency}}{\text{Air standard cycle efficiency}}$$

Practice Problems

1. The following data were obtained during the testing of a four-stroke diesel engine having a bore of 95 mm and stroke of 155 mm. Eng. speed is 400 r.p.m. Load on brake drum is 200 N. Spring balance reading is 20 N. Brake diameter for drum is 550 mm. Area and length of indicator diagram are 400 mm^2 and 60 mm respectively. Spring constant is 1 bar per mm. Specific fuel consumption is 0.28 kg/kWhr and calorific value of fuel is 42,000 kJ/kg. Find the mecha., brake thermal and indicated thermal efficiencies.

2. The Morse test on a four-stroke four cylinder petrol engine having bore of the cylinder = 80 mm, stroke length = 120 mm, clearance volume = 8 $\times 10^4$ mm^3 gave the following data.

   Fuel consumption = 4.74 kg/hr, calorific value of fuel = 44,000 kJ/kg, BP with all cylinders working = 14.5 kW, BP with cylinder 1 cut-off = 10 kW, BP with cylinder 2 cut-off = 10.2 kW, BP with cylinder 3 cut-off = 10.1 kW and BP with cylinder 4 cut-off = 10.3 kW. Estimate mechanical, indicated, brake thermal and relative efficiencies.

3. A test was conducted on a single-cylinder two-stroke engine where net brake load on engine is 600 N, speed is 400 rpm, mean effective pressure is 3.5 bar, fuel consumption is 4.5 kg/hr, cooling water flow rate is 500 kg/hr, water inlet and outlet temperatures are 30 °C and 55 °C, respectively, air-fuel ratio is 26, air temperature is 30 °C, exhaust gas temperature is 320 °C, cylinder bore is 21 cm, stroke length is 25 cm, brake drum diameter is 98 mm, calorific value of fuel is 44 000 kJ/kg. Estimate the mechanical efficiency, indicated thermal efficiency and draw up a heat balance chart on percentage basis.

4. A two-stroke cycle CI engine has 14 cylinders and a total engine piston displacement of 7375 cc. While operating at a speed of 1620 rpm, it produces a continuous torque reading of 8.1 kN.m. At this speed, the engine is known to have a brake specific fuel consu. of 0.371 kg/kWhr. Calculate BP, bmep and time to consume 18925 lit of fuel (specific gravity = 0.84).

Ossama
23-12-2006

Practice Problems

5. A four-stroke cycle automobile engine is tested while running at 3600 rpm. Inlet air temperature is 16 °C and the pressure is 101.36 kN/m². The engine has eight in-line cylinders with a total piston displacement of 4066 cc. The air-fuel ratio is 14 and the bsfc is 0.377 kg/kWhr. Dynamometer readings show a power output of 86 kW. Find the volumetric efficiency.

6. A four-cylinder, four-stroke diesel engine has a bore of 212 mm and a stroke of 292 mm. At full load at 720 rpm the bmep is 5.93 bar and the specific fuel consumption is 0.226 kg/kW.hr. The air-fuel ratio as determined by exhaust gas analysis is 25. Calculate the brake thermal efficiency and the volumetric efficiency of the engine. Atmospheric conditions are 1.01 bar and 15 °C and calorific value for the fuel may be taken as 44,200 kJ/kg.

7. An air box was fitted to the suction side of an engine which had an orifice of 42 mm diameter with a discharge coefficient of 0.6. The manometer reading observed is 35 mm of water. Assume the density of the air at inlet as 1.215 kg/m³, find the BP and air-fuel ratio if the fuel consumption is 2 ml/sec, load on the dynamometer is 162 N, dynam constant is 26,830, the engine speed is 3000 rpm and the specific gravity of fuel is 0.724.

8. A four-cylinder petrol engine has an output of 52 kW at 2000 rpm. A Morse test is carried out and the brake torque readings are 177, 170, 168 and 174 N.m respectively. For normal running at this speed the specific fuel consumption is 0.364 kg/kW hr. The calorific value of the fuel is 44,200 kJ/kg. Calculate the mechanical and brake thermal efficiencies of the engine.

Issam
23-12-2006

Ex1:- A twin-cylinder two-stroke engine has a swept volume of 150 cm³. The maximum power output is 19 kW at 1100 rpm, bsfc is 0.11 kg/MJ and the air/fuel ratio is 12. If ambient test conditions were 10 °C and 1.03 bar and the fuel has a calorific value of 44 MJ/kg. Calculate the bmep, overall efficiency and the volumetric efficiency.

Solution:-

$$bsfc. = \frac{\text{Fuel consumption}}{BP}$$

$$0.11 \times 10^{-6} = \frac{\text{Fuel consum.}}{19 \times 10^{3}\ J/sec}$$

$$\text{Fuel consumption} = 0.11 \times 10^{-6} \times 19 \times 10^{3}\ (J/sec \cdot kg/J)$$
$$= 2.09 \times 10^{-3}\ kg/sec$$

$$bmep = \frac{BP}{(\pi d^2/4)(l n K)}$$

$$(\pi d^2 l/4) \times n \times K = \frac{150 \times 10^{-6} \times 1100 \times 2}{60}\ m^3/sec$$
$$= 0.055\ m^3/sec$$

$$bmep = \frac{19 \times 10^{3}}{0.055} = 345454.55\ N/m^2 = 3.455\ bar$$

Overall efficiency or brake thermal efficiency,

$$\eta_{overall} = \frac{\text{Brake Power}}{\dot{m}_f \times CV}$$

$$\eta_{overall} = \frac{19000}{2.09 \times 10^{-3} \times 44 \times 10^{6}} \times 100 = 20.66\ \%.$$

$$\text{Volumetric efficiency}, \eta_{vol} = \frac{\text{Actual volume of air intake}}{\text{Theoretical volume flow rate}} = \frac{\dot{V}_a}{\dot{V}_s}$$

where, $\frac{\text{Air flow}}{\text{Fuel flow}} = 12$, so

$$\text{Air flow} = 12 \times 2.09 \times 10^{-3}\ kg/sec.$$
$$\text{Air flow} = 0.02508\ kg/sec.$$
$$\dot{m}_a = 0.02508\ kg/sec.$$

$$PV = mRT$$
$$P_1 \dot{V}_1 = \dot{m}_a R T_1$$

$$\dot{V}_1 = \frac{0.02508 \times (287) \times (283)}{1.03 \times 10^{5}} \Rightarrow \dot{V}_1 = \dot{V}_a = 0.01977\ m^3/sec.$$

$$\%vol. = \frac{0.01977}{150 \times 10^{-6} \times \frac{11000}{60}} = 0.719 = 71.9\%.$$

9. Combustion in SI Engines:

9.1. Introduction:

Combustion may be defined as a relatively chemical combination of hydrogen and carbon in the fuel with the oxygen in the air resulting in liberation of energy in the form of heat.

The conditions necessary for combustion are:
(1) the presence of a combustible mixture;
(2) some means of initiation combustion, and
(3) stabilization and propagation of flame in the combustion chamber.

In SI engines the combustible mixture is generally supplied by the carburettor and the combustion is initiated by an electric spark given by a spark plug.

A chemical equation for combustion process of any hydrocarbon can be easily written. For C_8H_{18} (iso-octane) the equation is;

$$C_8H_{18} + 12.5O_2 \longrightarrow 8CO_2 + 9H_2O \quad \text{---(9-1)}$$

It is well known that the combustion process is not a simple and direct combination of atoms as indicated by the chemical equation.

9.2. Ignition Limits :

Experiments have shown that ignition of the charge is only possible within certain limits of fuel-air ratio. These "ignition limits" correspond approximately to those mixture ratios, at lean and rich ends of the scale, where the heat released by spark is no longer so sufficient to initiate combustion in the neighbouring unburnt mixture. It is generally agreed that the flame will propagate only if the temperature of the burnt gases exceeds approximately 1500 K in the case of hydrocarbon-air mixture.

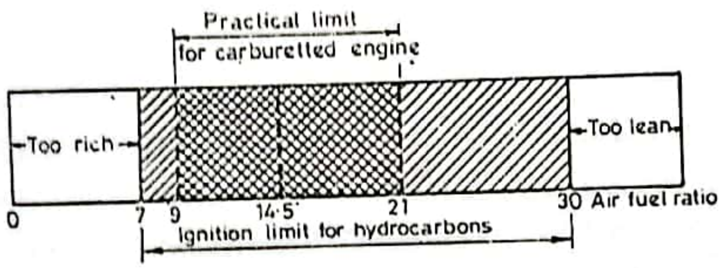


Fig. (9-1) – Ignition limits for hydrocarbons.

For hydrocarbon fuel the stoichiometric fuel-air ratio is about 1:15 and hence the fuel-air must be between about 1:30 and 1:7 as shown in Fig. (9-1). The lower and upper ignition limits of the mixture depend upon mixture ratio and temperature of the fuel itself. The ignition limits are wider at increased temperatures because of the higher rates of reaction and higher thermal diffusivity coefficients of the mixture.

9.3. Stages of Combustion in SI Engine:

In spark-ignition engine a sufficiently homogeneous mixture of a vapourized fuel-air and the residual gases is ignited by a single intense and high temperature spark between the spark plug electrodes (at the moment of discharge the temperature of the electrodes exceeds 10,000°C), leaving behind a thin thread of flame. From this thin thread combustion spreads to the envelop of mixture immediately surrounding it at a rate which depends primarily upon the temperature of the flame front itself and to a secondary degree, upon both the temperature and the density of the surrounding envelope.

In this manner there grows up, gradually at first, a small hollow nucleus of flame, much in the manner of a soap bubble. If the contents of the cylinder were at rest, this flame bubble will expand with steadily increasing speed until extended throughout the whole mass. In the actual engine cylinder, however, the mixture is not at rest. It is, in fact, in a highly turbulent condition. The turbulence breaks the filament of flame into a ragged front, thus presenting a far greater surface area from which heat is radiated; hence its advance is speeded up enormously. The rate at which the flame front travels is dependent primarily on the degree of turbulence, but its general direction of movement, that of radiating outward from the ignition point, is not much affected.

The theoretical diagram of combustion is shown in Fig.(9-2a) but the actual process is different. Where according to Ricardo the combustion can be imagined as if developing in two stages, one-the growth and the developement of a semipropagating nucleus of flame called "ignition lag" or preparation phase, and the other, and the other, the spread of the flame throughout the combustion chamber (see Fig (9-2b)).

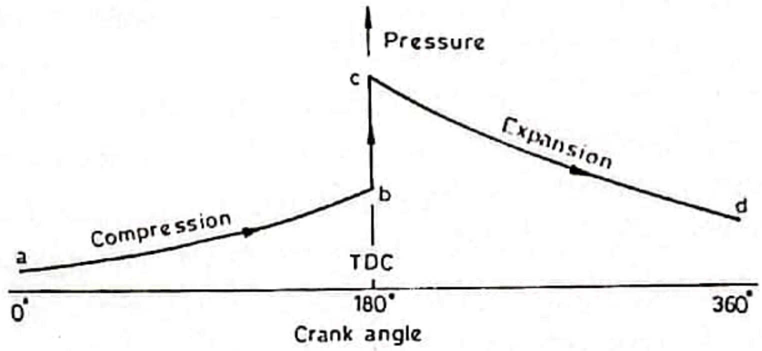


Fig.(9-2a) - Theoretical P-θ diagram.

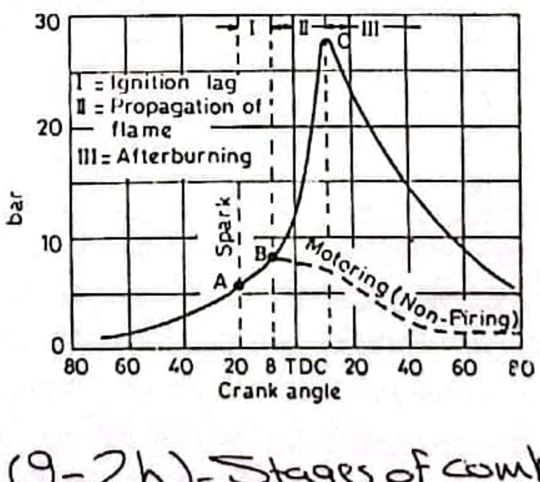


Fig.(9-2b) - Stages of combustion in SI engine.

The former is a chemical process depending upon the nature of the fuel, upon both temperature and pressure, the proportion of exhaust gas, and also upon the temperat. coefficient of the fuel, that is, the relationship between temp. and rate of acceleration of oxidation or burning. The second stage is a mechanical one, pure and simple. The two stages are not entirely distinict, since the nature and velocity of combustion change gradually. The starting point of the second stage is where first measurable rise of pressure can be seen on the indicator diagram, i.e., the point where the line of combustion departs from the compression line. In Fig (9-2b), -A- shows the point of passage of spark (say 20° before TDC); -B- the point at which the first rise of pressure can be detected (say, 8° before TDC) and C the attainment of peak pressure. Thus -AB- represents the first stage (about 20° crank angle rotation) and -BC- the second stage. Although the point C makes the completion of the flame travel, it does not follow that at this point the whole of the heat of the fuel has been liberated, for even after the passage of the flame, some further chemical adjustments due to reassociation, etc., and what is generally refferred to as afterburning, will to a greater or less degree continue throughout the expansion stroke.

The first stage -AB-, is called ignition lag, In spark ignition there is practically no ignition lag and a nucleus of combustion arises instantaneously near the spark plug electrodes. But during the initial period flame front spreads very slowly and the fraction of burnt mixture is small so that an increase of pressure cannot be detected on the indicator diagram. The increase in pressure may be just one per cent of maximum combustion pressure and corresponding to burning of about 1.5 per cent of the working mixture, and the volume occupied by the combustion products may be aboat 5 per cent of the combustion chamber space.

9.4. Effects of Engine Variables on Ignition Lag

The first phase of combustion called ignition lag is not a period of inactivity, but is a chemical process. The ignition lag in terms of crank angle is 10° to 20° and in terms of seconds, 0.0015 second or so. The duration of the ignition lag depends on the following factors:

1. Fuel. The ignition lag depends on the chemical nature of the fuel. The higher the self-ignition temperature of the fuel, the longer the ignition lag.

2. Mixture ratio. The ignition lag is smallest for the mixture ratio which gives the maximum temperature. This mixture ratio is somewhat richer than the stoichiometric ratio

3. Initial temperature and pressure. The rate of chemical reaction depends to a great extent on temperature, the rate being very small at low temperatures but increases rapidly with increase in temperature. The rate of chemical reaction also depends on pressure but to smaller extent. The ignition lag, therefore, decreases with an increase in temperature and pressure of the gas at the time of the spark. Thus, increasing the intake temperature and pressure, increasing the compression ratio and retarding the spark, all reduce the ignition lag.

4. Electrode gap. The electrode gap is important from the point of view of establishment of the nucleus of flame. If the gap is too small, quenching of the flame nucleus may occur and the range of fuel-air ratio for the development of a flame nucleus is reduced. The lower the compression ratio, the higher is the electrode gap required. For a compression ratio of 7 or more a gap of 0.625 mm is satisfactory. The voltage required at the spark plug electrode to produce the spark is found to increase with the decrease in fuel-air ratio and with increase in compression ratio and engine load.

5. Turbulence. Ignition lag is not much affected by the turbulence intensity. Turbulence is directly proportional to engine speed. Therefore, increase in engine speed does not affect much the ignition lag measured in milliseconds. (But as the speed is increased the crankangles in the same milliseconds are increased). Thus measured in degrees of crank rotation the ignition lag increases almost linearly with engine speed. For this reason it becomes necessary to advance the spark timing at the higher speeds.

Excessive turbulence of the mixture in the area of the spark plug is harmful, since it increases the heat transfer from the combustion zone and leads to unstable developement of the nucleus of flame. That is why the spark plug is usually arranged in a small recess in the wall of the combustion chamber.

9.5. Effect of Engine Variables on Flame Propagation

A study of the variables which affect the flame propagation velocity is important because the flame velocity influences the rate of pressure rise in the cylinder. There are several factors which affect the flame speed, the most important being fuel-air ratio and turbulence.

1. Fuel-air ratio. The composition of the working mixture influences the rate of combustion and the amount of heat evolved. With hydrocarbon fuels the maximum flame velocities occur when mixture strength is 110% of the stoichiometric (i.e., about 10% richer than that stoichiometric). When the mixture is made leaner or is enriched, the velocity of flame diminishes. Lean mixture release less thermal energy resulting in lower flame temperature and flame speed. The indicator diagram for rich (maximum power A/F ratio is 12:1), stoichiometric (A/F ratio 14.5:1) and weak (maximum economy A/F ratio 16:1)

2. Compression Ratio. A higher compression ratio increases the pressure and temperature of the working mixture and decreases the concentration of residual gases. These favourable conditions reduce the ignition lag of the combustion and hence less ignition advance is needed. High pressures and temperatures of the compressed mixture also speed up the second phase of combustion. Then the total ignition angle is reduced. Maximum pressure and indicated mean effective pressure are increased.

The increase in compression ratio results in increase in temperature which increases the tendency of the engine to detonate.

3. Intake temperature and pressure. Increase in intake temperature and pressure increases the flame speed.

4. Engine Load. With increase in engine load the cycle pressures increase. Hence the flame speed increases.

5. Turbulence. Turbulence plays a very vital role in the combustion phenomenon. The flame speed is very low in non-turbulent mixtures. A turbulent motion of the mixture intensifies the processes of heat transfer and mixing of the burned and unburned portions in the flame front (diffusion). These two factors cause the velocity of turbulent flame to be increased practically in proportion to the turbulence velocity.

6. Engine Speed. The higher the engine speed, the greater the turbulence inside the cylinder. For this reason the flame speed increases almost linearly with engine speed. Thus if the engine speed is doubled the time required, in milliseconds, for the flame to traverse the combustion space would be halved. Double the original speed and hence half the original time would give the same number of crank degrees for flame propagation. The crank angle required for the flame propagation, which is the main phase of combustion, will remain almost constant at all speeds. This is an important characteristic of spark ignition engines.

7. Engine Size.

Engines of similar design generally run at the same piston speed. This is achieved by smaller engines having larger rpm and larger engines having smaller rpm. Due to the same piston speed, the inlet velocity, the degree of turbulence, and flame speed are nearly same in similar engines, regardless of the size. In small engines the flame travel is small and in large engines is larged. Therefore, if the engine size is doubled the time required (in milliseconds) for propagation of the flame through combustion space will also doubled. But with lower rpm of larger engines the time for the flame propagation in terms of crank angle would be nearly same as in smaller engines.

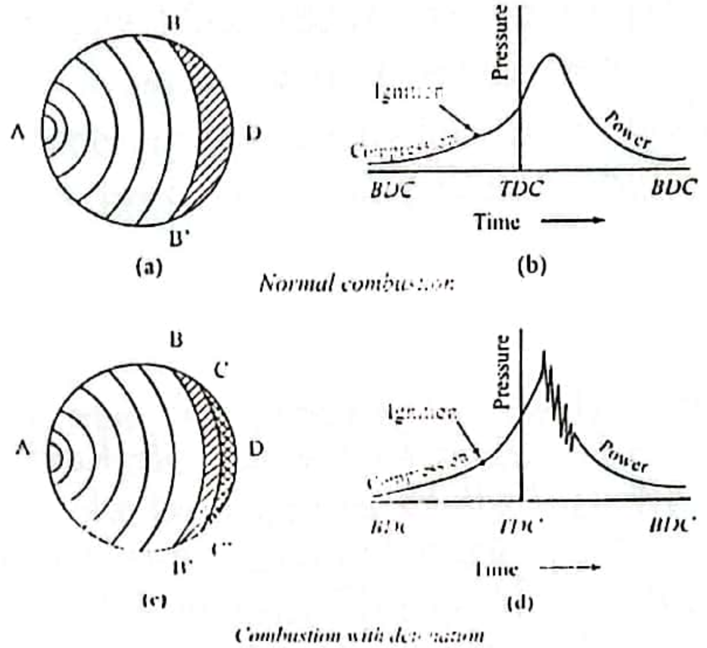
In other words the number of crank degrees required for flame travel will be about the same irrespective of the engine size, provided the engines are similar.

9.6. Abnormal Combustion.

Abnormal combustion means combustion is initiated without an ignition source (i.e., auto-ignition). In normal combustion the flame started by the spark travels across the combustion chamber in a fairly even way. Under certain engine operating conditions "abnormal" combustion may take place which is detrimental to life and performance of the engine. There are a variety of ways in which abnormal combustion can occur. The important abnormal combustions are 'detonation or knock', 'preignition', 'run-on', etc, of these detonation or knock is most important because it puts a limit on the compression ratio at which an engine can be operated, which, in turn, controls the efficiency and to some extent, power output.

9.7. Detonation or Knocking

The normal combustion and detonation or the knocking combustion are shown in Figs. (9 – 3a,b) and (9 – 3c,d), respectively.



Normal and Abnormal Combustion

In normal combustion Fig. (9 – 3a,b), a normal flame front travels across the combustion chamber, from A toward D. The speed of the flame front is about 15 to 30 m/s. As the flame front advances it compresses the unburned charge BB'D, raising its temperature. The temperature is also increased by radiation from the advancing flame front and due to reaction taking place in the unburned mixture itself. If this unburned charge does not reach its critical temperature for auto-ignition, it will not auto-ignite, and the flame front BB' will moves across the unburned charge to the farthest point of the chamber 'D' in the normal manner. The pressure-crank angle diagram is a smooth curve for the normal combustion.

In the abnormal combustion called 'detonation or knocking', the end charge auto-ignites before the flame front reaches it. In order to auto-ignite, the last unburned portion of the charge must reach above a certain 'critical temperature' (which depends upon conditions of pressure and density of the unburned charge) and remain at this temperature for a certain length of time. During this period certain chemical reactions take place which prepare the charge for auto-ignition. The time required in this preparation phase is called 'ignition delay'.

It is clear from the description of the phenomenon of the detonation that the onset of detonation is very much dependent on the properties of the fuel. We have seen that if the unburned charge does not reach its critical temperature there will be no detonation. Further, if the ignition delay period is longer than time required for the flame front to burn through the unburned charge, there again will be no detonation. Only when the critical temperature is reached and maintained, and the ignition delay is shorter than the time it takes for the flame front to burn through the unburned charge, there will be detonation. Hence in order to avoid or suppress detonation, a high auto-ignition temperature and along ignition delay are desirable qualities for SI engine fuels.

9.8. Effects of Detonation

The effects of detonation are as follows:

1. Noise and Roughness.

Mild knock is seldom audible and is not harmful. When the intensity of the knock increases a loud pulsating noise is produced due to the development of a pressure wave which vibrates back and forth across the cylinder. The presence of vibratory motion causes crankshaft vibrations and the engine runs rough.

2. Mechanical Damage.

(a) In most cases of knocking a local and very rapid pressure rise is observed with subsequent waves of large amplitude. This gives rise into increased rate of wear. Erosion of piston crown, in a manner similar to that of marine propeller blades by cavitation, occurs. The cylinder head and valves may also be pitted.

(b) Detonations is very dangerous in engines having high noise level. In small engines the knocking noise is easily detected and the corrective measures can be taken; but in large high duty engines, such as in aero-engines it is difficult to detect knocking noise and hence corrective measures cannot be taken. Hence severe detonation may persist for a long time which may ultimately result in complete wreckage of the piston

3. Carbon deposits.

Detonation results in increased carbon deposits.

4. Increase in Heat Transfer.

Knocking is accompanied by an increase in the rate of heat transfer to the combustion chamber walls. The minor reason is that the maximum temperature in a detonating engine is about 150°C higher than in a non-detonating engine, due to rapid completion of combustion. The major reason for increased heat transfer is the scouring away of protective layer of the inactive stagnant gas on the cylinder walls due to pressure waves. The inactive layer of gas normally reduces the heat transfer by protecting the combustion chamber walls and piston crown from direct contact with the flame.

5. Decrease in power output and efficiency.

Due to increase in the rate of heat transfer the power output as well as efficiency of a detonating engine is decreased.

6. Pre-ignition.

The increase in the rate of heat transfer to the walls has yet another effect. It may cause local overheating, especially of the sparking plug, which may reach a temperature high enough to ignite the charge before the passage of spark, thus causing pre-ignition. An engine detonating for a long period would most probably lead to pre-ignition and this is the real danger of detonation.