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EXPERIMENTAL AND THEORETICAL STUDY TO EVALUATE THE PERFORMANCE OF FOUR STROKE PETROL ENGINE

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Abstract:

Experimental and theoretical study to know the performance of four strokes petrol engine (bore = 66.69 mm, stroke = 49.23 mm) is produced at different engine speed (2000, 2500, 3000 and 3600 rpm) and compression ratio (ε=7.38) which is choiced equal to the compression ratio of the test engine to ease the comparison. The study consists of torque, brake power, specific fuel consumption, and volumetric efficiency and Air/Fuel ratio. The comparison between experimental and theoretical studies gave good agreement.

الخلاصة :

تم دراسة أداء محرك بترول رباعي الأشواط (قطر= 66.69 ملم وطول الشوط = 49.23 ملم) عمليا و نظريا عند سرع مختلفة (2000,2500,3000,3500 دورة/دقيقة) و نسبة انضغاط 7.38 وهي مساوية إلى نسبة أنضغاط الجهاز المستخدم في القياس. الدراسة تضمنت احتساب العزم,القدرة,معدل استهلاك الوقود,النسبة الحجمية و نسبة الهواء/الوقود).المقارنة بين النتائج العملية و النظرية أعطت توافق جيد.

Nomenclatures:

Symbol	Meaning	Units
A	Cross sectional area	m^2
BP	Brake power	kW
C_{dt}	Discharge coefficient of venturi throat	-
C_{dct}	Discharge coefficient of capillary tube	-
C_v	Specific heat at constant volume	kJ/kg.K
E	Internal energy	kJ/kg
M	Number of moles	-
m_d	Mass flow rate	Kg/sec.
N	Engine speed	rpm
n	Number of cycle	-
Q	Heat amount	kJ
P	Pressure	Kpa
PO	Power	kW
R_{mol}	Universal gas constant	kJ/kg.K
TO	Torque	N.m.
V	Volume	m^3
W	Work	kJ
Greek symbol		
Ω	Constant	-
$\prod$	Constant	-
γ	Specific heat ratio	-
$_v\eta$	Volumetric efficiency	-
ε	Compression ratio	-
$_r\upsilon$	Residual gas coefficient	-
ρ	Density	Kg/m^3
Φ	Equivalent ratio	-
Δ	Difference	-
φ	Additional flow coefficient	-

ω	Angular velocity	1/sec
π	Constant ratio 22/7	-
Subscript		
a	Air	
f	Fuel	
o	Atmosphere	
R	Reactants	
P	Products	

Introduction:

Internal-Combustion Engine, any type of machine that obtains mechanical energy directly from the expenditure of the chemical energy of fuel burned in a combustion chamber that is an integral part of the engine. Four principal types of internal-combustion engines are in general use: the Otto-cycle engine, the diesel engine, the rotary engine, and the gas turbine (Fernando 1998). The Otto-cycle engine, named after its inventor, the German technician Nikolaus August Otto, is the familiar gasoline engine used in automobiles and airplanes; the diesel engine, named after the French-born German engineer Rudolf Christian Karl Diesel, operates on a different principle and usually uses oil as a fuel. It is employed in electric-generating and marine-power plants, in trucks and buses, and in some automobiles. Both Otto-cycle and diesel engines are manufactured in two-stroke and four-stroke cycle models. The ordinary Otto-cycle engine is a four-stroke engine; that is, in a complete power cycle, its pistons make four strokes, two toward the head (closed head) of the cylinder and two away from the head. During the first stroke of the cycle, the piston moves away from the cylinder head while simultaneously the intake valve is opened. The motion of the piston during this stroke sucks a quantity of a fuel and air mixture into the combustion chamber. During the next stroke, the piston moves toward the cylinder head and compresses the fuel mixture in the combustion chamber. At the moment when the piston reaches the end of this stroke and the volume of the combustion chamber is at a minimum, the fuel mixture is ignited by the spark plug and burns, expanding and exerting a pressure on the piston, which is then driven away from the cylinder head in the third stroke. During the final stroke, the exhaust valve is opened and the piston moves toward the cylinder head, driving the exhaust gases out of the combustion chamber and leaving the cylinder ready to repeat the cycle.

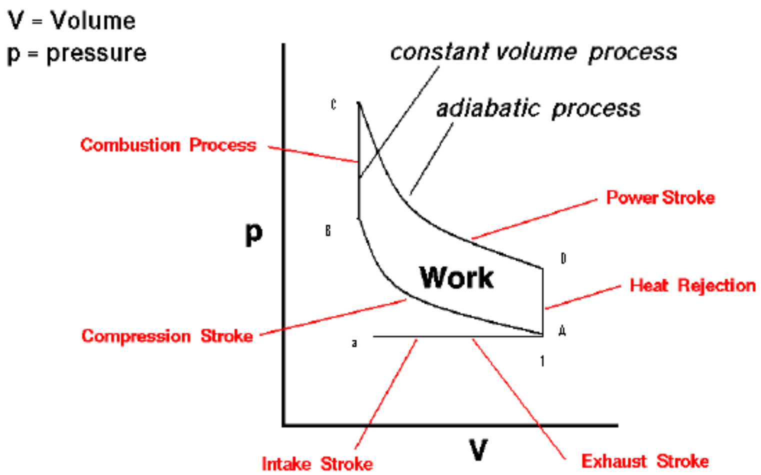

Fig.(1): P-V diagram for Otto cycle

THEORETICAL MODEL:

Air/Fuel Cycle:

The ideal Otto cycle with hydrocarbon –air mixture differs from the ideal cycle (constant volume cycle) in a number of respects. the most important are the variation in gas composition during combustion and expansion and the variation of specific heats with temperature and composition, since in order to satisfy thermodynamic equilibrium dissociation takes place in the combustion process and re-association in the expansion process. Normal fuels in spark ignition engines are mixture of hydrocarbon. We can represent these fuels by a general hydrocarbon C_nH_m (in this paper, Petrol C_8H_{18} is used as a fuel) Hence the basic stoichiometric equation for a hydrocarbon-air reaction is

$$C_nH_m + (n+\frac{m}{4})O_2 + 3.76(n+\frac{m}{4})N_2 = n\,CO_2 + \frac{m}{2}H_2O + 3.76(n+\frac{m}{4})N_2$$

The stoichiometric equation defines the correct mixture .To allow for mixture different from the correct mixture we introduce the equivalence Φ. This is defined as the ratio of actual fuel/air ratio to the stoichiometric fuel/air ratio.

The stoichiometric fuel/air ratio is

$$\frac{C_nH_m}{(n+\frac{m}{4})O_2+3.76(n+\frac{m}{4})N_2} \qquad (1)$$

Hence for a mixture of equivalence Φ, we have

$$C_nH_m + (n+\frac{m}{4})\frac{1}{\Phi}O_2 + 3.76(n+\frac{m}{4})\frac{1}{\Phi}N_2.$$

if the equivalence Φ is greater than unity, the mixture is said to be rich ,and if Φ is less than unity, the mixture is said lean. Air/Fuel cycle is divided into four strokes :

a- adiabatic compression

Consider a small change in volume from V1 to V2 (see Fig.1). The first law for the change is dQ-dW= dE

and since the process is adiabatic, dQ=0,we have dE+dW=0

for a change in pressure from P1 to P2 the work dW is approximately

$$dW = P\,dV = (\frac{p_1+p_2}{2})(V_2-V_1). \qquad (2)$$

The internal energy change is dE, which is given by

$$dE = E_2 - E_1. \qquad (3)$$

The internal energy of the mixture is a function of the composition of the cylinder contents and temperature. The internal energy is therefore the internal energy of the reactants in the hydrocarbon-air combustion process. The composition is considered to be constant during the compression stroke. $E_R=E_{OR}+E_R(T)$.

$(e_o)_{CNHM}$,$(e_o)_{O2}$,$(e_o)_{N2}$ are the specific internal energies at absolute zero for C_NH_M,O_2 and N_2, then the internal energy for the reactants at absolute zero E_{OR} is

$$E_{OR} = W((e_o)C_nH_m + (n+\frac{m}{4})\frac{1}{\Phi}(e_o)O_2 + 3.76(n+\frac{m}{4})\frac{1}{\Phi}(e_o)N_2). \qquad (4)$$

If the specific internal energies, relative to absolute zero, are $e(T)_{CNHM}$, $e(T)_{O2}$ and $e(T)_{N2}$,for C_nH_m , O_2 ,and ,N_2 ,then $E_R(T)$ for the mixture is given by

$$E_{R(T)} = W(e(T)C_nH_m + (n+\frac{m}{4})\frac{1}{\Phi}e(T)O_2 + 3.76(n+\frac{m}{4})\frac{1}{\Phi}e(T)N_2). \qquad (5)$$

The internal energy $E_R(T)$ can be calculated from the polynomial coefficients given in (Benson *et.al.*1979) numerical procedure which enables the internal energy for mixture to be evaluated is given .the internal energy change E_2-E_1 is then

$$E_2-E_1=(E_R)_2-(E_R)_1=(E_{OR})_2-(E_{OR})_1+E_R(T_2)-E_R(T_1)$$

or

$$E_2-E_1= E_R(T_2)- E_R(T_1) \qquad (6)$$

Since for a mixture of constant composition $(E_{OR})_2= (E_{OR})_1$.

For a change in state from V_1 to V_2 the first law is thus $E_2-E_1+dW=0$

Substituting for E_2-E_1 and dW we obtain

$$E_R(T_2)- E_R(T_1) +(\frac{p_1+p_2}{2}) (V_2-V_1)=0. \quad (7)$$

The internal energy term $E_R(T_2)$ depends on T_2, therefore in expression there are two unknown P_2 and T_2; P_1, T_1 and $E_R(T_1)$ are known at the beginning of the step. These are related by the state equation

$$P_2=P_1(\frac{V_1}{V_2})(\frac{T_2}{T_1}). \quad (8)$$

To solve for P_2 and T_2 we use the Newton-Raphson method

$$F(E)= E_R(T_2)- E_R(T_1)+ (\frac{p_1+p_2}{2}) (V_2-V_1) \quad (9)$$

And

$$F^{-}(E)=\frac{dF(E)}{dT}=\frac{d(E_R(T_2))}{dT}=M_RC_V(T_2) \quad (10)$$

Since from

$$\frac{d(E_R(T_2))}{dT}=M_RC_V(T_2) \quad (11)$$

and $\frac{dE_R(T1)}{dT}=0$

In the expansion we have assumed that the rate of change of work W with temperature T over the volume change is negligible and $\frac{dW}{dT}=0$.

Solution of equation is by Newton-Raphson. If $(T_2)_{n-1}$ is the estimated value of T_2,then

$$(T2)_N=(T2)_{n-1}-\frac{f(E)_{n-1}}{f'(E)_{n-1}} \quad (12)$$

the first estimate of T_2 can be obtained from the expression

$$T_2=T_1(\frac{V_1}{V_2})^{K-1} = T_1(\frac{V_1}{V_2})^{\frac{R_{MOL}}{C_V(T1)}} \quad (13)$$

b- adiabatic combustion at constant volume

This period corresponds to B-C path in the pressure and volume diagrams. In this simple analysis we shall assume the products of combustion to be carbon dioxide (CO_2), carbon monoxide (CO), water vapor (H_2O), hydrogen (H_2), oxygen (O_2) and nitrogen (N_2) . For the combustion of one mol of C_nH_m in air we shall have the following products:

$$a_1 CO_2+a_2 CO+a_3 H_2O+a_4 H_2+a_5 O_2+a_6 N_2 \quad (14)$$

where a_1 , a_2 , a_3 , a_4 , a_5 ,and a_6 are the number of mols of CO_2,CO,H_2O,H_2,O_2 and N_2, respectively per mol of C_nH_m.

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

$$W(C_nH_m+(n+\frac{m}{4})\frac{1}{\Phi}O_2+3.76(n+\frac{m}{4})\frac{1}{\Phi}N_2)$$

$$W(a_1 CO_2+a_2 CO+a_3 H_2O+a_4 H_2+a_5 O_2+a_6 N_2). \quad (15)$$

to determine the number of mols a_1 to a_5 (see Benson *et.al.*1979)

dQ - dW = dE.

For adiabatic combustion dQ=0 and for constant volume combustion $\Delta W=0$.It follows therefore that for the process from point B to point C,

$$dE = E_C - E_B = 0. \quad (16)$$

At the state B the internal energy is

$$E_B = E_R = E_{OR} + E_R(T_B). \quad (17)$$

At the state C the internal energy is

$$E_C = E_P = E_{OP} + E_P(T_C). \quad (18)$$

We can express the internal energies in terms of the composition of the mixtures at point B and point C and the specific internal energies, can obtain:

$$E_{OR}= w((e_o)C_nH_m+(n+\frac{m}{4})\frac{1}{\Phi}(e_o)O_2+3.76(n+\frac{m}{4})\frac{1}{\Phi}(e_o)N_2)) \quad (19)$$

$$E_R(TB)=w(e(TB)C_nH_m+(n+\frac{m}{4})\frac{1}{\Phi}e(TB)O_2+3.76(n+\frac{m}{4})\frac{1}{\Phi}e(T_B)N_2) \quad (20)$$

$$E_{OP}=w(a_1(e_o)CO_2+a_2(e_o)CO+a_3(e_o)H_2O+a_4(e_o)H_2+a_5(e_o)O_2+a_6(e_o)N_2). \quad (21)$$

$$E_P(TC)=w(a_1\ e(T_C)CO_2+\ a_2\ e(T_C)CO+\ a_3\ e(T_C)H_2O+\ a_4\ e(TC)H_2+\ a_5\ e(T_C)\ O_2+\ a_6\ e(T_C)N_2). \quad (22)$$

If we substitute for E_B and E_C into Eq.(16) we have

$$F(E)=(E_{OP}+E_P(TC))-(E_{OR}+E_R(T_B))=0. \quad (22)$$

If we assume that $dE_{OP}/dT=0$ we can write that

$$\frac{df(E)}{dT}=\frac{dE_P(T_c)}{dT}=\bar{E}_P(T_c) \quad (23)$$

and

$$E_P(T_C)M_CC_V(T_C)=w\ M_PC_V(T_C), \quad (24)$$

Where

$M_P=a_1+a_2+a_3+a_4+a_5+a_6$.

Equation (22) can then be solved numerically by Newton-raphson's method to calculate T_c

The first estimate for T_C can be obtained by the approximate expressions due to Annand (Benson *et.al.*1979), estimation of T_c

For $\Phi < 1.0$ $T_C=T_B+2500\ \Phi$.

For $\Phi > 1.0$ $T_C=T_B+2500\ \Phi-700(\Phi-1)$.

The maximum temperature in the cycle corresponds to T_C. This is dependent on the air/fuel ratio, the initial temperature and pressure at the commencement of combustion and the fuel.

c- Adiabatic Expansion

This corresponds to the period from C to D in the pressure and volume diagram.

A though during combustion the maximum temperature is reached the chemical reactions continue to take place during the expansion stroke; this is because the chemical species are in thermodynamic equilibrium. The composition of the gas mixture will therefore vary with pressure and temperature. To satisfy thermodynamic equilibrium the chemical reactions for water gas and carbon dioxide, namely

$$CO+H_2O \rightleftarrows H_2+CO_2.$$

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

if we consider a step from V_1 to V_2 ,then the pressure and temperature will change from P_1,T_1 to P_2,T_2 .

These will be related by

$P_1V_1=M_1R_{mol}\ T_1$ and $P_2V_2= M_1R_{mol}\ T_1$

And the number of mols at 1 and 2 will be

$M_1= w(a_1+a_2+a_3+a_4+a_5+a_6)T_1$,

$M_2= w(a_1+a_2+a_3+a_4+a_5+a_6)T_2$.

to obtain T_2 we apply the first law of thermodynamic,

$\Delta Q\ -\Delta W\ =\ \Delta E$

For an adiabatic process $\Delta Q=0$ and for a small step V_1 to V_2 the work done is approximately

$$\Delta W\ =\ Pdv\ =\frac{P_1+P_2}{2}(V_2-V_1). \quad (25)$$

The change in internal energy dE is

$\Delta E = E_2 - E_1$,

where, as before,

$E = E_P = E_{OP} + E_P(T).$ (26)

For temperature T_1 the internal energies are:

$E_{OPI} = w(a_1(eo)_{CO2} + a_2(eo)_{CO} + a_3(eo)_{H2O} + a_4(eo)_{H2} + a_5(eo)_{O2} + a_6(eo)_{N2})$ (27)

$E_P(T_1) = w(a_1e(T_1)_{CO2} + a_2e(T_1)_{CO} + a_3e(T_1)_{H2O} + a_4e(T_1)_{H2} + a_5e(T_1)_{O2} + a_6e(T_1)_{N2})$ (28)

And at T_2:

$E_{OP} = w(a_1(e_o)_{CO2} + a_2(e_o)_{CO} + a_3(e_o)_{H2O} + a_4(e_o)_{H2} + a_5(e_o)_{O2} + a_6(e_o)_{N2})$ (29)

$E_P(T_2) = w(a_1e(T_2)_{CO2} + a_2e(T_2)_{CO} + a_3e(T_2)_{H2O} + a_4e(T_2)_{H2} + a_5e(T_2)_{O2} + a_6e(T_2)_{N2})$ (30)

The number of mols a_1 to a_6 being different at each temperature. At the initial temperature T1 the internal energies E_{OPI} and $E_P(T_1)$ are fixed; however E_{OP2} will not necessarily be equal to E_{OPI} due to the change in composition of the mixture. The first law becomes

$$F(E) = (E_{OP2} + E_P(T_2)) - (E_{OP1} + E_P T_I)) + \left(\frac{P_1 + P_2}{2}\right)(V_2 - V_1) = 0, \quad (31)$$

$$\frac{df(E)}{dT} = \frac{dE_P(T2)}{dT} = E'_P(T_2) = M_2 C_V(T_2). \quad (32)$$

We can solve expression (31)numerically by Newton-Raphson to calculate T_2

the first estimate for T_2 is

$$T2 = T1\left(\frac{V_1}{V_2}\right)^{\frac{Rmol}{Cv(T1)}} \quad (33)$$

And the pressure ratio is

$$\frac{P_2}{P_1} = \frac{M_2}{M_1}\,\frac{V_1}{V_2}\,\frac{T_2}{T_1}. \quad (34)$$

Notice the calculation involves first estimating T_2 and calculating the composition of the mixture followed by the check using the first law of thermodynamics. To close the cycle we join P_1 to P_4 at constant volume.

Air/Fuel Ratio

From gas dynamics (Willard 1997), the air flow through a venturi throat can be written:

$md_a = (C_{dt}A_tP_0/(RT)^{0.5})(P_t/P_0)^{(1/\gamma)}\{(2\gamma/(\gamma-1))(1-(P_t/P_0)^{(\gamma-1/\gamma)})\}^{0.5}$ (35)

The pressure differential in the air will be

$\Delta P_a = P_0 - P_t$ (36)

Pressure differential through the fuel capillary tube will be

$\Delta P_f = \Delta P_a - \rho_f\, g\, h$ (37)

The second term in eq.(37) is the hydraulic head between the fuel reservoir and throat . The elevation h is built into a carburetor to avoid fuel leaking out when the vehicle is parked on slop. Values of h are typically rounded 1 to 2 cm.

Liquid fuel flow through a capillary tube is:

$md_f = C_{dc}A_c(2\rho_f\, \Delta P_f)^{0.5}$ (38)

Using Eqs.(35-38), the air-fuel ratio supplied by the carburetor can be obtained (Heywood 1988):

$A/F = md_a/md_f = (C_{dt}/C_{dc})(A_t/A_c)(\rho_{a/}\,\rho_f)^{0.5}\Omega\Pi$ (39)

With: $\Omega = \{\Delta P_a/\Delta P_f\}^{0.5}$

$\Pi = \{[\gamma/(\gamma-1)][(P_t/P_0)^{(2/\gamma)} - (P_t/P_0)^{(\gamma+1/\gamma)}]/[1-(P_t/P_0)]\}^{0.5}$

If the air velocity through the carburetor throat is increased by increasing the engine speed, a maximum flow rate will be reached when sonic velocity occurs. This will happen when

$P_t/P_0 = [2/(\gamma+1)]^{\gamma/(\gamma-1)}$ (40)

But

$md_a = \eta_v \rho_a V_z N_s / n$ (41)

And after some simplifications by substitute eq.(40) into eq. (35), it gives

$md_a = 236.5 A_t C_{dt}$ (42)

A_t can be calculated as a function of engine speed, the A/F ratio is calculated at any engine speed.

Specific fuel consumption:

The following expression may be used to calculate specific fuel consumption

$md_{fi} = md_{fh} *1000 / PO_e$ g/kw.hr. (43)

Where: md_{fh} = 3600*((number of fuel moles * molecular weight of the fuel)/time of one cycle)

Time of one cycle = n / N_s (n = number of revolutions per one cycle)

PO_e (effective power) = To * ω

To: torque (N.m)

$\omega = 2\pi N/60$

The volumetric efficiency:

The volumetric efficiency is calculated from the below equation

$\eta_v = \varphi (\varepsilon / (\varepsilon-1))(T_a /T_1)(P_1/P_a)(1/(1+\upsilon_r))$ (44)

Where: φ =additional charge coefficient (φ = 1.02 – 1.07)

ε =compression ratio

T_a & P_a =are atmospheric temperature and pressure respectively

T_1 & P_1 =are the temperature and pressure at the beginning of the compression stroke respectively

$T_1 = (T_a+ \upsilon_r T_r) / (1+ \upsilon_r)$ (45)

$P_1 = P_a (1-c N^2)$ (46)

υ_r = residual gases coefficient (in P.E. υ_r = 0.06 – 0.1)

c = constant = $(\rho_a/2 P_a)(A_pS/30A_v)^2$ (47)

T_r = residual gases temperature (in P.E. T_r = 900 – 1100 K)

A_p= cross section area of the piston

A_v= cross section area of the intake valve $=1.3B^2(\bar{U}_P)_{max}/c_i$ (48)

B= diameter of the piston

$(\bar{U}_P)_{max}$= average piston speed at maximum engine speed = SN/30

S = stroke length

c_i = speed of sound at inlet conditions = $(\gamma RT)^{0.5}$

Z = number of the strokes

Mean indicated pressure and indicated power:

P_i = work done of the cycle / stroke volume (kpa) (49)

$PO_i = P_i V_z N / 60Z$ (kW) (50)

Where: P_i = indicated pressure

PO_i = indicated power

EXPERIMENTAL WORK:

Description of Engine Test:

A single cylinder, four-stroke petrol engine, which provides a practical introduction to internal combustion engine operation and testing (see Fig.2). The engine includes a mounting kit for fixing it to the test bed, and easy-fit flexible coupling to connect it to the test bed dynamometer. Transducers in the engine and on the test bed connect to the instrumentation unit to measure the engines performance characteristics. The cylinder pressure transducer and the engine cycle analyzer are available separately.

Fig.(2):Engine test

Specification of Engine Test:

Dimensions: net 370mm x 570mm x 500mm
Stroke: 49.23mm
Bore: 66.69mm
Nominal output: 3.73 kW at 3600 rpm
Ignition: electronic

Parts of Measuring Instruments:

1-Torquemeter: is used to measure the torque (N.m.) of the engine.
2-Tachometer: is used to know the value of engine speed (rpm)
3-Exhaust Temperature Meter: is used to measure temperature of the exhaust gases directly in (^{0}C).
4-Air Flow Manometer: is used to measure the value of (h) in (mmH2O).
5-Measurment of specific fuel consumption : there is a glass tube which is divided into three parts to know the volume of fuel (8ml, 16ml and 32ml) and there is too timing watch to measure the time of the fuel consumption from the glass tube (in second)

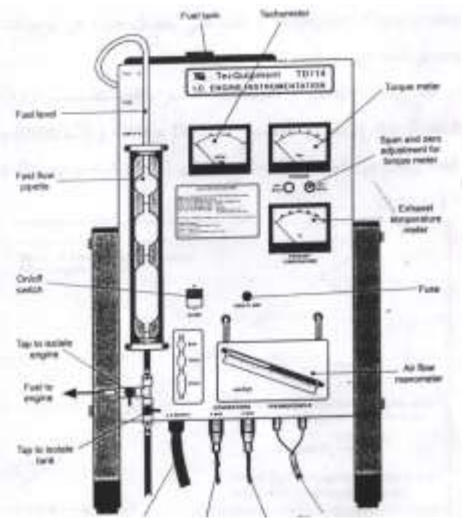

Fig.(3): Instrumentation unit

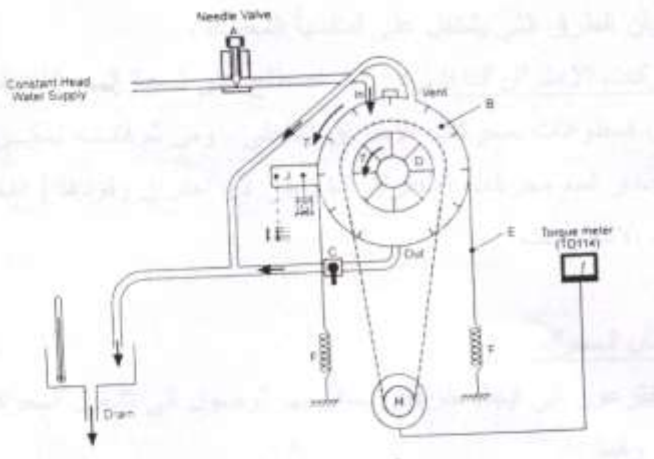

Fig.(4): Hydraulic dynamometer

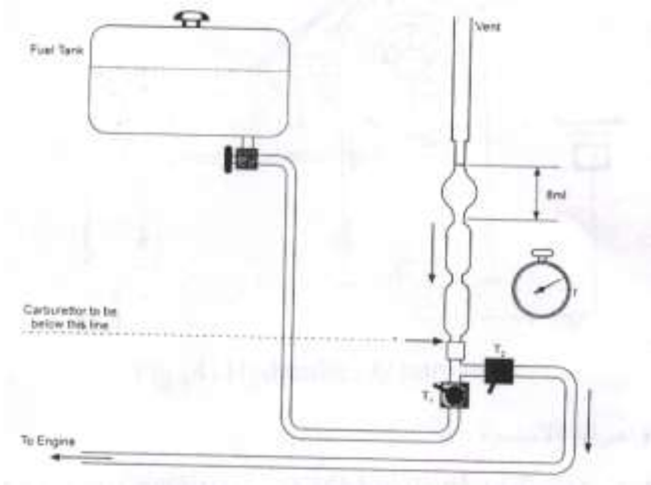

Fig.(5): The fuel system

Steps of Experimental Work:

1- Work the engine at 2000 rpm by control with the throttle opening and measuring the torque, exhaust temperature, volume of consumed fuel, time of certain volume of consumed fuel, h(mm.H2O) from air flow meter.

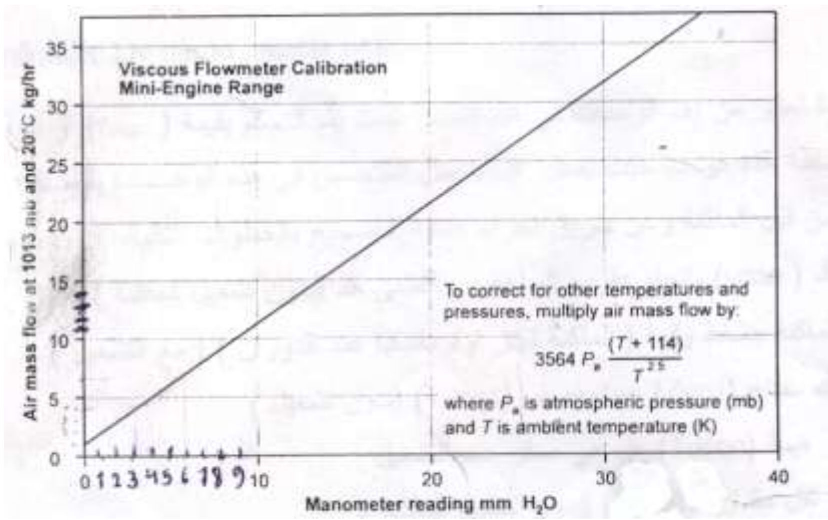

Fig.(6): Viscous flow meter calibration

2- Then getting the value of air flow rate (kg/sec.) from the Fig.6 after calibration with the operating temperature and pressure.

3- Calculate the mass fuel consumption per hour from below equation which related with the measuring values in point 5 in (2-1-2)

m_{fh} (kg/hr.)= ρ_f (kg/m^3)* (V_f (ml)*10^{-6})*3600 / time (in sec) (51)

4- Calculate the brake power (kW)

BP (kW) = To (N.M.)ω/1000 (52)

Where ω=2*π*N / 60

N= rpm, π=22/7

5-evaluate (Air / Fuel) ratio

Air/Fuel = m_{ah} (kg/hr from the Fig.(6)) / m_{fh} (kg/hr from the eq.(35))

6- Calculate the volumetric efficiency

η_v = (m_{ah}/3600) / (ρ_{air} V_z N/(60*n)) (53)

$\rho_{air} = (P_a*1000 / (R*(T_a+273))$ (kg/m^3) (54)

$V_z = (\pi/4)*bore^2*stroke$ (m$^{3)}$ (55)

n= number of strokes/2 (no. of revolution/cycle) (56)

Where: ρ_{air} :air density

P_a, T_a: ambient air pressure (kpa) and temperature (^{0}C) respectively

R= 287 kj/kg.k

V_z: stroke volume (m^3)

7-return the steps (1-6) at 2500, 3000 and 3600 rpm

Result and Discussion:

Both torque and brake powers are functions of engine speed. At low speed, torque increases as engine speed increases. When engine speed increases further, torque reaches a maximum and the decreases as shown in Fig.7. Torque decreases because the engine is unable to ingest a full charge of air at higher speeds. While Fig.8 shows the direct relation between the engine speed and the brake power because of increasing the speed lead to increase the pressure differential between the atmosphere and cylinder pressure at beginning of intake stroke then increasing the mass of fresh charge and the energy release. The small variation between experimental and theoretical values due to the losses of fresh charge by evaporation and leakage and unburned hydrocarbon inside the combustion chamber. Fig.9 represents the relation between the (Air/Fuel) ratio as a function of engine speed. As engine speed increases, (Air/Fuel) ratio increases too because the increasing of engine speed will create high pressure drop according Bernoulli’s equation then the mass flow rate of air and fuel increase but the response of air flow to the pressure drop grater than the fuel response because the specific gravity and friction losses in passageways of fuel. The viscous flow friction that affects the air as it passes through the air filter, carburetor, throttle plate, intake system. Viscous drag, which causes the pressure loss, increases with the square of flow velocity. This results in decreasing the efficiency on high speed such as in Fig.10 Much development work has been done to reduce pressure losses in air intake system. Smooth walls in the intake manifold, the avoidance of sharp corners and bends, elimination of the carburetor, and close fitting parts with no gasket protrusions all contribute to decreasing intake pressure loss (Willard 1997). One of the greatest flow restrictions is the flow through the intake valve. To reduce this restriction, the intake valve flow area has been increased by building multi valve engine having two or even three in take valves per cycle. Fig.11 explains the brake specific fuel consumption as a function of engine speed. Fuel consumption decreases as engine speed increases due to the shorter time for heat loss during each cycle. At higher engine speeds, fuel consumption again increases because of high friction losses. The small difference between the experimental values of all parameters and the theoretical one is due to the several losses such as leakage, evaporation, thermal and mechanical losses.

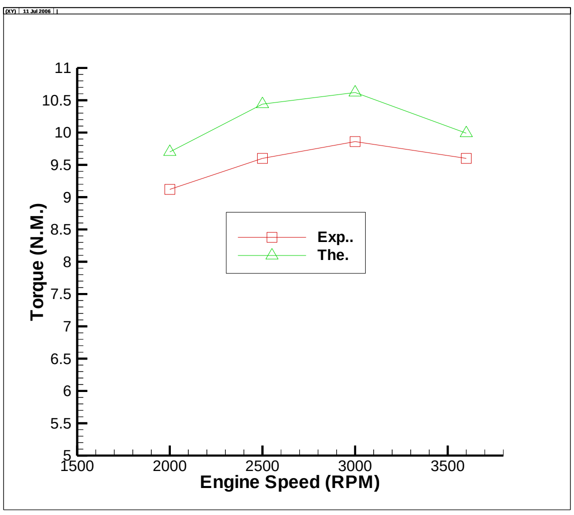

Fig.(7): Effect of a variable engine speed on the torque of engine

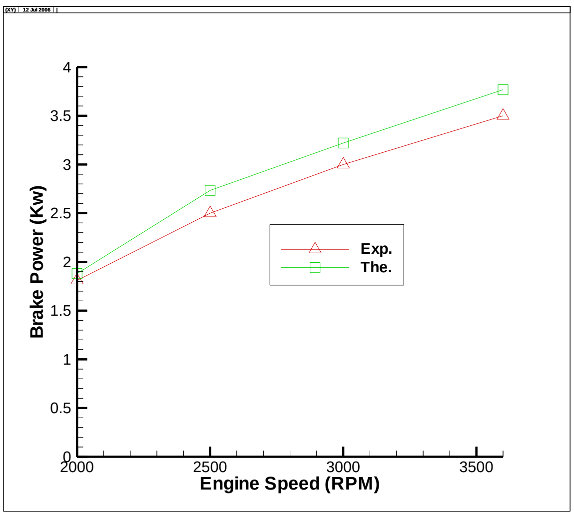

Fig.(8): Effect of a variable engine speed on the engine brake power

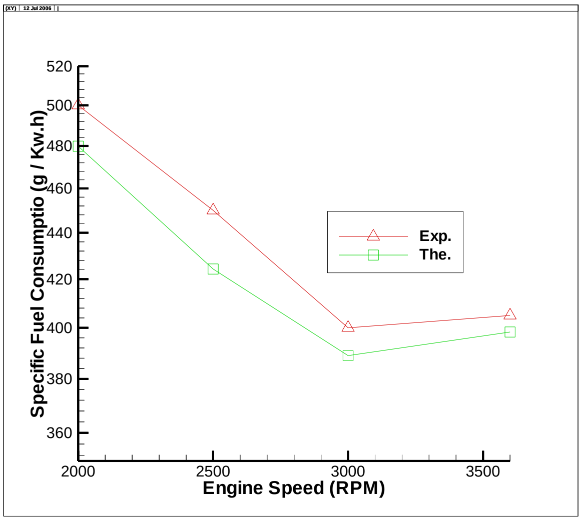

Fig.(9): Effect of a variable engine speed on the SFC

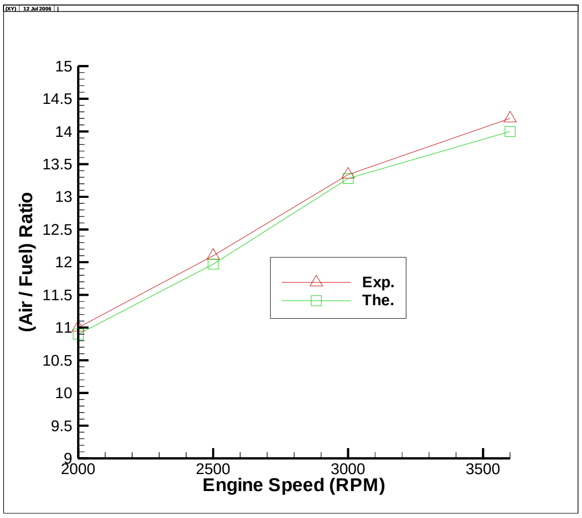

Fig.(10): Effect of a variable engine speed on the (Air/Fuel) ratio

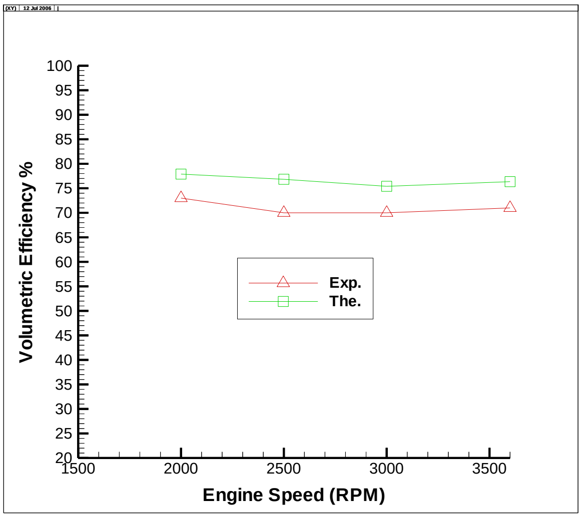

Fig.(11): Effect of a variable engine speed on the volumetric efficiency

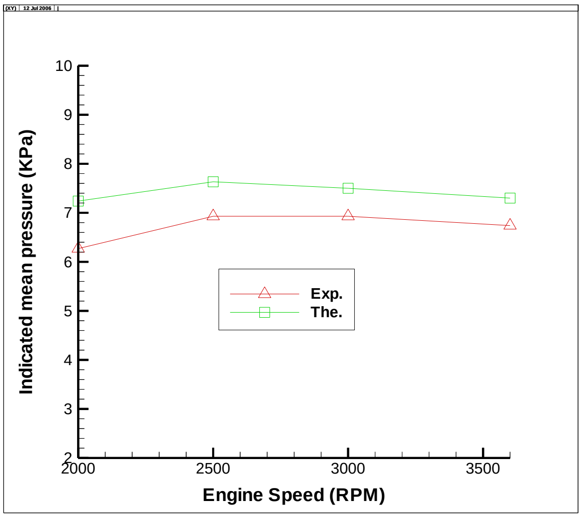

Fig.(12): Effect of a variable engine speed on the mean indicated pressure

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