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# PERFORMANCE CHARACTERISTICS OF ENGINE UNDER CONTROLLED TEMPERATURE AND VARIAABLE OPERATING PARAMETERS

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***Abstract: In the present scenario the S.I. engines being used in automotives by various manufactures are not properly suitable to out climatic condition. As our country is among tropical countries where the variation in the temperature is having very vast range i. e. from 0Ċ to 48Ċ in various regions of the country. Looking in to this vast varying temperature range it is very difficult to say that which temperature is most suited to operating condition of engines and gives us best performance level as for as sfc and brake power is concerned. In my work I have tried to investigate the best option to run the s.i engine and simultaneously to maintain the emission norms. Today research and development in the field of gasoline engines have to face a double challenge: on the one hand, fuel consumption has to be reduced, while on the other hand, ever more stringent emission standards have to be fulfilled. The development of engines with its complexity of in-cylinder processes requires modern development tools to exploit the full potential in order to reduce fuel consumption. There are many strategies for improving fuel economy and reducing exhaust emission. Hydrocarbon emissions (HC) and carbon monoxide (CO). A single cylinder, four stroke, petrol carburetor engine connected to rope brake type dynamometer for loading was adopted to study engine power, fuel economy, engine exhaust emissions of hydrocarbon, oxides of nitrogen in the exhaust. The performance results that are reported include brake power and specific fuel consumption (sfc) as a function of engine temperature; i.e. 50, 60, 70 and 80ºC with varying engine speed of 1500, 2000, 2500, rpm.***

## I. INTRODUCTION

*1.1 Introduction:*

The internal combustion engine is the key to the modern society. Without the transportation performed by the millions of vehicles on road and at sea we would not have reached the living standard of today. We have two types of internal combustion engines, the spark ignition, SI, and the compression ignition, CI. Both have their merits. The SI engine is a rather simple product and hence has a lower first cost. The problem with the SI engine is the poor part load efficiency due to large losses during gas exchange and low combustion and thermodynamics efficiency. The CI engine is much more fuel efficient and hence the natural choice in applications where fuel cost is more important than first cost. The problem with the CI engine is the emissions of nitrogen oxides. As the environmental problems caused by vehicle exhaust emissions become more severe, exhaust gas emission regulations and fuel economy standards become more stringent.

*1.2 Engine performance parameter:*

- Specific fuel consumption: Specific fuel consumption is defined as the amount of fuel consumed per unit of power developed per hour. It is clear indication of the efficiency with which the engine develops power from fuel. Brake specific fuel consumption (bsfc) is determined on the basis of brake output of the engine while indicated specific fuel consumption (isfc) is determined on the basis of indicated output of the engine.
- Brake power: The power developed by en engine at the output shaft is called the brake power (B.P.). The measurement of power involves the measurement of force (or torque) as well as speed. The power is done with the help of a dynamometer and torque is by a tachometer or by some other suitable device.

*1.3 Combustion:*

Combustion may be defined as a relatively rapid 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:

- The presence of combustible mixture
- Some means of initiation combustion
- Stabilization and propagation of flame in the combustion chamber

A chemical equation for combustion of any hydrocarbon can be easily written.

For $C_{18}H_{18}$ (iso octane) the equation is

$$C_{18}H_{18} + 12.5O_2 = 8CO_2 + 9H_2O$$

1.4 Stages of combustion in S.I. Engine:

A typical theoretical pressure crank angle diagram during the process of compression (a→ b) combustion (b → c) and expansion (c→ d) in an ideal four stroke spark ignition engine is shown in fig. 1.1. In an ideal engine, as can be seen form the diagram the entire pressure rise during combustion takes place at constant volume i.e., at TDC However in an actual engine this does not happen. The detailed process of combustion in an actual S.I. engine is described below. Sir Richardo, known as the father of engine research, describes the combustion process in a SI engine as consisting of three stages:

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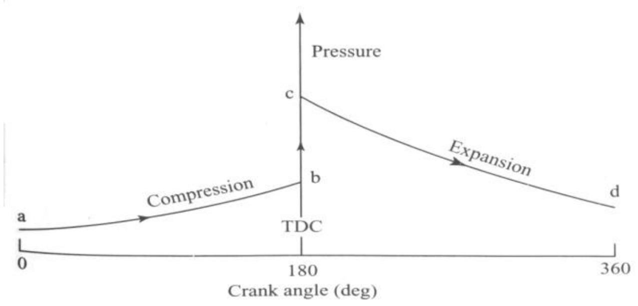


Fig 1.1 Theoretical pressure crank angle diagrams

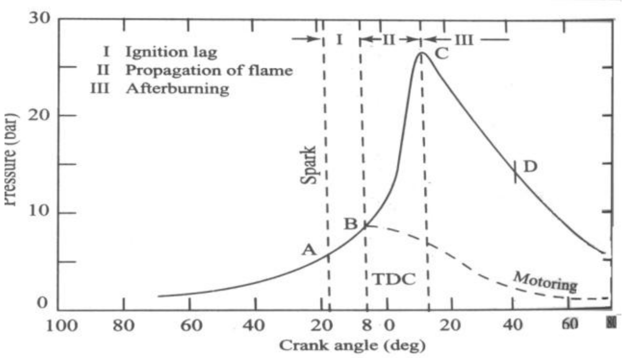


Fig.1.2 Stages of Combustion in an S.I. Engine

The first State (A→ B) in referred to as the ignition lag or preparation in which growth and development of a self propagating nucleus of flame takes place. This is a chemical process depending upon both temperature and pressure, the nature of the fuel and the proportion of the exhaust residual gas. Further, it also depends upon the relationship between the temperature and the rate of reaction.

The starting point of the third stage is usually taken as the instant at which the maximum pressure is reached on the indicator diagram (point C). The flame velocity decrease during this stage. The rate of combustion becomes low due to lower flame velocity decrease during this stage. The rate of combustion becomes low due to lower flame velocity and reduced flame front surface. Since the expansion stroke starts before this stage of combustion with the piston moving away form the top dead centre, there can be no pressure rise during this stage.

*1.5 Effect 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 $10^0$ to $20^0$ and in terms of seconds, 0.0015 second or so. The duration of the ignition lag depends on the following factors:

- 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.
- Mixture ratio: The ignition lag is smallest for the mixture ration which gives the maximum temperature. This mixture ratio is somewhat richer than the stoichiometric ratio.
- Initial temperature and pressure: The rate of chemical reaction depends to a great extent on temperature, the rate being very small at low temperature but increases rapidly with increase in temperature. The rate of chemical reaction also depends on pressure but to smaller extent. The ignition lag, therefore, decrease with an increase in the temperature and pressure of the gas at the spark. Thus, increasing the intake temperature and pressure of the gas at the spark. Thus, increasing the intake temperature and pressure, increasing the compression ratio and retarding the spark, at reduce the ignition lag.

*1.6 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, and has bearing or certain types of abnormal combustion. There are several factors which affect the flame speed, the most important being fuel-air ratio and turbulence.

- Fuel-air ratio: The composition of the working mixture influence the rate of combustion and the amount of heat evolved. With hydrocarbon fuels the maximum flame velocities occur when mixture strength is 110% of stoichiometric (i.e. about 10% richer than stoichiometric). When the mixture is made or is enriched and still more, the velocity of flame diminishes. Lean mixtures release less thermal energy resulting in lower flame temperature and flame speed. Very rich mixtures have incomplete combustion (Some carbon only burns to CO and not $CO_2$) which results in production of less thermal energy and hence flame speed is again low.
- Compression ratio: A higher compression ratio increases the pressure and temperature of the working mixture and decrease the concentration of residual gases. These favorable conditions reduce the ignition lag of combustion and hence less ignition advance is needed. High Pressures and temperatures of the compressed mixture also speed up the second phase of combustion.
- Intake temperature and pressure: Increase in intake temperature and pressure increase the flame speed.

## II. LITERATURE REVIEW

In this section, a review of work done so far in the field, heat transfer in S.I. engine, engine temperature, improvement of fuel economy, emissions for gasoline engine is given.

*2.1 Heat transfer in I. C. Engine:*

Satisfactory engine heat transfer is required for a number of important reasons, including material temperature limits, lubricant performance limits, emissions, and knock. Since the combustion process in an internal combustion engine is not continuous, as is the case for an external combustion engine, the component temperatures are much less than the peak combustion temperatures. Heat transfer to the air flow in the intake manifold lowers the volumetric efficiency, since the density of the intake air is decreased.

*2.2 Engine cooling systems:*

There are two types of engine cooling systems used for heat transfer from the engine block and head; liquid cooling and

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air cooling. With a liquid coolant, the heat is removed through the use of internal cooling channels with in the engine block as shown in fig.2.1.

*2.3 Engine Energy Balance:*

An energy balance is obtained through experiments performed on instrumented engines. Fig.2.2 depicts an engine instrumented to determine the quantities of heat rejected to oil water and to the ambient air. Flow meters are installed in the water, and oil circuits and thermocouples measure the inlet and out let temperatures. [2]

*2.4 Cylinder heat transfer measurements:*

There are a wide range of temperature and heat fluxes in an internal combustion engine. The values of local transient heat fluxes can vary by an order of magnitude depending on the spatial location in the combustion chamber and the crank angle. 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. When an engine is running at steady state, the heat transfer throughout most of the engine structure is steady.

*2.5 Theory on IC Engine heat transfer:*

The heat transfer process in an internal combustion can be modeled with a variety of methods. These methods range from simple thermal networks to multidimensional differential equation modeling. Thermal network models, using resistors and capacitors, are very useful for rapid and efficient estimation of the conduction, radiation, and convection heat transfer processes in engines. Using a thermal network, the significant resistances to heat flow and the effects of changing material thermal conductivity, thickness, and coolant properties can be easily determined.

The agreement obtained is reasonable except in the skirt area where the commutated temperatures are too low. The calculated results show that three areas are particularly important in dissipating the piston heat input: (1) the ring groove surfaces, (2) the underside of the dome, and (3) the upper portion of the pin bearing surface. From the ring grooves, heat flows in to the rings, through the bore, and is eventually absorbed by coolant. From the underside of the dome and the surface of the pin bearing, the heat is converted in to air-oil mist and is eventually absorbed by the oil in the sump. [5]

*2.6 Effect of Engine Temperature on S.I. Engine:*

Temperature control is very important for combustion engines as temperature is a critical factor both for chemical reactions and mechanical stresses. Traditionally, temperature control is performed by feedback of a global quantity, the coolant temperature, which however is a poor indicator of specific temperatures. The use of electrical pumps opens new possibilities for thermal control, in particular in terms of efficiency, but also of pollution, especially in the cold start phase. Shows that predictive control and the use of electrical coolant pumps allow to regulate specific temperatures - here as an example the cylinder head temperature. The experiment based results shown in impressive reduction of the thermal swing for the use of an electrical pump speed control using a model predictive approach. The work related has been performed in the framework of a Regions Interred project in cooperation with a leading producer of coolant pumps. [17]

*2.7 Fuel Economy For Gasoline Engine:*

Today research and development in the field of gasoline engines have to face a double challenge on the one hand; fuel consumption has to be reduced, while on the other hand, ever more stringent emission standards have to be fulfilled. The development of engines with its complexity of in-cylinder processes requires modern development tools to exploit the full potential in order to reduce fuel consumption. Combustion of gasoline by homogeneous compression ignition is widely accepted as a technology with a high potential for ultra-low emissions and high fuel economy.

Engine exhaust emission and fuel consumption during warm-up period was experimentally investigated. Experiment was conducted on a four-stroke four-cylinder spark ignition engine alternatively equipped with CIS (conventional ignition system) and EIS (electronic ignition system). Fuel consumption; and exhaust emissions included hydrocarbon, carbon monoxide and carbon dioxide were measured as a function of ambient temperature; i.e. 7, 25 and 40ºC. In order to simulate engine operation condition during warm - up period under various ambient temperatures axillaries cooling water and cooling air systems were designed and coupled to the engine being tested. Results show that as the ambient temperature increases the concentration of both hydrocarbon and carbon monoxide and fuel consumption decreases while the carbon dioxide increases. Also, the time required for the engine to fully warm-up is shortened. Moreover, operating the engine when equipped with EIS has a greater effect on hydrocarbon, carbon monoxide emissions and fuel consumption reduction compared to when equipped with CIS at the same operation conditions. The primary factors affecting vehicle exhaust emissions and fuel consumption during engine warm-up period can be classified into three categories; which are vehicle technology-related factors, fuel quality-related factors and engine-related factors. [19]

*2.8 Latest update in S.I. Engine:*

The old vehicles usually consumed more fuel and emit more pollutants than new ones as results of deterioration in engine efficiency and lack of proper maintenance. In addition, engine of old vehicles used conventional ignition system with mechanical breaker points. In practice operation of these systems encountered the following faults: poor performance at high speed, inability to fire partially fouled spark plugs, relatively short life of the breaker points and spark plugs, poor starting and poor reproducibility of the secondary voltage rise to the maximum value. The effect of these factors combined led to increasing fuel consumption, hydrocarbon and carbon monoxide emissions. To overcome these faults in modern vehicle engines conventional ignition system have been replaced with coil ignition system (high energy electronic ignition systems). Electronic ignition systems (EIS) produce larger output voltage (35 kV) and longer spark duration. Therefore extending the engine operating conditions over which satisfactory ignition is achieved. [19]

*2.9 Heat release rates in spark ignition Engines:*

Net chemical heat-release rates have been estimated experimentally, throughout the combustion, from a single-cylinder gasoline engine, running on paraffin and aromatic

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fuels. These rates are compared for auto igniting and non-auto igniting cycles and, by means of a differencing procedure, the heat release rate due to auto ignition found. Comparison of heat release rates in the propagating flame and in auto ignition show that in knocking combustion, almost half the total energy release can occur in auto ignition. Pressures were measured with transducers and gas temperatures prior to auto ignition by the CARS technique. The measurements enabled volumetric auto ignition heat release rates to be obtained. When plotted against the reciprocal temperature of the unburned gas just prior to auto ignition, an activation temperature and Arrhenius constant were obtained for each fuel in a single global expression for the auto ignition heat release rate. These constants are expressed in terms of the actual temperature and fuel concentration

*2.10 Emissions from gasoline Engines:*

The experimental study results carried out on an electronically controlled fuel injection 'stoichiometric gasoline engine' by using cold EGR and increasing 'compression ratio' to improve fuel economy and reduce emissions. After the compression ratio of the engine is raised from 8 to 11.8, and EGR rate and air swirl ratio are optimized, the fuel economy is improved by 6.02%, and the NOx and (NOx + HC) emissions are decreased by 52.96% and 44.94%, respectively at full-load speed characteristics. The calculation results of heat release rates according to the measured indicator diagram show that the combustion process is remarkably improved. Electronically controlled gasoline injection, stoichiometric air–fuel ratio adaptively controlled by oxygen sensor and three-way catalyst converter are widely used in gasoline engine to control CO, HC and NOx emissions. However, stoichiometric mixture is not the economical mixture for gasoline engines, which may lead to the poor fuel economy and consequently the increase in CO2 emission, a global green-house gas. So it is an important topic to improve the fuel economy of gasoline engines and make it operate at stoichiometric air–fuel ratio without deteriorating emissions from the engine. An investigation is carried out to improve both fuel economy and emissions on gasoline engine at stoichiometric ratio by using cold EGR to suppress the knocking and increasing in compression ratio of the engine. [20]

## III. EXPERIMENTAL SETUP

*3.1 Description:*

The setup consists of single cylinder, four strokes, and Petrol Carburetor engine connected to rope belt type dynamometer for loading. It is provided with necessary instruments for combustion pressure and crank-angle measurements. Provision is also made for interfacing airflow, fuel flow, temperatures and load measurement. The setup has standalone panel box consisting of air box, fuel tank, and manometer, fuel measuring unit, process indicator, load indicator and engine indicator. Rota meters are provided for cooling water and calorimeter water flow measurement. The setup enables study of engine performance for brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, Mechanical efficiency, volumetric efficiency, specific fuel consumption, A/F ratio and heat balance.

The project experiment was carried out on Single Cylinder 4-Stroke constant speed SI engine. It consists of following components:-

- Fuel tank
- Engine
- Rope brake drum type dynamometer
- Manual loading arrangement
- AVL Five gas analyzer

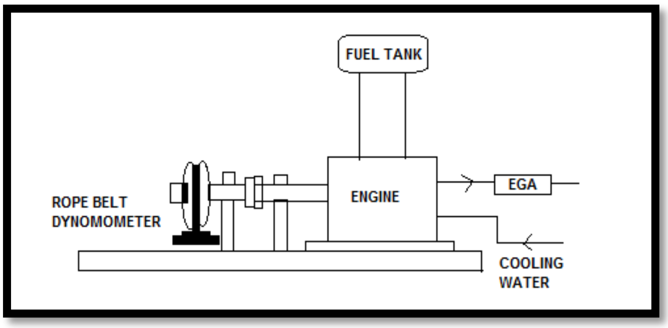


Fig .3.1 Schematic view of experimental setup

3.2. Engine Specifications:-

Maker- Grieves ltd.
Type- Single Cylinder 4-Stroke SI engine
Rated Power- 2.6 KW
Rated RPM- 3000
Stroke- 66.7 mm
Bore- 70 mm
Compression ratio- 4.7
Capacity- 256 cc
Arm length- 0.1036 m
Propeller shaft : With universal joints
Air box : M S fabricated with orifice meter and manometer
(Orifice dia 35 mm)
Fuel tank : Capacity 8 lit with glass fuel metering column
Calorimeter : Type Pipe in pipe
Temperature sensor : Type RTD, PT100 and Thermocouple, Type K
Temperature : Type two wire, Input RTD PT100, Range 0–100o C
Pump : Type Monoblock

*3.3 Installation requirements:*

Electric supply: Provide 230 +/- 10 VAC, 50 Hz, single phase electric supply with proper earthing. (Neutral – Earth Electric supply with proper earthing. (Neutral – Earth voltage less than 5 VAC) • 5A, three pin socket with switch (2 Nos.)

Water supply Continuous, clean and soft water supply @ 4000 LPH, at 10 m. head. Provide tap with 1" BSP size connection

Space : 3500Lx4000Wx2000H in mm
Drain : Provide suitable drain arrangement (Drain pipe 65 NB/2.5" size)
Exhaust : Provide suitable exhaust arrangement (Exhaust pipe 32 NB/1.25" size)

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Foundation : As per foundation drawing
Fuel, oil : Petrol @ 5 liter Oil @ 1.5 lit. (20W40)

Fig.3.2 Single cylinder for stroke petrol engine with rope brake Dynamometer

The setup consists of single cylinder, four stroke, Petrol Carburettor engine connected to rope belt type dynamometer for loading. It is provided with necessary instruments for combustion pressure and crank-angle measurements. Provision is also made for interfacing airflow, fuel flow, temperatures and load measurement. The setup has standalone panel box consisting of air box, fuel tank, manometer, fuel measuring unit, process indicator, load indicator and engine indicator. Rotameters are provided for cooling water and calorimeter water flow measurement. The setup enables study of engine performance for brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, Mechanical efficiency, volumetric efficiency, specific fuel consumption, A/F ratio and heat balance.

*3.4 Theoretical constants:*

Fuel density : 740 kg/m^3
Calorific value : 44000 kJ/kg
Orifice coefficient of discharge: 0.60
Sp heat of exhaust gas: 1.00 kJ/kg-K
Max sp heat of exhaust gas: 1.25 kJ/kg-K
Min sp heat of exhaust gas: 1.00 kJ/kg-K
Specific heat of water: 4.186 kJ/kg-K
Water density : 000 kg/m^3
Ambient temperature : 300C
Sensor range
Exhaust gas temp. trans. (Engine):0-1200 C
Load cell : 0-50 kg

*3.5 Gas analyzer:*

The AVL DIGAS 444 the standard configuration is recommended to be used with S.I.Engines with diesel vehicle testing. a special probe required and for test bed application sampling system required.

Type - AVL DiGas 444

| Measured Quality | Measuring Range |
|---|---|
| CO | 0… 10 % vol |
| CO2 | 0… 20 % vol |
| HC | 0… 20000 ppm |
| O2 | 0… 22 % vol |
| NOx | 0… 5000 ppm |

Fig. 3.3 AVL DIGAS 444 Gas Analyzer

## IV. METHODOLOGY

Experiment was conducted on a three cylinder, four stroke, Petrol engine which is connected to rope belt type dynamometer for loading. The performance results which include Brake Power (B.P.) and Specific Fuel Consumption (SFC) as a function of engine temperature; i.e. 50, 60, 70 and 80ºC are reported. The emissions results reported include the concentrations of hydrocarbon, oxides of nitrogen in the exhaust. The test has been conducted to study the effect of engine temperature on SFC and B.P. with varying engine speed i.e. 1500, 2000, 2500 rpm with the load of 6,9,12 kg.

Engine temperature has been controlled by controlling cooling water flow rate. The cooling water flow rate for engine is measured manually by rotameter. The values of engine performance parameter are directly obtained by "Engine Soft" software.

A test matrix is created to record the engine performance parameter but main focal point was on specific fuel consumption and brake power of the engine at different engine speed 1500, 2000, 2500 rpm with the engine load of 6,9,12 kg at engine temperature 50,60,70,80 ºC.

Calculation

Fuel consumption in kg/sec

$$\text{fuel consumption rate} = \frac{\text{fuel consumption}}{\text{time}} \quad \text{kg/sec}$$

Torque in NM

$$\text{Torque} = \text{force} * \text{arm length}$$

Break power in kw

$$\text{bp} = \frac{2 * 3.142 * \text{N} * \text{T}}{60000}$$

Break specific fuel consumption kg/kwh

$$\text{bsfc} = \frac{\dot{m}_f}{\text{bp}}$$

Break mean effective pressure in bar

$$\text{bmep} = \frac{\text{bp} * 60}{\text{L} * \text{A} * \text{N} * \text{K}}$$

Break thermal efficiency:

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$$\eta_{bth} = \frac{bp}{m_{\dot{f}*C_v}}$$

Break specific energy consumption:

bsec = bsfc * calorific value

Table Test Matrix

| S. No. | Engine Speed (rpm) | Engine Load(kg) | Engine Temp. (ºC) |
|---|---|---|---|
| 1 | 1500 | 6<br>9<br>12 | 50,60,70,80 |
| 2 | 2000 | 6<br>9<br>12 | 50,60,70,80 |
| 3 | 2500 | 6<br>9<br>12 | 50,60,70,80 |

Table SFC, BP, HC AND NO Emission At Engine Speed 1500rpm And 6 Kg Engine Load

| S.NO. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 1500 | 6 | 50 | 376 | 0.26 | 114 | 255 |
| | | | 60 | 324 | 0.28 | 121 | 262 |
| | | | 70 | 340 | 0.26 | 119 | 282 |
| | | | 80 | 272 | 0.28 | 118 | 296 |

Table SFC, BP, HC AND NO Emission at Engine speed 2000rpm and 6 kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 2000 | 6 | 50 | 356 | 0.3 | 125 | 217 |
| | | | 60 | 280 | 0.34 | 127 | 246 |
| | | | 70 | 288 | 0.34 | 132 | 290 |
| | | | 80 | 280 | 0.36 | 133 | 307 |

Table No. 3.4 SFC, BP, HC AND NO Emission At Engine Speed 2500rpm And 6 Kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 2500 | 6 | 50 | 232 | 0.45 | 150 | 201 |
| | | | 60 | 312 | 0.41 | 151 | 199 |
| | | | 70 | 276 | 0.46 | 150 | 281 |
| | | | 80 | 264 | 0.46 | 151 | 322 |

Table SFC, BP, HC AND NO Emission At Engine Speed 1500rpm And 9 Kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 1500 | 9 | 50 | 288.4 | 0.42 | 157 | 360 |
| | | | 60 | 256 | 0.4 | 152 | 484 |
| | | | 70 | 257.6 | 0.38 | 153 | 507 |
| | | | 80 | 249.6 | 0.39 | 154 | 390 |

Table No. 3.6 SFC, BP, HC AND NO Emission At Engine Speed 2000rpm And 9 Kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 2000 | 9 | 50 | 256.4 | 0.49 | 160 | 421 |
| | | | 60 | 262.4 | 0.51 | 163 | 540 |
| | | | 70 | 257.2 | 0.49 | 159 | 528 |
| | | | 80 | 228 | 0.52 | 157 | 584 |

Table No. 3.7 SFC, BP, HC AND NO Emission At Engine Speed 2500rpm And 9 Kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 2500 | 9 | 50 | 257.2 | 0.57 | 148 | 322 |
| | | | 60 | 249.2 | 0.64 | 161 | 584 |
| | | | 70 | 251.2 | 0.64 | 155 | 524 |
| | | | 80 | 244 | 0.66 | 153 | 426 |

Table No. 3.8 SFC, BP, HC AND NO Emission At Engine Speed 1500rpm And 12 Kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 1500 | 12 | 50 | 232 | 0.5 | 125 | 466 |
| | | | 60 | 236 | 0.52 | 132 | 524 |
| | | | 70 | 240 | 0.48 | 128 | 547 |
| | | | 80 | 228 | 0.5 | 114 | 494 |

Table SFC, BP, HC AND NO Emission At Engine Speed 2000rpm And 12 Kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 2000 | 12 | 50 | 240 | 0.65 | 162 | 783 |
| | | | 60 | 232 | 0.66 | 168 | 807 |
| | | | 70 | 224 | 0.7 | 169 | 839 |
| | | | 80 | 212 | 0.65 | 172 | 848 |

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Table SFC, BP, HC AND NO Emission At Engine Speed 2500rpm And 12 Kg Engine Load

| S.N. | Engine Speed (rpm) | Engine Load (kg) | Engine Temp. (ºC) | SFC in (g/kwhr) | B.P. in KW | HC in ppm | NO in ppm |
|---|---|---|---|---|---|---|---|
| 1. | 2500 | 12 | 50 | 220 | 0.81 | 164 | 851 |
| | | | 60 | 216 | 0.81 | 164 | 856 |
| | | | 70 | 212 | 0.84 | 162 | 922 |
| | | | 80 | 196 | 0.81 | 162 | 970 |

## V. CONCLUSION

A single cylinder, four stroke, petrol carburetor engine connected to rope brake type dynamometer for loading was adopted to study engine power, fuel economy, engine exhaust emissions of hydrocarbon, and oxides of nitrogen in the exhaust. The performance results that are reported include brake power and specific fuel consumption (sfc) as a function of engine temperature; i.e. 50, 60, 70 and 80ºC with varying engine speed of 1500, 2000, 2500, rpm.

## REFERENCES

[1] Bendersky, d. (1953),"a special thermocouple for measuring transient temperature," mech. eng., 75, p.117.

[2] WHITEHOUSE, N.D. (1970-1971), "Heat Transfer in a Quiescent Chamber Diesel Engine," Proc. instn. Mech. Engrs., 185, p. 963-975.

[3] HIRES, S.D. and G.L. POCHMARA (1976),"An Analytical Study of Exhaust Gas Loss in a Piston Engine Exhaust Port", SAE paper 760767.

[4] CATON, J.A. and J.B.HEYWOOD (1981),"An Experimental and Analytical Study of Heat Transfer in An Engine Exhaust Port", Int. J. Heat Mass Transfer, 24(4), p. 581-595.

[5] LI, (1982),"Piston Thermal Deformation and Friction Considerations," SAE paper 820086.

[6] TAYLOR, C.F. (1985), "The Internal Combustion Engine in Theory and Practice," MIT Press, Cambridge, Massachusetts.

[7] BORMAN, G. and K. NISHIWAKI (1987), "International Combustion Engine Heat Transfer," Prog. Energy Combustion Sci., 13, p. 1 – 46.

[8] HEYWOOD,J.(1988),"International Combustion Engine Fundamentals," Mc Graw Hill, New York

[9] SHAYLER, P., S. CHARISTIAN, and T. MA, (1993), "A model for The investigation of Temperature, Heat Flow, and Friction Characteristics During Engine Warm up," SAE paper 931153

[10] Brehob, DIANA d., ANDERSON, RICHARD YANG, JIA LIN, (January 9, 1996), "Reduction of Cold-Start Emissions and Catalyst Warm-Up Time With Direct Fuel Injection".

[11] D. BRADLEY, G. T. KALGHATGI, M. GOLOMBOK, JINKU YEO, (1996) "Heat Release Rates Due to Autoignition, and Their Relationship to Knock Intensity In Spark Ignition Engines", Twenty-Sixth Symposium (International) on Combustion/The Combustion Institute, pp. 2653–2660.

[12] ROBINSON, K., N.CAMPBELL, J.HAWLEY. (1999) and D. TILLEY (1999),"A Review of Precision Engine Cooling," SAE paper 1999-01-0578.

[13] Y. OHYAMA, (February 2001), "Engine Control Using Combustion Model," Hitachi Car Engineering Co., Ltd., 2520 Takaba, Hitachinaka 312-8503, Japan

[14] INCROPERA, F. and D. DEWITT, (2001), "Fundamental of Heat and Mass Transfer," John Wiley and Sons, new york.

[15] J.W.G.TURNER, R.J.PEARSON, M.D.BASSETT, (2003),"Performance and Fuel Economy Enhancement of Pressure Charged SI Engines Through Turboexpansion- An Initial Study,"

[16] Osman Akin Kutlar, Hikmet Arslan, Alper Tolga Calik, (December 2005)," Methods to Improve Efficiency of Four Stroke, Spark Ignition Engines at Part Load", Energy Conversion and Management, Volume 46, Issue 20, Pages 3202-3220.

[17] BRUCKNER,M,GRUENBACHER,E,ABERER,D, RE,L.D.,ATSCHREITER,F,(Oct 2006), "Predictive Thermal Management of Combustion Engine,", page 2778-2783,

[18] Li Jafari, Siamak Kazemzadeh Hannani, (January 2006),"Effect of Fuel and Engine Operational Characteristics on The Heat Loss From Combustion Chamber Surfaces of SI Engines", International Communications in Heat and Mass Transfer, Volume 33, Issue 1, Pages 122-134.

[19] M. Al-Hasan, (2007),"Evaluation of Fuel Consumption and Exhaust Emissions During Engine Warm-up," American Journal of Applied Sciences 4 (3), 106-111, ISSN 1546-9239.

[20] Li WEI, Wang YING, Zhou LONGBAO, Su Ling, (10 April 2007), "Study on Improvement of Fuel Economy and Reduction in Emissions for Stoichiometric Gasoline Engines,

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