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A SIMULATE MODEL FOR ANALYZING THE EFFECT OF ENGINE DESIGN PARAMETERS ON THE PERFORMANCE AND EMISSIONS OF SPARK IGNITION ENGINES

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ABSTRACT

A mathematical and simulation model has been developed to simulate a spark ignition engine operation cycle. The programme written from this simulation model and modified so can be used to assist in the design of a spark ignition engine for alternative fuels as well as to study many design parameters such as the effect of engine design parameter like stroke and diameter of the cylinder on the performance and exhaust emissions of spark ignition engines. In this paper, the description of three type of engines design parameters according to (stroke/diameter) which are called under square, square and over square engines using three type of fuel (gasoline, LPG, CNG) singly have been taken. The variations of power output, thermal efficiency, specific fuel consumption, ignition delay, temperature of exhaust gases, heat loss and the exhaust emissions (CO2, CO, NO, O2) by volume, were calculated for each type of engine and then comparing the results to investigate how to increase engine efficiency and reduce exhaust emissions. The accuracy of the present model is confirmed by comparison with experimental result that have established by previous literatures and the results were agreement with literatures. The main results obtained from the present study shows that, the square engines (S/D=1) can be considered as most efficient, modern and suitable design for engines running by gasoline and alternative fuels like LPG and CNG, because of higher power output due to the suitable piston area for pressure distribution. The area of flame front which may create higher combustion qualities and higher thermal efficiency through faster burning and lower overall chamber heat loss, also, the potential of the square engines (S/D=1) for indicate specific fuel consumption was 4% improved.

Keywords: Two-zone combustion, Combustion simulation, Percentage heat loss, Spark ignition engine.

INTRODUCTION

Considering the energy crises and pollution problems today, investigations have concentrated on decreasing fuel consumption by using alternative fuels and on lowering the concentration of toxic components in combustion products**,** there are so many different engine manufacturers, past, present, and future, that produce and have produced engines which differ in size, geometry, style, and operating characteristics that no absolute limit can be stated for any range of engine characteristics (i.e., size, number of cylinders, strokes in a cycle, etc.

Combustion is an important subject of internal combustion engine studies, to reduce the air pollution from internal combustion engines and to increase the engine performance; it is required to increase combustion efficiency. In this study, two basic types of models have been developed. They can be categorized as thermodynamic and fluid dynamic in nature, depending on whether the equations, which give the model its predominant structure are name given to thermodynamic energy conservation based models are : zero-dimensional phenomenological and quasi-dimensional.

Fluid dynamic based models are often called multidimensional models due to their inherent ability to provide geometric information on the flow field, based on the solution of governing flow equations.

Many mathematical models have been developed to help understand, correlate, and analyze the operation of engine cycles. These include combustion models, models of physical properties, and models of flow into, through, and out of the cylinders. Even though models often cannot represent processes and properties to the finest detail, they are a powerful tool in the understanding and development of engines and engine cycles. With the use of models and computers in the design of new engines and components, great savings are made in time and cost. Historically, new design was a costly, time-consuming practice of trial and error, requiring new part construction and testing for each change. Now engine changes and new designs are first developed on the computer using the many models which exist. Often, only after a component is optimized on the computer is a part actually constructed and tested.

Generally, only minor modifications must then be made to the actual component. Models range from simple and easy to use, to very complex and requiring major computer usage. In general, the more useful and accurate models are quite complex. Models to be used in engine analysis are developed using empirical relationships and approximations, and often treat cycles as quasi-steady state processes, Normal fluid flow equations are often used [8].

Some models will treat the entire flow through the engine as one unit, some will divide the engine into sections, and some will subdivide each section (e.g., divide the combustion chamber into several zones-burned and unburned, boundary layer near the wall, etc.). Most models deal only with one cylinder, which eliminates any interaction from multi-cylinders that can occur, mainly in the exhaust system.

Models for the combustion process address ignition, flame propagation, flame termination, burn rate, burned and unburned zones, heat transfer, emissions generation, knock, and chemical kinetics. They are available for spark ignition engines with either direct injection or indirect injection. Values for properties are obtained from standard thermodynamic equations of state and relationships for thermo-physical and transport properties.

Theoretical Analysis

The power cycle analysis in 4-stroke, spark ignition engine uses the first law of thermodynamics by integrating various models for combustion, heat transfer, and the value of flow rates, using integration methods from. Integration proceeds by crank angle position, allowing for various fuels, air-fuel ratios, EGR or other inert gas for charge dilution, and/or valve-opening profiles. The power cycle is divided into three principal sections: compression stroke, combustion stages and expansion stroke.

Compression stroke: Its adiabatic processes in which the integration starts at the closing of the intake valve and proceeds until the crank angle of ignition is reached. Residual gases from the previous cycle are included in the cylinder gas mixture, and a number of iterations are performed until the percentage and chemical content of the residual gases remain at a steady state value after each cycle [10].

1. The total cylinder volume at each crank angle degrees can be written as [18]

$$V(\theta) = V_c + \frac{V_s}{2}\left[1 - \mathrm{Cos}\,(\theta) + \frac{2L}{S} - \sqrt{\left(\frac{2L}{S}\right)^2 - \mathrm{Sin}^2\theta}\right] \tag{3.1}$$

The temperature of unburned mixture [T_2] can be calculated from the following equation

$$T_2 = T_1 * \left[\frac{V_1}{V_2}\right]^{k-1} = T_1 * \left[\frac{V_1}{V_2}\right]^{\frac{R_o}{C_v(T_1)}} \tag{3.2}$$

The pressure of unburned mixture also can be calculated from the following equation:

$$P_2 = \left[\frac{V_1}{V_2}\right] * \left[\frac{T_2}{T_1}\right] * P_1 \tag{3.3}$$

2. Work can be calculated in compression stroke because of small variation of pressure and volume inside the cylinder and the equation can be written as:

$$dW = PdV = \left(\frac{P_1 + P_2}{2}\right) * \left(V_2 - V_1\right) \tag{3.4}$$

3. The first law of thermodynamic for checking values of (T_2, P_2), and the whole control volume (two zones) can be written as:

$$dQ - dW = dE = E(T_2) - E(T_1) \tag{3.5}$$

where dQ_{conv} is the rate of heat transfer by convection through the cylinder walls can be determine by applying (Eichelberg Equation) which is written as

$$dQ_{conv} = h * dA_s * (T - T_{wall}) \tag{3.6}$$

Where:

$$h = 2.466 * 10^{-4} (\overline{U}_p)^{\frac{1}{3}} (P)^{\frac{1}{2}} (T)^{-\frac{1}{2}}$$

$$A = \left(2A_p + 4 * \frac{V_c}{d} + \pi de \right)$$

$$T = \left(\frac{T_1 + T_2}{2} \right)$$

$$P = \left(\frac{P_1 + P_2}{2} \right)$$

$$\overline{U}_p = \left\{ \frac{2 * S * N}{60} \right\}$$

4. The first law of thermodynamic can be written as[18] :

$$f(E) = E(T_2) - E(T_1) + dW - dQ_{conv} \tag{3.7}$$

The equation (3.7), can be solved numerically by Newton-Raphson method of iteration and the equation (3.7) to be achieved when f (E) equal to zero.

By this method we can find $(T_2)_{n-1}$ which is more accurate:

$$(T_2)_n = (T_2)_{n-1} - \frac{f(E)_{n-1}}{f'(E)_{n-1}} \tag{3.8}$$

$f'(E)$ its derivation of equation (3.7) for rate of (T) and for determine the value of $f'(E)$ we can consider $(\frac{dW}{dT} = 0)$ because (dW) not affected by (T_2):

$$f'(E) = \frac{dE\ (T_2)}{dT} \tag{3.9}$$

The rate of internal energy $f'(E)$ at constant volume can be written as:

$$f'(E) = mC_v (T_2)$$

After substituting $f'(E)$ in equation (3.8) finally it will become:

$$(T_2)_n = (T_2)_{n-1} - \frac{f(E)_{n-1}}{mC_v (T_2)_{n-1}} \tag{3.10}$$

To find the value of pressure at combustion duration can be written as:

$$PV = mRT \tag{3.11}$$

By derivation of equation (3.11) with respect to (θ) can be written as:

$$P\frac{dV}{d\theta} + V\frac{dP}{d\theta} = mR\frac{dT}{d\theta} \tag{3.12}$$

After substitute the value of (mR) from eq.(3.11) to eq.(3.12) and we now have :

$$P\frac{dV}{d\theta} + V\frac{dP}{d\theta} = \frac{PV}{T}\frac{dT}{d\theta} \tag{3.13}$$

By dividing eq.(3.13) on (PV) and rearrangement, the equation drive to :

$$\frac{1}{P}\frac{dP}{d\theta} + \frac{1}{V}\frac{dV}{d\theta} = \frac{1}{T}\frac{dT}{d\theta} \tag{3.14}$$

The first law of thermodynamic incases of ideal gas and constant specific enthalpy can be written as:

$$mC_v\frac{dT}{d\theta} = \frac{dQ}{d\theta} - P\frac{dV}{d\theta} \tag{3.15}$$

After dividing left side by (mRT) and right side on (PV) of eq.(3.15) and become :

$$\frac{1}{T}\frac{dT}{d\theta} = (K-1)\left(\frac{1}{PV}\frac{dQ}{d\theta} - \frac{1}{V}\frac{dV}{d\theta}\right) \tag{3.16}$$

By summation two equations, eq. (3.14) and eq. (3.16) now we have:

$$\frac{dP}{d\theta} = -K\frac{P}{V}\frac{dV}{d\theta} + (K-1)\frac{1}{V}\frac{dQ}{d\theta} \tag{3.17}$$

The rate of $\frac{dQ}{d\theta}$ can be calculated from the following equation:

$$\frac{dQ}{d\theta} = \left|\frac{dQ}{d\theta}\right|_{app} + h * A_s(T - T_{wall}) \tag{3.18}$$

To find the value of $\left(\frac{dP}{d\theta}\right)$, we need to solve eq.(3.17) numerically using Runge-Kutta method, fourth order.

The calculation start for the combustion products at every step of combustion duration using general equation of combustion of hydrocarbon and air which is represent by :

$$a(C_nH_mO_r) + \frac{1}{\Phi}(n + \frac{m}{4} - \frac{r}{2})\left[O_2 + \frac{78}{21}N_2 + \frac{1}{21}Ar\right] \rightarrow \sum_{i=1}^{i=q} X_i Z_i \tag{3.19}$$

From eq. (3.19) we can calculate amount of exhaust emissions for each compound like (CO, CO2, NO, O2).

Expansion stroke : Two control masses are considered during this process, cylinder gases and exhaust gases downstream of the exhaust valve, the method of calculations is same like compression stroke, except the mixture include product compound and their concentrations will calculated till exhaust valve opened.
Exhaust emission compound can be express by following chemical equations [18]:

$$\frac{1}{2}H_2 \rightarrow H \tag{3.20}$$

$$\frac{1}{2}O_2 \rightarrow O \tag{3.21}$$

$$\frac{1}{2}N_2 \rightarrow N \tag{3.22}$$

$$2H_2O \rightarrow 2H_2 + O_2 \tag{3.23}$$

$$H_2O \rightarrow OH + \frac{1}{2}H_2 \tag{3.24}$$

$$CO_2 + H_2 \rightarrow H_2O + CO \tag{3.25}$$

$$H_2O + \frac{1}{2}N_2 \rightarrow H_2 + NO \tag{3.26}$$

Generally, the chemical equilibrium constant (KP) of any chemical reaction (stoichiometric reaction) for those element, A,B,C,D can be represent by the following chemical equation:

$$V_a A + V_b B \leftrightarrow V_c C + V_d D \tag{3.27}$$

By rearrangement:

$$KP = \left[\frac{X_c^{V_c} X_d^{V_d}}{X_a^{V_a} X_b^{V_b}}\right] \tag{3.36}$$

Where:
V = Stoichiometric coefficient, which is calculated from chemical equation.
X = Mole fraction, can be calculated from general equation of combustion of hydrocarbon and air.

Using chemical reaction equations from eq. (3.20) to eq. (3.26), and by applying reaction equation (3.36), chemical equilibrium constants can be written as:

$$KP_1 = \left(X_4 / \sqrt{X_2}\right)\sqrt{P} \tag{3.37}$$

$$KP_2 = \left(X_{11} / \sqrt{X_{10}}\right)\sqrt{P} \tag{3.38}$$

$$KP_3 = \left(X_7 / \sqrt{X_5}\right)\sqrt{P} \tag{3.39}$$

$$KP_4 = \left(X_{10} / b^2\right)P \tag{3.40}$$

$$KP_5 = \left(X_3 / b\sqrt{X_2}\right)\sqrt{P} \tag{3.41}$$

$$KP_6 = \left(bX_9 / X_8\right) \tag{3.42}$$

$$KP_7 = \left(X_6 / b\sqrt{X_5}\right)\sqrt{P} \tag{3.43}$$

Where, $b = \dfrac{X_1}{X_2}$

The chemical equilibrium constant (KP) for those reactions can be calculated from the following relation equation:

$$LnKP = \left[\sum\left(\frac{vg(T)}{RT}\right)_R - \sum\left(\frac{vg(T)}{RT}\right)_P\right] - \frac{\Delta H_\circ}{RT} \tag{3.44}$$

Where,
ΔH_0 is the enthalpy drop among the reactants and products at zero absolute (Kj).
g(T) is Gibbs Function and represent by following equation:

$$\frac{g(T)}{RT} = a_1(1 - \ln T) - a_2 T - \frac{a_3}{2}T^2 - \frac{a_4}{3}T^3 - \frac{a_5}{4}T^4 - a_6 \tag{3.45}$$

where $a_1, a_2, a_3, a_4, a_5, a_6$ are the constant values.

Now we have 7 reaction equations. 3 of them represent the reactants and 4 of them for products, this equations are converted to computer language so that can be run by Quick Basic program, some of input data are constant and others are variable.

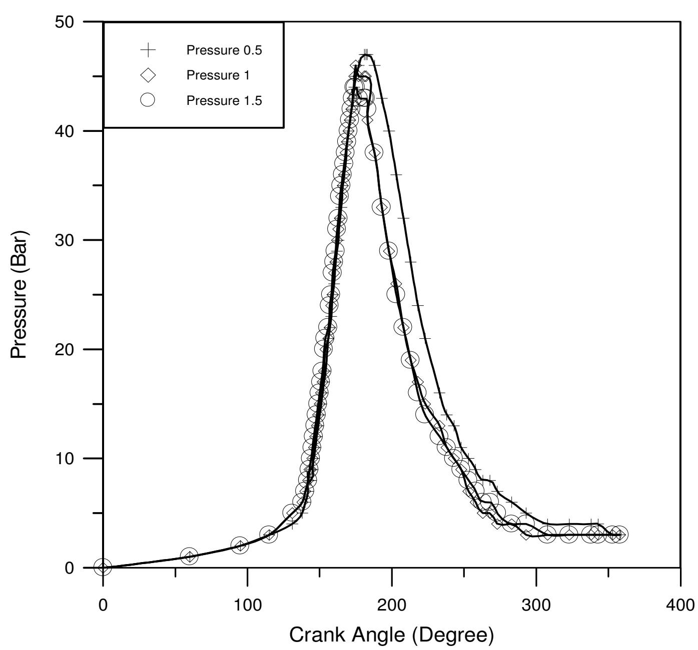

Fig. 3.1 Effect of variation (S/D) on pressure and crank angle curves using gasoline fuel (C_8H_{18}) at speed (1500 rpm)

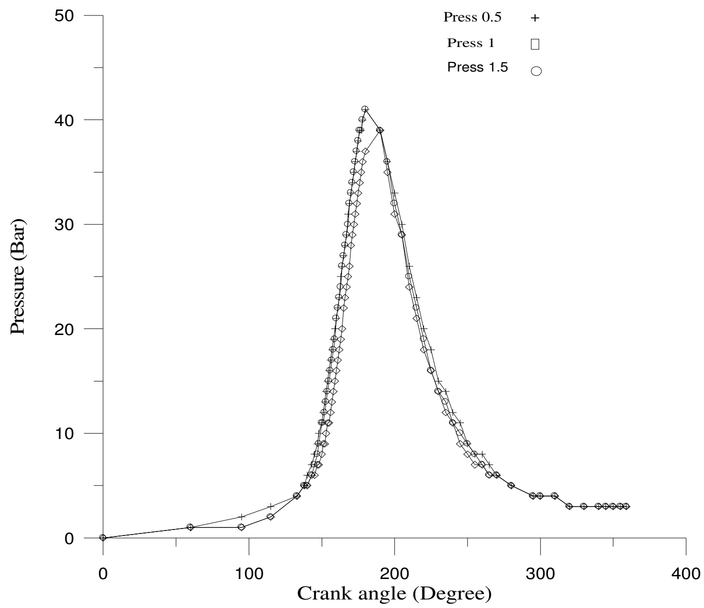

Fig.3.2 Effect of variation (S/D) on pressure and crank angle curves using LPG fuel (C_3H_8) at speed (1500 rpm)

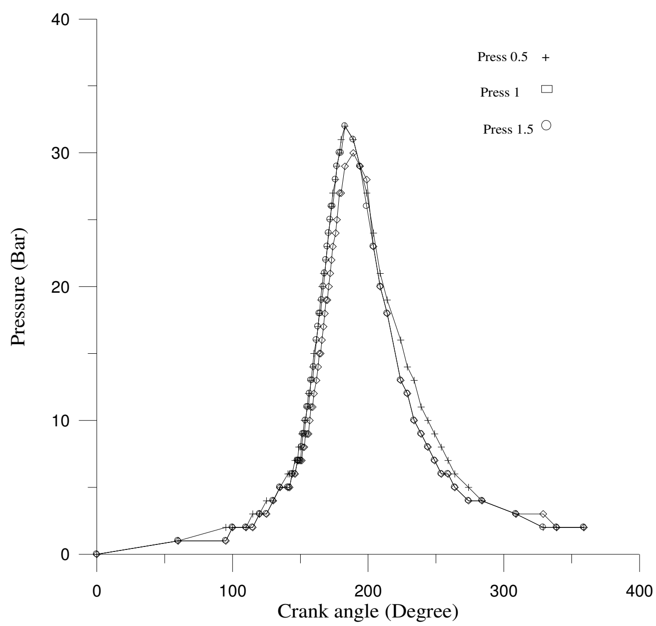

Fig. 3.3 Effect of variation (S/D) on pressure and crank angle curves using CNG fuel (CH_4) at speed (1500 rpm)

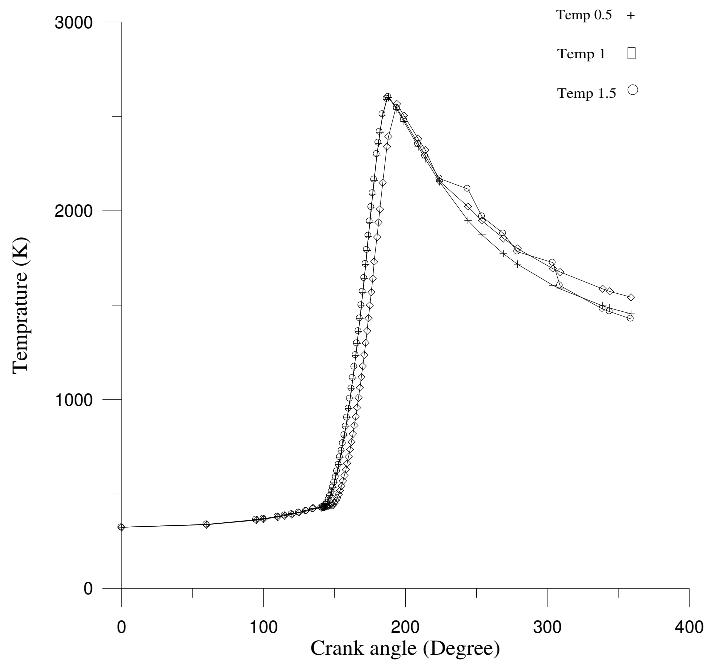

Fig. 3.4 Effect of variation (S/D) on temperature and crank angle curves using gasoline fuel (C_8H_{18}) at speed (1500 rpm)

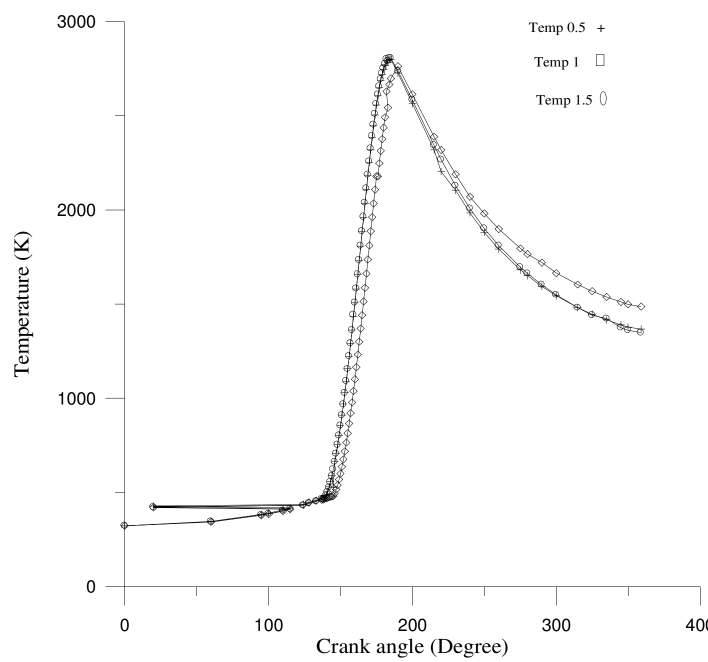

Fig. 3.5 Effect of variation (S/D) on temperature and crank angle curves using LPG fuel (C_3H_8) at speed (1500 rpm)

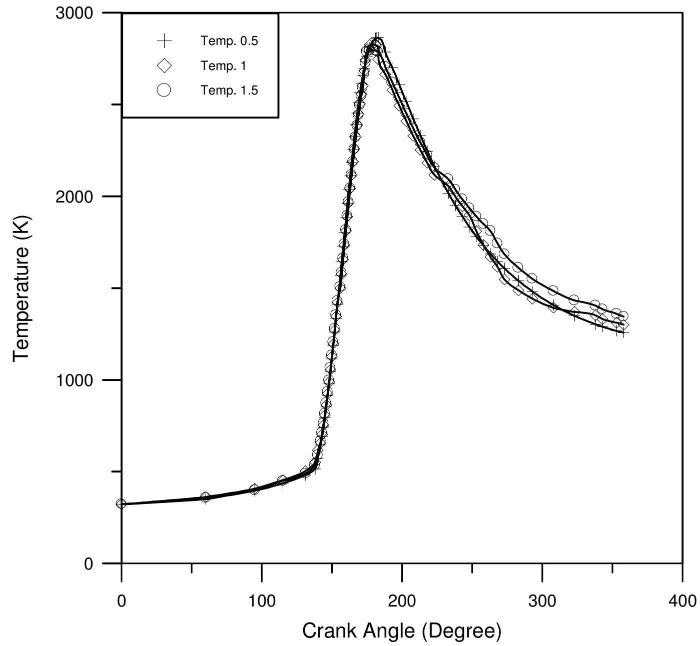

Fig. 3.6 Effect of variation (S/D) on temperature and crank angle curves using CNG fuel (CH_4) at speed (1500 rpm)

RESULTS AND DISCUSSION

4.1 The Effect of Engine Design Parameters on the Performance

The data results from computer programme were converted to graphs, so that to be easy to the reader shown in figures below. The important parameters will discuss in this chapter according to the results from programme which is include, power output, indicate specific fuel consumption, pressure mean effective, emissions, thermal efficiency and heat loss.

4.2 The Effect of Engine Design Parameters on the Power Output

According to the data from the programme, exactly when the engines fueled by gasoline the maximum indicate power or power output from the cylinder was obtained from square engine, then 50% of power output decrease when engine size changed to over square, also for under square engine, the power output decrease about 60% as shown in fig.4.1.

For the engines running by LPG, the maximum power output obtained from square engine, but to compare it with square engine running by gasoline is lesser because of the amount of air injected to cylinder is less, so that reduce volumetric efficiency.

The power output decrease about 55% from maximum power output for over square engines and then 60% decrease when engine size is under square as shown in fig.4.2.

Finally, for the engines using CNG as fuel, the maximum power output achieved from square engine, but to compare it with square engine running by gasoline and LPG is lesser for the same reason, also the power output decrease about 60% for over square and decrease 65% for under square as shown in fig.4.3.

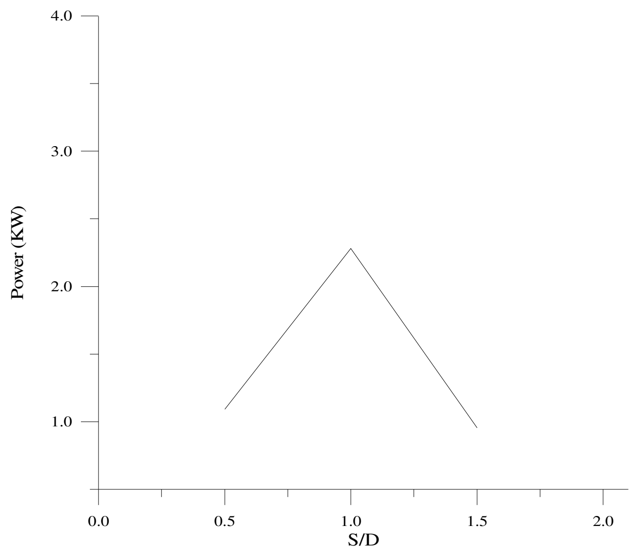

Fig.4. 1 Effect of S/D on Power using gasoline fuel (C_8H_{18})

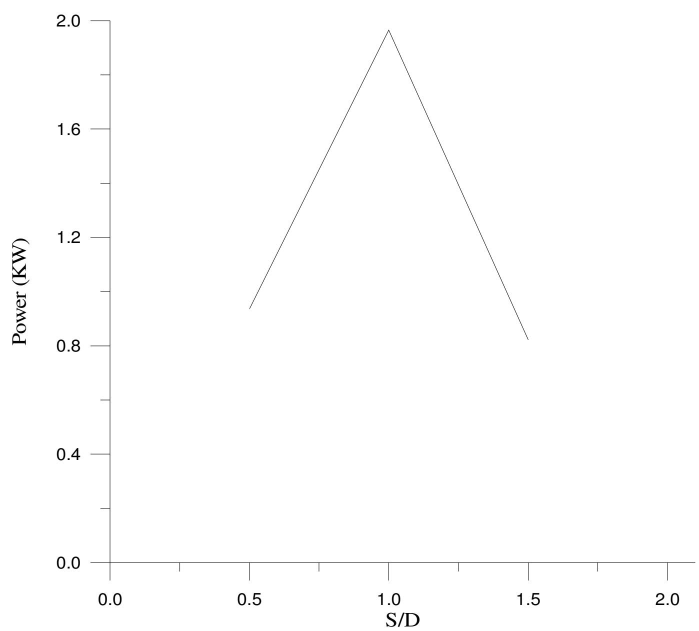

Fig.4.2 Effect of variation S/D on Power using LPG fuel (C_3H_8)

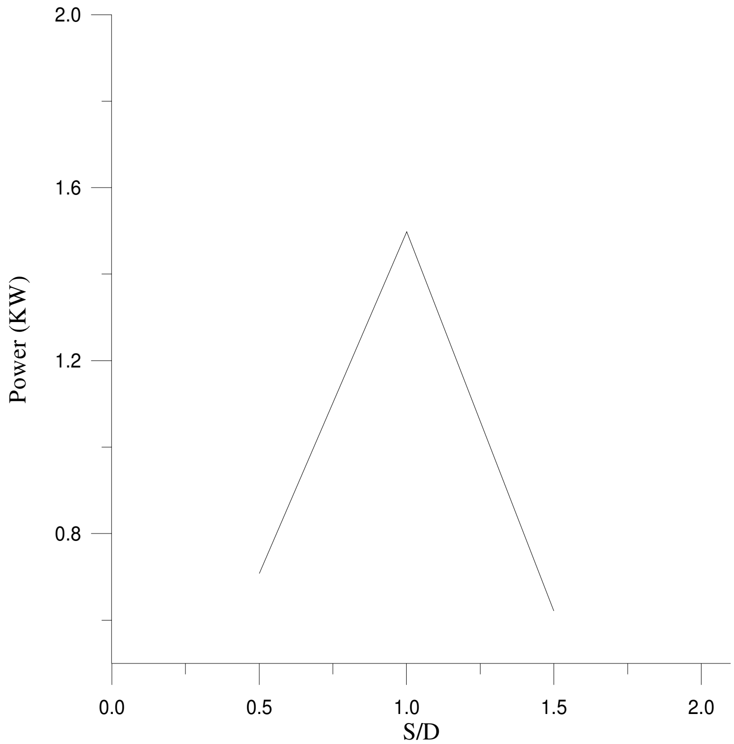

Fig.4.3 Effect of variation S/D on Power using CNG fuel (CH_4)

CONCLUSIONS

1- The results show that, the (S/D) ratio has a significant effect on both turbulence levels and geometric interaction of the flame front with the combustion chamber walls.
2- In general, a square engines (S/D=1) leads to higher power output up to 50% improved.
3- Square engines (S/D=1) generate a higher thermal efficiency through faster burning and lower overall chamber heat loss.
4- The potential of the square engine (S/D=1) for indicate specific fuel consumption was 4% improved, according to output data from the programme the minimum ISFC was for the square engine running by gasoline fuel.
5- The effect of (S/D) on pressure indicate mean effective also limit but vary with type of fuel used, maximum PIME was calculated from gasoline engine then the value 8-10% decrease when engine running by LPG and CNG fuels.
6- The (S/D) ratio has a significant effect on exhaust emission characteristics, exactly on (NO) and (CO) emissions, for (S/D=1) the amount of (NO) 92% decrease when compared with amount of (NO) emission from (S/D=0.5)and (S/D=1.5).
7- The amount of (CO) emission by volume, 10-15% increased slightly when (S/D=1), the amount can be control by install thermal converter in exhaust manifold provides significant reduction in (CO) concentrations on contrast temperature of exhaust increases due to more heat release for (S/D=1) engines.

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