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PREDICTION OF PERFORMANCE CHARACTERISTICS OF A FOUR STROKE CYCLE SPARK IGNITION ENGINE BASED ON COMPUTER MODEL

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Abstract- *This paper presents a SI engine combustion modeling. The past several years have seen a substantial growth in mathematical modeling activities whose interests are to describe the performance efficiency and emissions characteristics of various types of internal combustion engines. The key element in the simulations of various aspect of engine operation is the model of the engine combustion process. Combustion modeling activities help to develop of engine cycle simulations. This model indicate the relationship of engine performance and efficiency and also the relationship between pressure temperature and crank angle by using (P-θ) and (T-θ) diagram. In this model zero dimensional modeling themes is used where first law of thermodynamics is applied to calculate the temperature variation with respect to crank angle and with the help of gas state equation, the pressure variation in the cylinder is calculated. For calculating the rate of change burning with respect to crank angle wiebe function is used. To calculate the heat loss through the cylinder wall Annand's and Woschni's expressions are used for calculating coefficient of heat transfer for inside boundary layer (hi). Turbo C++ language is used for programming the model. The results from the computer model are plotted in the form of different graphs which will indicate the performance of this model. The validation of this model is justified by comparing the result of computer program with real engine data.*

Keywords: SI engine, Performance, Computational Modeling, Turbo C++ language, Real engine

1. INTRODUCTION

An internal combustion engine is the most remarkable achievement of the last 100 years. Today IC engines play a vital role as its uses almost everywhere in every sector. A large portion of total energy in the world is utilized through these engines. Since energy and maintenance costs of these engines stand for a significant percentage of the total operating costs in industries or vehicles so it is essential to operate those engines as efficiently as possible. To achieve these goals various design concepts are attempted.

For the next 80 years developments concern on improving the accuracy with which the thermodynamic properties of the unburned and burned gas mixture could be evaluated. W.J.D. Annand, worked on " Heat transfer in the cylinders of reciprocating internal combustion engines" [1]. He developed an empirical relation of convection heat transfer coefficient to calculate heat loss through the IC engine cylinder wall but it has some restriction because of phase lag between gas temperature change and heat flux variation. J. B. Heywood, worked on "Pollutant formation and control in SI engine [2]. G. Woschni worked on " A universally applicable equation for instantaneous heat transfer coefficient in the IC engine" [3]. His formula was based on the dimensional analysis of convective heat transfer during turbulent flow in pipes. Similarly various study has been accomplished in this aspect to predict engine performance and phenomena. Sherman and Blumberg, worked on " The influence of induction and exhaust process on emissions and fuel consumption in the SI engine". [4]. Hires et al. experimented on "The prediction of ignition delay and combustion intervals for a homogeneous charge SI engine". [5]. R.J. Tabaczynski worked on "Turbulence and turbulent combustion in SI engines" [6]. Lorusso and Tabaczynski worked on " Combustion and emission characteristics of methanol, methanol water and gasoline methanol blends in SI engine" [7] The modeling of engine processes continues to develop as our basic understanding of the physics and chemistry of the phenomena of interest steadily expands and as the capability of the computers to solve complex equations continues to increase. Modeling activities can make major contributions to engine developments in three general areas [8].These combustion come from different stages of any model development process. They are:

1. The development of a more complete understanding of the important physical process that emerges from the requirements of formulating model.
2. The identification of key controlling variables which provide guidelines for more rational and therefore less costly experimental programs and for data reduction and

3. The ability to predict behavior over a wide range of design and operating variables which can be used to screen concepts prior to major hardware programs to determine trends and tradeoffs and if the model is sufficient accurate to optimize design and control.

The three phases of model development which correspond to these contribution are

1. Models development through an analysis of the individual process which are linked together the engine operating cycle.

2. The exploratory use of the model its validation and sensitivity of model predictions to initial assumption.

3. The use of model extensive parametric studies which examine the effect of changes in engine operating and design parameters on performance efficiency and emissions.

Whether a model is ready to pass from one stage to the next depends on the accuracy with which it represents the actual process, the extent to which it has been tested and validated and the time and effort required to use the model for extensive sets of calculations and to interpret the results. Models describe the thermodynamic fluid flow heat transfer combustion and polloutant formation phenomena that govern these performance aspects of engine. The problems of interest in engine combustion modeling can be grouped under three headings:
1. Engine performance and efficiency
2. Engine emission (oxide of nitrogen carbon monoxide odor and noise unburned hydrocarbons etc)
3. Detonation

For prediction of performance of a spark ignition engine firstly a mathematical model has to determine upon which other performance criteria can be determined. The different engine parameters are input to the model and also input the different operating parameters and the model produced he performance therefore the different performance parameters on the basis of those engine and operating parameters. The different operating parameters that can be studied are speed fuel flow rate air fuel ratio exhaust gas fraction (EGR) and starting of combustion. The performance parameters studied are brake horsepower, brake mean effective pressure, brake specific fuel consumption and brake thermal efficiency. As a further work to a more comperensive model the model can be extended to predict the emission form the models based on the combustion modes give in this work.

Various models on IC engine are already existing most of those one CI engine and a fewer on SI engine. The major task of those modeling was to develop a model which was closer to the real engine. But most of those models deviate from the reality because of numerous assumptions that did not satisfy the real engine combustion phenomena and the thermodynamic properties of the unburned and burned gas mixture. However further work on the same model it may come closer to reality. For this reason the aim of this project work is to develop a SI engine model considering such assumptions that satisfy the real engine combustion phenomena and the thermodynamic properties of gas mixture so that it become closer to the real engine.

2. METHODOLOGY

A simulation model is presented for a spark ignition engine for this study. The cycle simulation model presented here requires just one empirical relation for the fraction of the fuel air mixture burnt at any instant. This function (Wiebe's function) is given by the following relation:

$$X(\theta) = 1 - exp[-a(\theta - \theta_0)/\theta_b]^{m+1}$$

Where
θ = crank angle
a and m = are the parameters which depend on the design of the engine. The values of a and m can be varied. Typical values are 5 and 2 respectively.
θ_0= starting angle of combustion
θ_b= total burning duration
X = fraction of the fuel air mixture burnt

The wiebe's function is found to be more consistent with the experimental results and hence it is adopted. The engine parameters which are input to the model are :
1. Engine geometry (bore, stroke, clearance volume and wall thickness)
2. Thermal conductivity of the engine wall
3. Mechanical efficiency of the engine

The model then predicts the brake thermal efficiency, brake specific fuel consumption and brake horsepower of the engine under the operating conditions.

The model is a very simplified model of the processes occurring in the operation of the SI engine. Hence a large number of assumptions are involved in the model

Assumptions of the models:
1. Gas mixture is supposed to behave as ideal gas before and after the combustion. Hence equation of state for ideal gases are used in cycle calculations.
2. Heat loss to the coolant is there throughout the engine calculation but the temperature of the coolant is assumed to be constant at 90^0 through the calculation (constant temperature heat sink)
3. There is no dissociation of different compounds present at any instant in the engine.
4. At any instant fuel is burnt is assumed to have burnt fully to give CO_2 and H_2O only.
5. A single zone combustion model is assumed i.e. the gases in the cylinder are assumed to form homogeneous mixture free from pressure and temperature gradients. The homogeneous mixture at any time is considered to consist of air the products of combustion and recycled products only.
6. Inside film heat transfer coefficient (h_i) is assumed to vary as described by Annand and Woschni.
7. Outside film heat transfer coefficient remains constant throughout
8. Heat loss through the piston is neglected
9. There is no ignition delay and the combustion starts as soon as ignition is initiated by the spark plug
10. Intake and exhaust valves are assumed to open and close instantaneously at TDC or BDC as the case may be i.e. there is no valve opening or closing delay and overlap of the two valves
11. The heat of combustion produced assumed to be

taken up for power heat loss and to increase the temperature at the gases inside only.
12. There is no pumping loop i.e. the work done during exhaust and intake strokes are neglected
13. The properties of gas mixed inside are calculated directly on the basis of gas temperature
14. The closed cycle under consideration consists of real working substance whose mass remains constant throughout the cycle but varies in composition. The working substance has actual specific heats depending on temperature. The closed cycle consist of polytropic compression combustion and expansion of products of combustion.
15. No combination of combustion product takes place to form new products.
16. An increment of 1° crank angle for each iteration is assumed.

Based on these assumptions compression, combustion and expansion stroke, composition of combustion mixture at various stages and heat transfer model are simulated. A few of these simulations and their corresponding equation are demonstrated below.

2.1 Cycle of Events

The cycle of events between air valve closing and exhaust valve opening comprises
1. Compression stroke. 2. Ignition and propagation of combustion. 3. Expansion

2.1.1 Compression Stroke

The compression process starts at the trapped condition . Using a perfect mixing model for fresh charge and residuals from the previous cycle derives the state of the gas at this point. During compression any reaction is neglected.

The first law of thermodynamics is

$dQ = dW = (u_2\text{-}u_1)$

$dQ = pdV + mC_v dt$

$$dQ/_{d\theta} = mC_v\left(dT/_{d\theta}\right) + p\left(dV/_{d\theta}\right) \quad (1)$$

The equation of state is

$$PV = mRT \quad (2)$$

$d(PV)/d\theta = mR\,(dT/d\theta)$

$$V(dP/d\theta) + P\,(dV/d\theta) = mR(dT/d\theta) = R/C_v\ \{(dQ/d\theta) - P\,(dV/d\theta)\} \quad (3)$$

$V(dP/d\theta) = (R/C_v)\ (dQ/d\theta) - (R/C_v)\ P\ (dV/d\theta) - P\ (dV/d\theta)$

$$dP/d\theta = [-\{1+(R/C_v)\}\ P\,(dV/d\theta) + (R/C_v)\,(dQ/d\theta)]/V \quad (4)$$

From equation (3) we get

$mR(dT/d\theta) = V\,(dP/d\theta) + P\,(dV/d\theta)$

$dT/d\theta = (T/PV)\ \{V(dP/d\theta) + P\,(dV/d\theta)\}$

$$dT/d\theta = T\ \{\ (1/P)\,(dP/d\theta) + (1/V)\,(dV/d\theta)\} \quad (5)$$

Where
θ = crank angel
P = pressure inside the cylinder
V = volume of the mixture at a crank angle θ
T = temperature of the mixture

Heat transfer rate from gas to wall can be obtained by Annand's and Woschni equations for convection heat transfer.

For compression dQ = -Q

The work doen is $dW/d\theta = P\,(dV/d\theta)$

As the compression process continues the variable are incremented by using the following general expression

$X_{n+1} = X_n + (dX/d\theta)\,(d\theta)$

2.1.2 Combustion Stage:

As explained in the assumptions a condition at no ignition delay is assumed. As soon as ignition takes place combustion starts. A single zone combustion model is assumed.

The mass friction at the charge within the cylinder which has burned at the given crank angle is specified by the Wiebe's function

$X = 1\text{-}\ \exp[-a(\theta\text{-}\theta_0)/\theta_b]^{\ m+1}$

Where
X = fraction of charge brunt
a = Efficiency parameter
m = form factor
θ = crank angle
θ_0 = ignition crank angle
θ_b = combustion duration

Hence total heat added during the increase of 1^0crank angle

$Q = CV\ x\ (X_n - X_{n-1})\ x\ m_f$

Where
CV = calorific value of fuel
mf = total mass of fuel supplied in one cycle

Therefore temperature at the end of the iteration is

$T_{n+1} = T_n + \{(Q - Q_{loss})/mC_p\ \}$

Where
Q_{loss} = heat loss to the coolant during the crank angle
M = total mass of charge
C_p = specific heat at constant pressure of the charge inside the cylinder

The specific heat at constant pressure (Cp) varies with the temperature and composition of the charge in the cylinder composition at any point during the combustion depend on whether the charge is richer or lean.

For a rich charge the composition is

$X_{CO2} = (x+f_r\ )xF'x\ 0.070175$

$X_{\ H2O} = (x+f_r\ )xF'x\ 0.078947$

$X_{\ Air} = A'\ (1\text{-}x)/29$

$X_{N2} = (x+f_r\ )xA'x\ 0.02741$

$X_{fuel} = \{(1+f_r\ )xF - (f_r + x\ )F'\}/114$

For a lean mixture the composition at any point is

$X_{CO2} = (x+f_r\ )xFx\ 0.070175$

$X_{\ H2O} = (x+f_r\ )xFx\ 0.078947$

$X_{N2} = (x+f_r\ )xA'x\ 0.02741$

$X_{\ Air} = \{(1+f_r\ )xA' - A_{st}(f_r + x\ )\}/29$

$X_{fuel}\ (1\text{-}x)\ x\ F/114$

Where
X_{CO2} = Fraction of mixture by mass that is CO_2
$X_{\ H2O}$ = Fraction of mixture by mass that is H_2O
X_{N2} = Fraction of mixture by mass that is N_2
$X_{\ Air}$ = Fraction of mixture by mass that is air
X_{fuel} = Fraction of mixture by mass that is fuel
f_r = Recycling friction
x = Fraction of the charge that is burnt (given by the wiebe's function)

F' = Stoichiometric fuel flow /cycle
A_{st} = Stoichiometric air flow /cycle
F = Actual amount of fuel/cycle
A' = Actual amount of air /cycle
The pressure under the new condition is given by
$P_{n+1} = (m_a x R x T_{n+1})/V_{n+1}$
$V_{n+1} = V_n + dV$
V = The cylinder volume at any crank position θ
$= \{V_c + (\pi D^2/4)(1+r-s)\}$
Where
Vc = clearance volume
D = Bore diameter
L = Length of connecting rod
R= Crank radius
$S = r \cos\theta + (1^2 - r^2 \sin^2\theta)^{1/2}$
dW = P. dV
$W_{n+1} = W_n + dW$

2.1.3 Expansion Stroke

After the combustion is over the expansion of end products of combustion starts. Composition of the mixture at this stage is assumed to remain constant. The expansion is assumed to be an ideal gas expansion and hence the formula used in the compression stage are used .

The work at the end of the expansion stage gives the work done/cycle by the engine. Hence work done per unit time is
$W_t = (W \text{ x rpm})/(2x60)$
Where
W_t = Work done per unit time
W = Work done per cycle
If η_m = Mechanical efficiency
Then useful work done per unit time
$W_u = W_t + \eta_m$
Therefore
BHP = $W_u/0.736$
i.e.
BSFC = Fuel flow rate / BHP
Brake thermal efficiency $\eta_{th} = W_u$/(fuel flow rate x CV)
Where
CV = calorific value of fuel

2.2 Model for Calculation of Composition of Mixture

The fuel that is most commonly used in SI engine is gasoline or petrol blends of many different hydrocarbon compounds obtained by refining petroleum or crude oil. Hydrocarbon such as paraffin (C_nH_{2n+2}) olefins(C_nH_{2n}) napthenes (C_nH_{2n})and aromatics (C_nH_{2n-6}).
However to simplify chemical calculations octane (C_8H_{18}) is often used as a fair average of commercial gasoline

Properties of Octane:
Molecular weight = 114
Specific gravity = 0.692
Boiling temperature at 1 atm = 99.4^0C
Ignition temperature at 1 atm = 447^0C
Higher calorific value (HCV) = 47813 kj/kg
Lower calorific value (LCV) = 43500kj/kg
Critical compression ratio = 7.3
Air fuel ratio = 15.1

For compete combustion the general reaction can be written as
$C_aH_b + (a+b/4)\ (O_2 + 3.773N_2) = aCO_2 + b/2\ H_2O + 3.773\ (a+b/4)N_2$
If the fuel is octane
$C_8H_{18} + (8+18/4)\ (O_2 + 3.773N_2) = 8CO_2 + 9\ H_2O + 3.773\ (8+18/4)N_2$
$C_8H_{18} + 12.5\ O_2 + 47.1625N_2 = 8CO_2 + 9\ H_2O + 47.1625N_2$
114 400 1320.55 352 162 1320.55
i.e. 114 gm of octane react with (1320.55 gm of N_2 + 400 gm of O_2) = 1720.55 gm of air to produce 352 gm of CO_2, 162 of H_2O and 1320.55 gm of N_2.
Thus for stoichiometric combustion 1 mole of fuel requires 59.66 moles of air and produces 64.16 moles of products. The stoichiometric air fuel ratio $(A/F)_s$ = 1720.55/114 = 15.1 and $(A/F)_s$ =0.066

2.3 Heat Transfer Model

In the design and development of SI engines it is frequently desired to calculate the heat transfer rate from the working fluid to surfaces which contain it. Often an approximate estimate of time average heat transfer rate to the surface as a whole is sufficient but there are many purpose for which a detailed analysis of the variation of the heat transfer rate during cycle is essential. Repeated attempts have been made to provide an empirical formula for the estimation of instantaneous transfer rate; at least ten formulas now exist of which two or three are in current use.

Attention will be confined here to heat transfer occurring during the compression and expansion strokes; that is the charge exchange period is excluded from consideration. During the compression stroke transfer is for all practical purposes entirely convective it is easy to show theoretically that radiation must play a negligible part during this phase of the cycle. During the expansion stroke both convection and radiation may be important and both modes of transfer should be considered.

Let,
h_i = coefficient of heat transfer of the inside boundary layer
h_0 = coefficient of heat transfer of the outside file
k = thermal conductivity of the walls of the cylinder
A_i = total inside heat transfer area
A_0 = total outside heat transfer area
A = mean area of the heat transfer for conduction
T = temperature of the gas
T_1 = temperature at the inside wall
T_2 = temperature at the outside wall
T_w = temperature at the coolant
T = thickness at the wall of the cylinder
r_0 = outer radius of the cylinder
r_i = inner radius of the cylinder
V = volume enclosed at any moment

If Q is the rate of heat transfer then
$Q = h_i \text{ x } A_i \text{ x } (T - T_1)$ (6)
$Q = \{(k \text{ x } A)/T\} \text{ x } (T_1 - T_2)$ (7)
$Q = h_0 \text{ x } A_0 \text{ x } (T_2 - T_w)$ (8)
A_i = area of inside cylinder heat + area of inside cylinder

wall

$= \pi r_i^2 + 2\pi r_i L$ where L = length of cylinder at any instant

$= \pi r_i^2 + 2r_i (\pi r_i^2 L/r_i^2)$

$$A_i = \pi r_i^2 + 2r_i (V/r_i^2) \quad (9)$$

A_0 = Area of outside cylinder head + area of outside cylinder wall

$= \pi r_0^2 + 2r_0 x(\pi r_i^2 L/r_i^2)$

$$= \pi r_0^2 + 2r_0 x(V/r_i^2) \quad (10)$$

$$A = \pi r_i^2 + 2(V/r_i^2)x[(r_0-r_i)/In(r_0/r_i)] \quad (11)$$

From equation (7) and (8)

$(kA)(T_1 - T_2) = h_0 A_0 x t(T_2 - T_w)$

$kAT_1 - kAT_2 = h_0 A_0 tT_2 - h_0 A_0 tT_w$

$$T_2 = (kAT_1 + h_0 A_0 tT_w) - (kA + h_0 A_0 t) \quad (12)$$

From equation (6), (8) and (12)

$h_i A_i (T - T_1) = h_0 A_0 (T_2 - T_w)$

$= h_0 A_0[\{(kAT_1/t + h_0 A_0 T_w\}/\{(kA/t) + h_0 A_0)\} - T_w]$

$T_1 = [h_i A_i \{(kA/t) + h_0 A_0\}T + \{T_w (kA/t) h_0 A_0 + (kA/t) h_i A_i + h_0 A_0 h_i A_i]$

Now

$Q_{loss} = h_i A_i (T - T_1)$

By using the value of T_1 we get

$Q_{loss} = h_i A_i [(kA/t)(T - T_w) h_0 A_0]/[(kA/t) h_0 A_0 + \{(kA/t) h_0 A_0\} h_i A_i]$

i.e.

$Q_{loss} = B(T - T_w)/C$

Where

$B = (kA/t) h_0 A_0$

$C = [(kA/t) h_0 A_0 + \{(kA/t) h_0 A_0\}] h_i A_i$

3. RESULTS AND DISCUSSION

In this project Turbo C++ language is used for programming the model. The results from the computer model are plotted in the form of different graphs which indicated the performance of a SI engine. The outputs from this model are BHP, BSFC, BTEFF pressure and temperature at various crank angles. For checking the validation of the model the output of a real SI engine is compared with the outputs of the model using same specification of that real engine. Various graphs are plotted such as BHP, BSFC, BTEFF corresponding to various fuel flow rate and speed.

From the comparison, it is evident that performances curves are almost similar at higher speed but real engine produces little bit poor performance compare to model at lower speed. In case of Annand heat transfer equation at 2900 rpm and 0.00098 kg/sec for real engine BHP is equal to 10.32835 and $(BHP)_{model}$ = 10.9979 hence the percentage of error of the model with respect to the real engine is 6.0879%. While at 2100 rpm and fuel flow rate of 0.00086 kg/sec the error is 13.42% and at 1200 rpm and 0.00074kg/sec the error is 33.83%. The value of BHP of the model is always greater than that of the real engine of the same condition (This difference is smaller at high speed but grater at low speed) as it is assumed that gas mixture is ideal gas before and after the combustion but in real case it is not true, it also assumed that the coolant temperature is constant no dissociation of different compound complete burnt of fuel no heat loss through the piston no ignition delay no pumping loop i.e. the work done during exhaust and intake strokes are neglected but all this are not valid for the real engine. It also can be stated that the real engine performance is lower at lower speeds due to incomplete combustions. Again figure 4, figure 5 and figure 6 are drawn based on the result found by using Woschni equation. In Woschni heat transfer equation the radiation term is not separately used as model assumed that the radiation term is included in his equation. It is seen that if a separate radiation term is added with Woschni heat transfer equation then the BHP at 2900 rmp become 10.1632 which is lower than the real model so it can be said that claimed done by Woschni heat transfer equation is correct. From the comparison curves it is seen that for same operating conditions error of the model at 2900rpm is 7.4% and at 1200 rpm is 35.5%. With this result it is clear that using of Annand heat transfer equation is more accurate than that of Woschni.

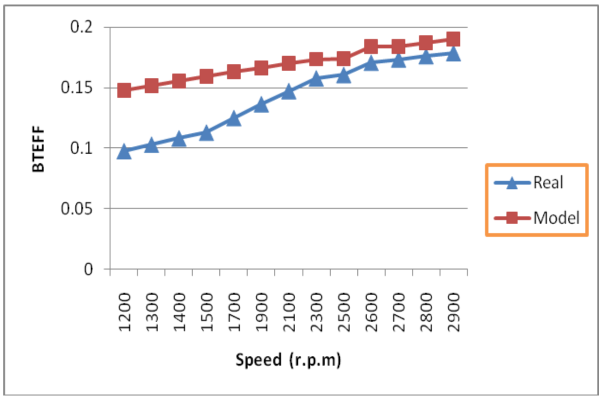

Fig. 1 BTEFF vs Speed curve (using Anand equation)

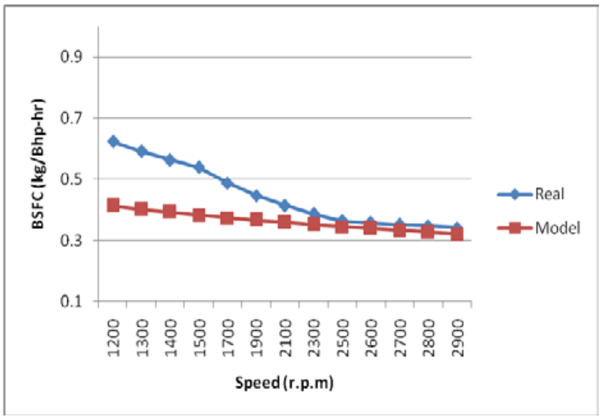

Fig. 2 BSFC vs Speed curve (using Annand equation)

4. CONCLUSION

The models outline in this study can be used satisfactorily to predict the performance of a four stroke SI engine. From the comparison of models and real engine, it is evident that they are closer at higher speed and spread away at lower speed.

In case of Annand heat transfer equation at 2900 rpm and 0.00098 kg/sec for real engine BHP = 10.32835 and model BHP = 10.9979, hence, the percentage of error of the model with respect to the real engine is 6.0879%. When speed decreased the percentage of error is also increase. At 2100 rpm and fuel flow rate 0.00086 kg/sec the error is 13.42% and at 1200 rpm and 0.00074kg/sec the error is 33.83%.

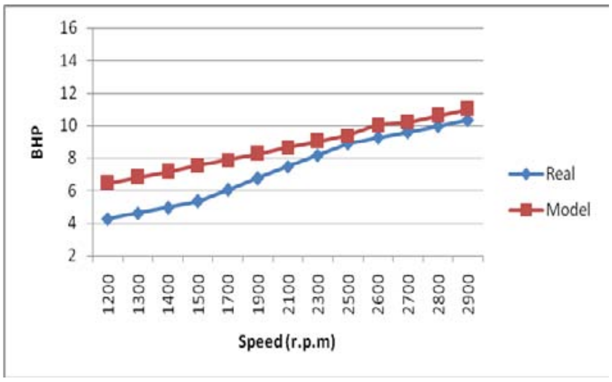

Fig. 3 BHP vs Speed curve(using Annand equation)

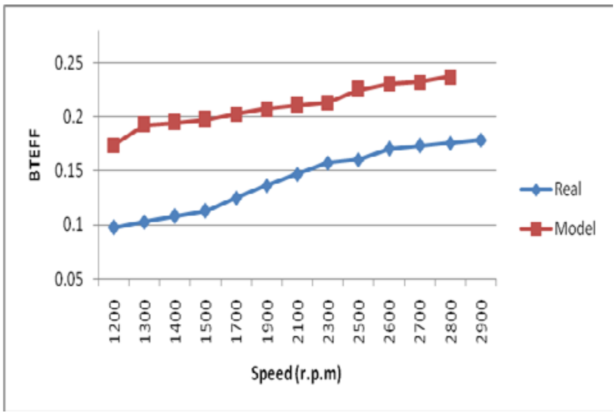

Fig. 4 BTEFF vs Speed curve (using Woschni equation)

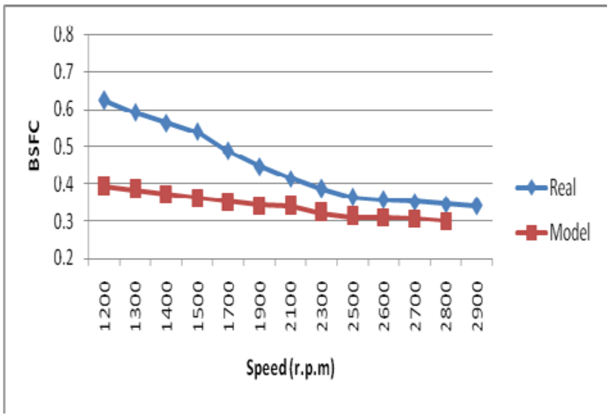

Fig. 5 BSFC vs Speed curve (using Woschni equation)

On the other hand from the comparison of Woschni models with real engine comparison curves, it is seen that for same operating conditions error of the model at 2900rpm is 7.4% and at 1200 rpm is 35.5%. With this result it is clear that using of Annand heat transfer equation is more accurate than that of Woschni.

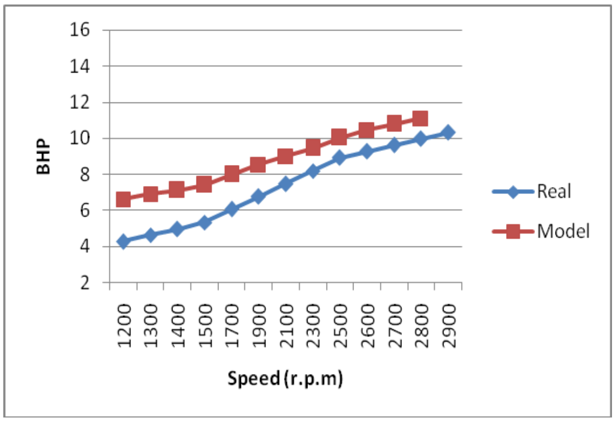

Fig. 6 BHP vs Speed curve (using Woschni equation)

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