

**COMPUTATIONAL EFFECT OF SPEED AND AIR-FUEL
RATIO ON PERFORMANCE OF A TURBOCHARGED FOUR
CYLINDER FOUR STROKE CYCLE DIRECT INJECTION
DIESEL ENGINE USING HYDROGEN AND DIESEL AS
ALTERNATIVE FUELS**

M. Marouf Wani*

ABSTRACT

This paper describes the results of computational studies on a turbocharged four cylinder four stroke cycle direct injection diesel engine using Hydrogen as an alternative fuel to diesel. The simulation is done in the professional engine simulation software from AVL company of Austria named as BOOST. The modeling methodology involves the use of first law of thermodynamics and Navier-Stokes equations for engine cylinder and manifolds. The design parameters are fixed by engine geometry. A matrix was prepared for the operating variables to carry out the simulation. First the data was used as per diesel engine needs and results for engine performance were generated. The operation was revised with data for proposed hydrogen engine and its performance was studied. The software gave successful results in both the cases. It was observed that the power output and brake specific fuel consumption were improved with hydrogen as an alternative fuel due to its higher calorific value. It is proposed that hydrogen can be successfully used in diesel engine using pilot injection of diesel towards the end of the compression stroke or else by using a spark plug in diesel engine. It is proposed that only air will travel through compressor and intake manifold and liquefied hydrogen will be injected as per norms in multi point fuel injection systems using intake port for injection.

Keywords: Turbo-charged diesel engine, hydrogen, diesel, performance

* Mechanical Engineering Department, National Institute of Technology, Hazratbal, Srinagar, INDIA.

INTRODUCTION

Computer simulation studies helps to predict the behavior of the engine in the diesel and hydrogen fuel modes. We prepare the models for Hydrogen and Diesel direct injection modes for the engine systems and feed the actual data corresponding to the design and operating conditions of the system. It helps to simulate the results without actually performing experiments. Thus a lot of money and time is saved. Moreover we can simulate and compute those results which are very difficult to be measured experimentally. Favorable computed results pave the way for further experimental investigations.

The objectives are to investigate the feasibility of Hydrogen as future alternative fuel in diesel engines. It is also intended to choose hydrogen as fuel because it has no carbon atoms and least hydrogen atoms as compared to diesel and therefore will produce zero or minimal pollution. Table 1 at the end gives physico-chemical properties of hydrogen and diesel which help us to investigate the feasibility of using hydrogen as an alternative fuel to diesel.

THEORETICAL BASIS

The theoretical background including the basic equations for all elements used in the present model is summarized below to give a better understanding of the program.

The Cylinder , High Pressure Cycle, Basic Equation.

The calculation of the high pressure cycle of an internal combustion engine is based on the first law of thermodynamics:

$$\frac{d(m_c.u)}{d\alpha} = -\frac{p_c.dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_w}{d\alpha} - \frac{h_{BB}.dm_{BB}}{d\alpha} \text{-----(Eq.1)}$$

where

$$\frac{d(m_c.u)}{d\alpha} = \text{change of the internal energy in the cylinder.}$$

$$-\frac{p_c.dV}{d\alpha} = \text{piston work.}$$

$$\frac{dQ_F}{d\alpha} = \text{fuel heat input.}$$

$$\sum \frac{dQ_w}{d\alpha} = \text{wall heat losses}$$

$\frac{h_{BB}.dm_{BB}}{d\alpha}$ = enthalpy flow due to blow-by

$\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow

The first law of thermodynamics for high pressure cycle states that the change of internal energy in the cylinder is equal to the sum of piston work, fuel heat input, wall heat losses and the enthalpy flow due to blow-by.

Eq.1 is valid for engines with internal and external mixture preparation. However the terms which take into account the change of gas composition due to combustion, are treated differently for internal and external mixture preparation.

For internal mixture preparation it is assumed that

- The fuel added to the cylinder charge is immediately burnt.
- The combustion products mix instantaneously with the rest of cylinder charge and thus form a uniform mixture.
- As a consequence, the Air-Fuel ratio of the charge diminishes continuously from a high value at the start of combustion to the final value at the end of combustion.

In order to solve this equation, models for the combustion process and the wall heat transfer, as well as the gas properties as a function of pressure, temperature, and gas composition are required.

Together with the gas equation

$$p_c = \frac{1}{V} .m_c.R_o.T_c \text{ -----(Eq.2)}$$

Establishing the relation between pressure, temperature and density, Eq. 2 for in-cylinder temperature can be solved using a Runge-Kutta method. Once the cylinder gas temperature is known, the cylinder gas pressure can be obtained from the gas equation.

Combustion Model

The following equation for the stoichiometric air requirement specifies how much air is required for a complete combustion of 1 kg fuel:

$$L_{st} = 137.85 .\left(\frac{c}{12.01} + \frac{h}{4.032} + \frac{s}{32.06} - \frac{o}{32.0}\right) \text{ [kg Air/kg Fuel] -----(Eq.3)}$$

For lean combustion, the total heat supplied during the cycle can be calculated from the amount of fuel in the cylinder and the lower heating value of the fuel. The lower heating value is a fuel property and can be calculated from the following formula:

$$H_u = 34835 \cdot c + 93870 \cdot h + 6280 \cdot n + 10465 \cdot s - 10800 \cdot o - 2440 \cdot w \text{ [kJ/kg]} \text{ -----(Eq.4)}$$

In rich air fuel mixture combustion, the total heat supplied during the cycle is limited by the amount of air in the cylinder. The fuel is totally converted to combustion products even if the amount of air available is less than the amount of stoichiometric air.

Heat Release Approach.

The vibe function is used to approximate the actual heat release characteristics of an engine:

$$\frac{dx}{d\alpha} = \frac{a}{\Delta\alpha_c} \cdot (m+1) \cdot y^m \cdot e^{-a \cdot y^{(m+1)}} \text{ -----(Eq.5)}$$

$$dx = \frac{dQ}{Q} \text{ -----(Eq.6)}$$

$$y = a \cdot \frac{\alpha_0}{\Delta\alpha_c} \text{ -----(Eq.7)}$$

The integral of the vibe function gives the fraction of the fuel mass which was burned since the start of combustion:

$$x = \int \left(\frac{dx}{d\alpha} \cdot d\alpha \right) = 1 - e^{-a \cdot y^{(m+1)}} \text{ -----(Eq.8)}$$

Gas Exchange Process , Basic Equation

The equation for the simulation of the gas exchange process is also the first law of thermodynamics:

$$\frac{d(m_c \cdot u)}{d\alpha} = - \frac{p_c \cdot dV}{d\alpha} - \sum \frac{dQ_w}{d\alpha} + \sum \frac{dm_i}{d\alpha \cdot h_i} - \sum \frac{dm_e}{d\alpha \cdot h_e} \text{ -----(Eq.9)}$$

The variation of the mass in the cylinder can be calculated from the sum of the in-flowing and out-flowing masses:

$$\frac{dm_c}{d\alpha} = \sum \frac{dm_i}{d\alpha} - \sum \frac{dm_e}{d\alpha} \text{ -----(Eq.10)}$$

Piston Motion

Piston motion applies to both the high pressure cycle and the gas exchange process.

For a standard crank train the piston motion as a function of the crank angle α can be written as:

$$s = (r+l) \cdot \cos\psi - r \cdot \cos(\psi + \alpha) - l \cdot \sqrt{1 - \left\{ \frac{r}{l} \cdot \sin(\psi + \alpha) - \frac{e}{l} \right\}^2} \quad \text{-----(Eq.11)}$$

$$\psi = \arcsin\left(\frac{e}{r+l}\right) \quad \text{-----(Eq.12)}$$

Heat Transfer

The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from:

$$Q_{wi} = A_i \cdot \alpha_w \cdot (T_c - T_{wi}) \quad \text{-----(Eq.13)}$$

In the case of the liner wall temperature, the axial temperature variation between the piston TDC and BDC position is taken into account:

$$T_L = T_{L,TDC} \cdot \frac{1 - e^{-cx}}{x \cdot c} \quad \text{-----(Eq.14)}$$

$$c = \ln\left\{ \frac{T_{L,TDC}}{T_{L,BDC}} \right\} \quad \text{-----(Eq.15)}$$

For the calculation of the heat transfer coefficient, the Woschni 1978 heat transfer model is used.

Woschni Model

The woschni model published in 1978 for the high pressure cycle is summarized as follows:

$$\alpha_w = 130 \cdot D^{-0.2} \cdot p_c^{0.8} \cdot T_c^{-0.53} \cdot \left[C_1 \cdot c_m + C_2 \cdot \frac{V_D \cdot T_{c,1}}{p_{c,1} \cdot V_{c,1}} \cdot (p_c - p_{c,o}) \right]^{0.8} \quad \text{-----(Eq.16)}$$

$$C_1 = 2.28 + 0.308 \cdot c_u / c_m$$

$$C_2 = 0.00324 \text{ for DI engines}$$

$$C_2 = 0.00622 \text{ for IDI engines}$$

For the gas exchange process, the heat transfer coefficient is given by following equation:

$$\alpha_w = 130 \cdot D^{-0.2} \cdot p_c^{0.8} \cdot T_c^{-0.53} \cdot (C_3 \cdot c_m)^{0.8} \quad \text{-----(Eq.17)}$$

$$C_3 = 6.18 + 0.417 \cdot c_u / c_m$$

Fuel Injector

The fuel injector model is based on the calculation algorithm of the flow restriction. This means that the air flow rate in the fuel injector depends on the pressure difference across the injector and is calculated using the specified flow coefficients. In addition, the amount of fuel specified is fed into the air flow.

In the case of carburetor model, the fuel flow is set to a specified percentage of the instantaneous mass flow.

For the injector model, a measuring point must be specified at the location of the air flow meter. In this case the mean air flow at the air flow meter location during the last complete cycle is used to determine the amount of fuel. As is the case for continuous fuel injection, the fuelling rate is constant over crank angle.

The fuel is added in gaseous form to the pipe flow. No evaporation is considered.

Pipe Flow

The one dimensional gas dynamics in a pipe are described by the continuity equation

$$\frac{\partial \rho}{\partial t} = -\frac{\partial(\rho.u)}{\partial x} - \rho.u.\frac{1}{A}\frac{dA}{dx}, \text{-----(Eq.18)}$$

the equation for the conservation of momentum

$$\frac{\partial(\rho.u)}{\partial t} = -\frac{\partial(\rho.u^2 + p)}{\partial x} - \rho.u^2.\frac{1}{A}\frac{dA}{dx} - \frac{F_R}{V}, \text{-----(Eq.19)}$$

and by the energy equation

$$\frac{\partial E}{\partial t} = -\frac{\partial[u.(E + p)]}{\partial x} - u.(E + p).\frac{1}{A}\frac{dA}{dx} + \frac{q_w}{V}. \text{-----(Eq.20)}$$

The wall friction force can be determined from the wall friction factor λ_f :

$$\frac{FR}{V} = \frac{\lambda_f}{2.D}.\rho.u.|u| \text{-----(Eq.21)}$$

Using the Reynold's analogy, the wall heat flow in the pipe can be calculated from the friction force and the difference between wall temperature and gas temperature:

$$\frac{q_w}{V} = \frac{\lambda_f}{2.D}.\rho.|u|.c_p.(T_w - T) \text{-----(Eq.22)}$$

During the course of numerical integration of the conservation laws defined in the Eq.20, Eq.21 and Eq.22, special attention should be focused on the control of the time step. In order to achieve a stable solution, the CFL criterion (stability criterion defined by Courant, Friedrichs and Lewy) must be met:

$$\Delta t \leq \frac{\Delta x}{u + a} \text{-----(Eq.23)}$$

This means that a certain relation between the time step and the lengths of the cells must be met. The time step to cell size relation is determined at the beginning of the calculation on the basis of

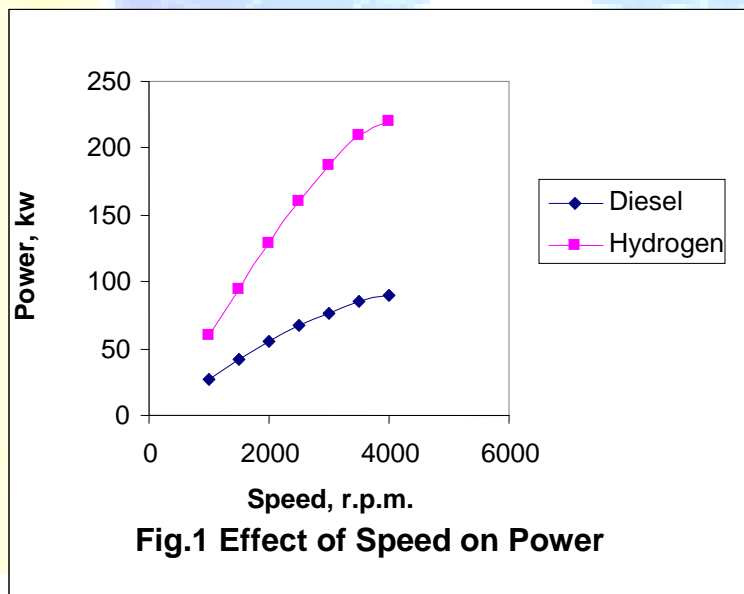
the specified initial conditions in the pipes. However, the CFL criterion is checked every time step during the calculation. If the criterion is not met because of significantly changed flow conditions in the pipes, the time step is reduced automatically.

An ENO scheme is used for the solution of the set of non-linear differential equations discussed above. The ENO scheme is based on a finite volume approach. This means that the solution at the end of the time step is obtained from the value at the beginning of the time step and from the fluxes over the cell borders.

RESULTS AND DISCUSSION

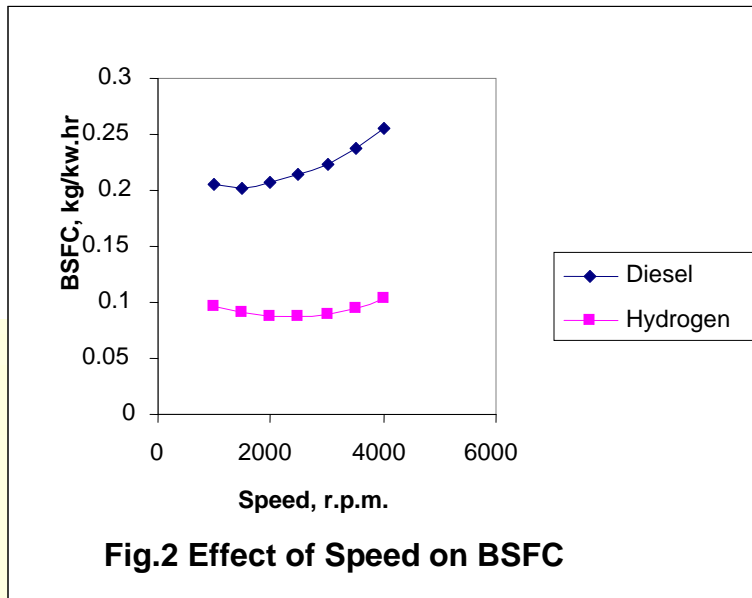
Effect of Speed on Power

The Fig.1 below shows the effect of speed on power. It is seen as the speed increases the power also increases due to more number of power cycles per unit time. Further it is seen that the power developed by hydrogen as fuel is much higher than diesel due to higher calorific value of hydrogen.



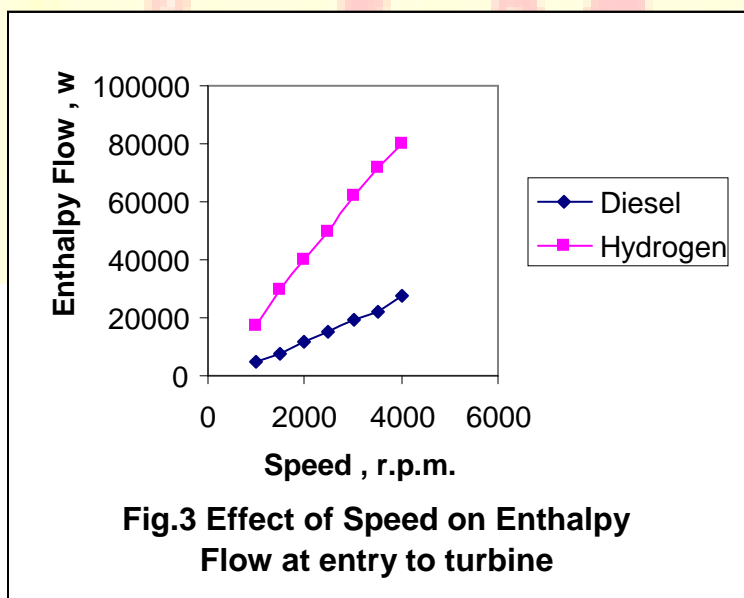
Effect of Speed on Brake Specific fuel Consumption.

Fig.2 below shows the effect of speed on BSFC (brake specific fuel consumption). It is seen that the operation of hydrogen engine is more economical as the BSFC of hydrogen fuel is less than that of diesel. It is due to higher calorific value of hydrogen.



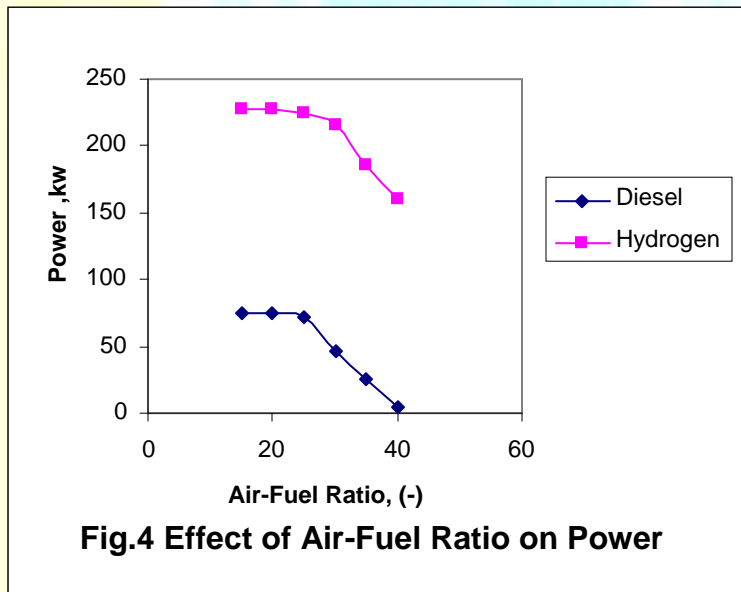
Effect of Speed on Enthalpy Flow at entry to turbine.

Fig.3 below shows the effect of speed on enthalpy flow at entry to turbine in the selected turbocharged engine in cases of hydrogen and diesel as alternate fuels. Enthalpy flow at entry to turbine in case of hydrogen engine is higher due to better combustion and higher rate of heat released during combustion in cylinders. This will obviously result in larger enthalpy drop across the turbine in case of hydrogen fuel with the result more power can be recovered to run the compressor through turbine.



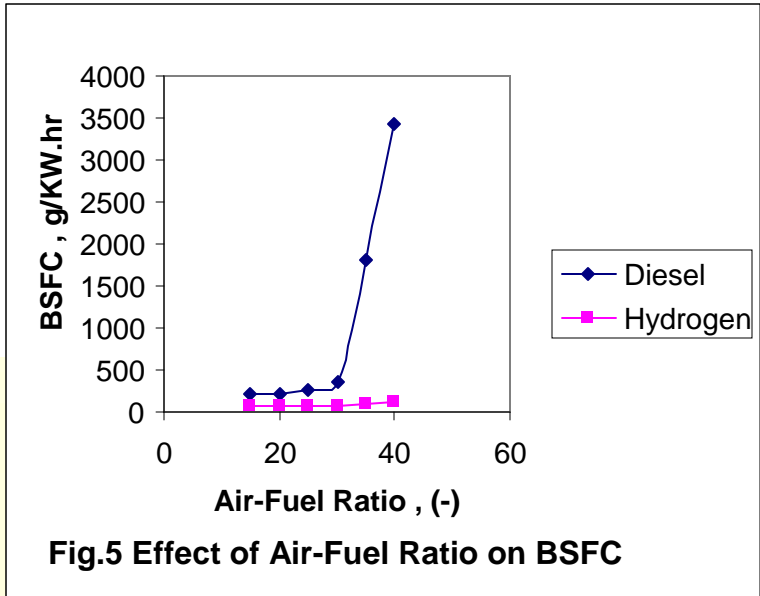
Effect Of Air-Fuel Ratio On Power

The Fig.4 shows the effect of Air-Fuel ratio on power. It is seen as the the air-fuel ratio increases the power output decreases as the mixture becomes leaner. Further it is seen that the power developed by hydrogen as fuel is much higher than diesel due to higher calorific value of hydrogen. Highest power is developed at an air-fuel ratio of about 20 in both cases. This is true for diesel , as diesel engines are operated on this air-fuel ratio for maximum power generation. The air-fuel ratio for diesel engines cannot be decreased further as it produces lot of smoke due to heterogeneous combustion. It is seen that hydrogen engines can also be operated on an air-fuel ratio of about 20 for maximum power generation.



Effect Of Air-Fuel Ratio On Brake Specific Fuel Consumption.(BSFC)

Fig.5 shows the effect of air-fuel ratio on BSFC (brake specific fuel consumption). It is seen that the operation of hydrogen engine is more economical as the BSFC of hydrogen fuel is less than that of diesel. It is due to higher calorific value of hydrogen. Further hydrogen engine remains economical over a wide range of air-fuel ratio. This feature will make hydrogen a more suitable fuel for automotives and power generators over a wide range of speed and load.



Effect Of Air-Fuel Ratio On Enthalpy Flow At Entry To Turbine

Fig.6 shows the effect of air-fuel ratio on enthalpy flow at entry to turbine in the selected turbocharged engine in cases of hydrogen and diesel as alternate fuels. Enthalpy flow at entry to turbine in case of hydrogen engine is higher due to better combustion and higher rate of heat released during combustion in cylinders. This will obviously result in larger enthalpy drop across the turbine in case of hydrogen fuel with the result more power can be recovered to run the compressor through turbine. This will also help to increase the power and thermal efficiency of hydrogen engine through turbo-compounding.

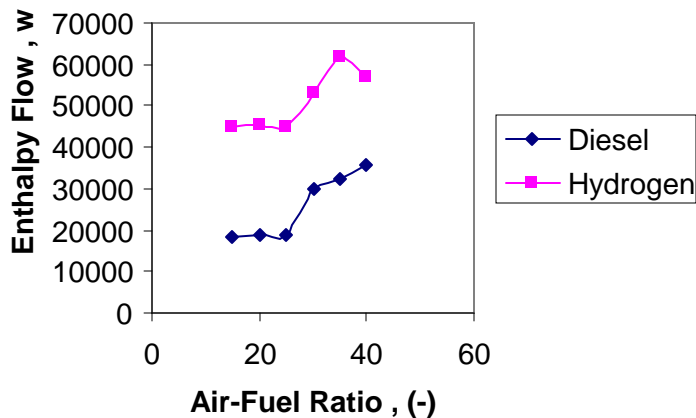


Fig.6 Effect of Air-Fuel Ratio on Enthalpy Flow at entry to turbine

CONCLUSIONS

- Hydrogen can safely be used in diesel engines as an alternative fuel to conventional diesel.
- Pollution formation from engines using Hydrogen as fuel will be minimal due to no carbon atoms in hydrogen fuel.
- Power is increased by using Hydrogen as fuel in diesel engines.
- The Hydrogen engine operation is economical as the brake specific fuel consumption with Hydrogen is less as compared to diesel.

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- [2] Richard L. Bechtold , Alternative Fuels Handbook , SAE Publication , P.73

APPENDIX-A

NOMENCLATURE

a	=	speed of sound
A	=	pipe cross-section
A _{eff}	=	effective flow area
A _i	=	surface area (cylinder head, piston, liner)
AF _{CP}	=	air fuel ratio of combustion products
A _{geo}	=	geometrical flow area
c	=	mass fraction of carbon in the fuel
c _v	=	specific heat at constant volume
c _p	=	specific heat at constant pressure
C1	=	2.28+0.308.cu/cm
C2	=	0.00324 for DI engines
C2	=	0.00622 for IDI engines
C _m	=	mean piston speed
C _u	=	circumferential velocity
c _u	=	circumferential velocity
D	=	cylinder bore
D	=	pipe diameter
dm _i	=	mass element flowing into the cylinder
dm _e	=	mass element flowing out of the cylinder
d _{vi}	=	inner valve seat diameter (reference diameter)
$\frac{dm_{BB}}{d\alpha}$	=	blow-by mass flow
e	=	piston pin offset
E	=	energy content of the gas ($=\rho \cdot c_v \cdot T + \frac{1}{2} \cdot \rho \cdot u^2$)
f	=	fraction of evaporation heat from the cylinder charge
F _R	=	wall friction force
h	=	mass fraction of hydrogen in the fuel
h _{BB}	=	enthalpy of blow-by

h_i	=	enthalpy of in-flowing mass
h_e	=	enthalpy of the mass leaving the cylinder
H_u	=	lower heating value
k	=	ratio of specific heats
l	=	con-rod length
m	=	shape factor
\dot{m}	=	mass flow rate
m_c	=	mass in the cylinder
m_{ev}	=	evaporating fuel
m_{pl}	=	mass in the plenum
n	=	mass fraction of nitrogen in the fuel
o	=	mass fraction of oxygen in the fuel
p	=	static pressure
P_{01}	=	upstream stagnation pressure
$P_{c,o}$	=	cylinder pressure of the motored engine[bar]
$P_{c,l}$	=	pressure in the cylinder at IVC[bar]
p_{pl}	=	pressure in the plenum
p_c	=	cylinder pressure
p_2	=	downstream static pressure
q_{ev}	=	evaporation heat of the fuel
q_w	=	wall heat flow
Q	=	total fuel heat input
Q_F	=	fuel energy
Q_{wi}	=	wall heat flow (cylinder head, piston, liner)
r	=	crank radius
R_0	=	gas constant
s	=	piston distance from TDC
t	=	time
T	=	temperature
$T_{c,l}$	=	temperature in the cylinder at intake valve closing (IVC)
T_c	=	gas temperature in the cylinder

T_{wi}	=	wall temperature (cylinder head, piston, liner)
T_L	=	liner temperature
$T_{L,TDC}$	=	liner temperature at TDC position
$T_{L,BDC}$	=	liner temperature at BDC position
T_w	=	pipe wall temperature
T_{01}	=	upstream stagnation temperature
u	=	specific internal energy
u	=	flow velocity
V	=	cylinder volume
V	=	cell volume ($A \cdot dx$)
VD	=	displacement per cylinder
w	=	mass fraction of water in the fuel
x	=	relative stroke (actual piston position related to full stroke)
x	=	coordinate along the pipe axis
α	=	crank angle
α_o	=	start of combustion
$\Delta\alpha_c$	=	combustion duration
α_w	=	heat transfer coefficient
ρ	=	density
$\mu\sigma$	=	flow coefficient of the port
ψ	=	crank angle between vertical crank position and piston TDC position
λf	=	wall friction coefficient
Δt	=	time step
Δx	=	cell length

APPENDIX-B

Turbo charged Diesel Engine Specifications	
Bore	90 mm
Stroke	98 mm
Compression Ratio	19
Number of Cylinders	4

APPENDIX-C

Table 1: Physico-Chemical Properties of Hydrogen and Diesel. [2]

Fuel Property	Hydrogen	Diesel
Formula	H ₂	C ₈ to c ₂₅
Molecular weight	2.02	200 approx.
Composition, weight%		
Carbon	0	84-87
Hydrogen	100	13-16
Oxygen	0	0
Freezing point	-275	-40 to -1
Lower heating value, Kj/g	119.93	42.5
Flammability limits, vol%		
Lower		
Higher	49	1.0
	75	6.0
Stoichiometric air-fuel ratio, weight	34.3	14.7
Cetane number	-	40-55