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A MODELING STUDY OF THE EFFECT OF HYDROGEN ADDITION ON THE PERFORMANCE OF A MEDIUM SPEED FOUR STROKE DIESEL ENGINE

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ABSTRACT

In the present work, the effect of hydrogen addition on the performance of a diesel engine has been studied. A quasi-dimensional multizone combustion model to simulate a four-stroke cycle of a diesel engine fueled with hydrogen-diesel mixture is developed. It is found that the addition of (12)% by mass of hydrogen causes an increase in the maximum (pressure, zonal temperature, and rate of heat release). This is due to increase in the rate of fuel mass burning, therefore, the time required to complete the combustion is reduced. In general, the addition of (10)% of hydrogen by mass gives a maximum improvement in the power and efficiency and a maximum reduction in specific fuel consumption (60)%.

الخلاصة

تضمن هذا البحث دراسة تاثير أضافة الهايدروجين بنسب وزنية مختلفة الى وقود الديزل على أداء محرك ديزل. تم تطوير موديل احتراق شبه بعدي متعدد المناطق لنمذجة عملية الاحتراق في محرك ديزل رباعي الاشواط يعمل بخليط من وقود الديزل والهايدروجين. لقد وجد ان اضافة(١٢) ٪ من الهيدروجين ادت الى رفع (قيم الضغط، درجة الحرارة لكل منطقة والطاقة المتحررة) العظمى وهذا بسبب زيادة في المعدل الكتلي لحرق الوقود ولهذا يقل الوقت اللازم لانهاء عملية الاحتراق. وبشكل عام، ان اضافة(١٠) % من الهيدروجين سببت زيادة في القدرة والكفاءة وسببت انخفاضا في معدل استهلاك الوقود بنسبة (٦٠)%.

DESCIPTION OF THE PRESENT WORK

Diesel engine simulation models can be classified into three categories, zero-dimensional single zone models, quasi-dimensional multizone models (which is used in the present study), and multi-dimensional model. A computer program has been written to simulate the compression, combustion and expansion processes in direct injection diesel engine (see table (2). Spray zones are formed as a result of fuel injection in which these zones are distributed in the axial and radial direction. The first law of thermodynamics, equation of state, and conservation of mass, energy, momentum, and volume were applied to each zone. A system of first order ordinary differential equations was solved using Runge Kutta method to obtain cylinder pressure, zonal temperature, and heat transfer. These equations were solved under the following assumptions:

1. The fuel injected into the chamber is initially assumed to form a liquid column that travels at a speed equal to the fuel injection speed until the fuel break-up time elapses. After that, the injected fuel is distributed within a spray as parcels. The amount of in each parcel is unique to each spray parcel and varies from one time step to another depending on the conditions.

2. The zones in a parcel are assumed to contain equal mass of fuel, however the amount of fuel in the zones may vary according to the amount of fuel contained within each parcel of fuel.

3. No mixing or heat transfer among zones is permitted.

4. Following breakup, it is assume the fuel spray atomized to fine droplets, each with a diameter equal to the Sauter mean diameter (SMD).

5. The effect of droplet size distribution in a spray parcel is neglected.

6. The flow leakage through the valves, crevices and past the piston rings is neglected.

7. The spray impingement and air swirl are neglected.

8. The effect of dissociation is accounted for.

The total number of zones in the radial direction is fixed regardless of the amount of fuel injected or the time which is limited by cone angle. However, the total number of zones in the axial direction equals the number of spray parcels and therefore can be determined by the injection duration and the computational time step size. The total number of the zones used in the present work is 200, and due to the symmetry about spray axis only half spray was considered. **Fig.(1)** shows the division of spray into parcels and zones at a certain instance.

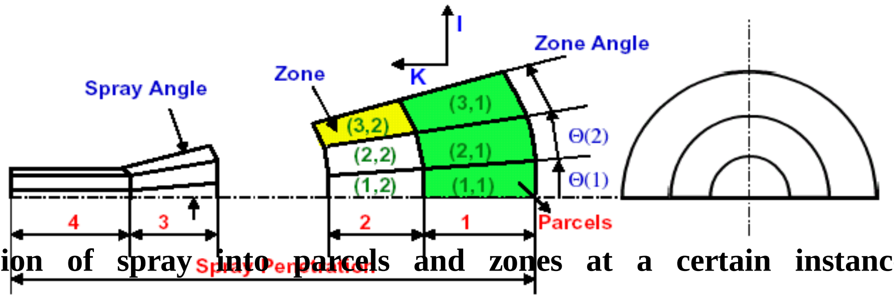


Fig. (1) Division of spray into parcels and zones at a certain instance (Dennis,1999).

HEAT TRANSFER MODEL

The total heat transfer is the sum of the heat transfer to the cylinder wall via convection and radiation and heat transfer to the liquid fuel droplets during the evaporation process. Therefore, the heat transfer rate term $\frac{dQ}{dt}$ is expressed as **(Jung,2001);**

$$\frac{dQ_i}{dt} = -\frac{dQ_{Convection}}{dt} \cdot \frac{m_i T_i}{\sum_{i=zones} m_i T_i} - \frac{dQ_{Radiation}}{dt} \cdot \frac{m_i T_i^4}{\sum_{i=zones} m_i T_i^4} - q_i \frac{m_i T_i}{m_m T_m + m_i T_i} \quad (1)$$

The convective heat transfer from the gas to cylinder wall expressed as;

$$\frac{dQ_{Convection}}{dt} = h_C . A . \left(T_g - T_w \right) \quad (2)$$

Where h_C heat transfer coefficient [W/m^2.^{o}K].

For convection term **(Woschni,1967)** developed the following empirical correlation;

$$h_C = 3.26 B^{-0.2} . T^{-.55} . P^{0.8} . w^{0.8} \quad (3)$$

During the compression stroke, **(Woschni,1967)** argued that the average gas velocity should be proportional to the mean piston speed. During combustion and expansion he attempted to account directly for the gas velocities induced by the change in density that results from combustion. The following expression is used;

$$w = \left[C_1 u_p + C_2 \left(\frac{V_{disp}}{V_{cyl}} \right) . \left(\frac{P(\theta) - P_{motor}(\theta)}{P_{cyl}} \right) . T \right] \text{ [m/s]} \quad (4)$$

Where C_1 and C_2 are model constants.

The primary sources of radiative heat transfer in a diesel engine are the high temperature burned gases and the soot particles which are formed in the turbulent, diffusion controlled diesel flame. The instantaneous radiant heat flux is expressed as **(Szekely & Alkidas,1987):**

$$\frac{dQ_{Radiation}}{dt} = \varepsilon \ . \delta . A . \left(T_g^4 - T_w^4 \right) \quad (5)$$

Where ε total emissivity.

δ **Stephan-Boltzman** constant (5.67*10^{-8} W/m^2.^{o}K^4).

This equation assumes that each zone radiates to the whole combustion chamber surface area without any radiative interactions between zones.

Radiation in diesel engine is due to luminous (soot) and non-luminous (gaseous) radiation. Thus, the total emissivity is calculated from the following equation **(Szekely & Alkidas,1987):**

$$\varepsilon = \varepsilon_g + \varepsilon_s - \varepsilon_g . \varepsilon_s \tag{6}$$

$$\varepsilon_g = \varepsilon_{H_2O} + \varepsilon_{CO_2} - \varepsilon_{CO_2} . \varepsilon_{H_2O} \tag{7}$$

The gas emissivities of carbon dioxide, and water vapor are obtained by the following empirical relations**(Szekely & Alkidas,1987)**:

$$\varepsilon_{CO_2} = 0.711 \left(P_{CO_2} . Lr \right)^{1/3} / \left(T_Z / 100 \right)^{1/2} \tag{8}$$

$$\varepsilon_{H_2O} = 0.707 \left(P_{H_2O} . Lr \right)^{1/3} / \left(T_Z / 100 \right)^{1/2} \tag{9}$$

The soot emissivity is obtained from the following equation, derived by **(Szekely & Alkidas,1987):**

$$\varepsilon_s = 1 - \exp \left\{ P^C \ln \left[1 - \varepsilon_{go} \left(\frac{0.09}{\left(\tau^2 - sg^2 + 0.35 sg - 0.38 \right)} + 6.8 sg - 5.95 \right) \right] \right\} \tag{10}$$

Where subscript o on the gas emissivity term refers to the gas emissivity at atmospheric pressure

τ air to fuel ratio (defined as $(1/\phi)$)

sg specific gravity of the fuel

The exponent of the pressure P is given by the following expression:

$$C = 4.95 / (\tau + 1.5) - 0.25$$

The total heat transfer to the liquid fuel droplets is given by **(Jung,2001)**;

$$\frac{dQ_{fg}}{dt} = \sum_{i=zones} q_i \tag{11}$$

During the evaporation process convective heat transfer from the hot gas to liquid fuel droplets in a zone q is modeled following the methodology of **(Gosman & Johns,1980).**

$$q = \pi . d_l . N_d . K_m \left(T_g - T_l \right) . Nu . f_b \tag{12}$$

To correct these effects of boundary layer thickening due to mass transfer, the heat transfer coefficient in equation (12) is multiplied by a factor f_b which is given by **(Gosman & Johns,1980)**;

$$f_b = \frac{Z}{(\exp(Z)-1)} \tag{13}$$

$$Z = \frac{C_{p_v}.\frac{dm_{f,g}}{dt}}{\pi d_l N K_m Nu} \tag{14}$$

MODELING OF POWER CYCLE

The power cycle of internal combustion engine consists of three processes namely compression, combustion and expansion**.** These processes are modeled as follows;

Compression Process

The compression starts at the trapped condition, and ends at the end of delay period. The first law of thermodynamic for a closed system is given by **(Heywood,1989)**;

$$\frac{dQ}{d\theta} = \frac{dE_{comp}}{d\theta} + \frac{dW}{d\theta} \qquad \text{[kJ/Degree]} \tag{15}$$

dQ Heat transfer rate to the cylinder wall [kJ]

dE_{comp} is the change of internal energy of the system [kJ]

$$\frac{dE_{comp}}{d\theta} = N_{comp}.C_{v_{comp}}\left(T_{comp}\right).\frac{dT_{comp}}{d\theta} \tag{16}$$

dW is the work transfer to the system [kJ].

$$\frac{dW}{d\theta} = P.\frac{dV}{d\theta} \tag{17}$$

A combination of first law of thermodynamic with the equation of state yields cylinder pressure and temperature as follows;

$$\frac{dP}{d\theta} = \left[-\left(1+\frac{R_{mol}}{CV_{comp}\left(T_{comp}\right)}\right).P.\frac{dV}{d\theta} + \frac{R_{mol}}{CV_{comp}\left(T_{comp}\right)}.\frac{dQ_w}{d\theta}\right]/V \tag{18}$$

$$\frac{dT_{comp}}{d\theta} = T_{comp}.\left[\frac{1}{P}.\frac{dP}{d\theta} + \frac{1}{V}.\frac{dV}{d\theta}\right] \tag{19}$$

Equations (18) and (19) are solved by Runge-Kutta method to calculate the cylinder temperature and pressure respectively during each time step.

COMBUSTION PROCESS

Spray zones are formed as a result of fuel injection; these zones are surrounded by a mixture of hydrogen, air and residual gas. In the present combustion model each zone is considered as an open system. The mass, energy and momentum equations are applied for each zone to yield cylinder pressure and zonal temperature. The effect of dissociation and rate kinetics accounts are accounted for and a 12 species of combustion products are considered with complete chemical equilibrium for same reactions.

Conservation of Momentum

Mixture entrainment (hydrogen+air) rate into a given zone is controlled by the conservation of momentum applied to that zone. It is assumed that the initial momentum imparted in a zone upon nozzle exit is equal to the momentum of the zone at any subsequent distance traveled **(Jung,2001)** i.e.

$$m_f . u_i = \left(m_f + m_m\right).\left(\frac{dS}{dt}\right) \tag{20}$$

Where $\left(\frac{dS}{dt}\right)$ is the velocity of zone (m/sec)

m_f mass of fuel in a zone.

m_m mass of mixture entrainment in the zone.

Therefore; the amount of entrained mixture in a zone is proportional to the decrement in the zone velocity. Equation (20) can be rewritten as;

$$m_m = m_f \left(\frac{u_i}{\frac{dS}{dt}} - 1 \right) \tag{21}$$

By differentiating equation (21) with respect to time, the mixture entrainment rate can be obtained as;

$$\frac{dm_m}{dt} = -\frac{m_f . u_i}{\left(\frac{dS}{dt}\right)^2} . \frac{d^2 S}{dt^2} \tag{22}$$

Conservation of Mass

The rate of change of total mass in any open system is equal to the sum of the flow rates into and out of the system **(Assanis,1986)**.

$$m_i^{\cdot} = \sum m_{i,j}^{\cdot} \tag{23}$$

Where $m_i^{\cdot}$ mass flow rate of fuel in parcel of fuel i .

$m_{i,j}^{\cdot}$ mass flow rate of fuel in parcel of fuel zone i, j

Conservation of Energy

The first law of thermodynamic supplemented with a heat transfer model can be applied to yield cylinder pressure and temperature of the air zone and spray zones. Treating the contents of each zone as an ideal gas, with thermodynamic properties that are functions of temperature, pressure and fuel-air equivalence ratio an equation for the time rate of change of pressure and zonal temperature are obtained **(Jung,2001)**:

$$\frac{dP}{dt} = \frac{\rho}{d\rho / dP}\left(-\frac{V^{\cdot}}{V} - \frac{1}{\rho}\frac{dP}{dT}\frac{dT}{dt} - \frac{1}{\rho}\frac{d\rho}{d\phi}\frac{d\phi}{dt} + \frac{m^{\cdot}}{m} \right) \tag{24}$$

$$\frac{dT_i}{dt} = [\ \sum m_{i,j}^{\cdot} h_{i,j} + \frac{dQ_i}{dt} + \left(V_i - C_{Ti}\right)\frac{dP_i}{dt} - m_i C_{\phi i}\frac{d\phi_i}{dt} - m_i^{\cdot} h_i\ ] / m_i^{\cdot} Cp_i \tag{25}$$

The first term in equation (25) is the net rate of influx of enthalpy, $\frac{dQ}{dt}$ is the total heat transfer rate to the system. Equations (24), and (25), are solved by Runge-Kutta method to calculate the cylinder pressure and zonal temperature respectively during each time step.

The heat release during premixed combustion phase is given by **(Nishida,1989);**

$$RRp = B1.\rho_{mix}^{2}.X_{fv}.X_{ox}\exp\left(-\frac{1200}{T_Z} \right).V_Z \tag{26}$$

Equation (26) is assumed to be valid until all the fuel evaporated during the ignition delay period has been consumed. During diffusion phase the heat release is given by **(Nishida,1989);**

$$RRm = B2.m_{f,v}.\frac{P_{ox}}{P}P^{2.5}\exp\left(-\frac{2500}{T_Z} \right) \tag{27}$$

Where $B1$ frequency factor equal to 1.5 & $B2$ is constant equal to 800 **(Jung,2001)**

EXPANSION PROCESS

Once the combustion process is complete, the cylinder is considered as a single zone with a uniform temperature and pressure. The number of moles of each species is calculated during each time step. The calculation stops at exhaust valve opening. This completes the power cycle. The Runge-Kutta method is used for the same analysis for compression stroke.

ENGINE PERFORMANCE CALCULATIONS

This section covers engine performance parameters which include: indicated mean effective pressure, indicated power, specific fuel consumption, volumetric efficiency and finally thermal efficiency.

The indicated mean effective pressure is given by **(Heywood,1989)**;

$$i.m.e.p = \frac{\oint work}{V_d} \tag{28}$$

Where

$\oint work$ summation of the work during the cycle

V_d stroke volume.

The indicated power is obtained from the following equation **(Heywood,1989):**

$$I.Power = (i.m.e.p) * V_d * \frac{N}{n} \text{ [kW]} \tag{29}$$

Where

N engine speed rps.

n number of power stroke per thermodynamic cycle and its equal to 2 for 4-stroke engine, and 1 for 2-stroke engine.

A more useful parameter is the indicated specific fuel consumption "I.S.F.C" which can be found from **(Heywood,1989)**;

$$I.S.F.C. = \frac{\dot{m}_f \times 3600}{I.Power} \text{ [g/kW.h]} \tag{30}$$

The Indicated thermal efficiency $I.\eta_{th}$ can be defined as the ratio of the Indicated power to the energy content of the fuel supplied;

$$I.\eta_{th} = -\frac{I.Power}{\dot{m}_f \times Q_{vt}} \tag{31}$$

Where: Q_{vt} is the summation of lower calorific values of the dual fuels used.

By combining equation (30) with equation (31), the efficiency can be written as follows **(Heywood,1989)**;:

$$I.\eta_{th} = -\frac{3600}{I.S.F.C. \times Q_{vt}} \tag{32}$$

RESUITS AND DISCUSSSIONS

In this section the predicted results of the effect of hydrogen blending on the thermodynamic cycle parameters of the engine, at 1500 RPM engine speed, 15 compression ratio, with 20 ^{o}CA injection duration and a fixed injection timing of -17 ^{o}CA, are presented and discussed.

Figs.(1&2) present the relation between cylinder pressure and crank angle for different hydrogen ratios started with (8-12)% and ends with(12-16)% by mass respectively. In general the global behavior for these figures is the same except the difference in the value and timing of peak pressure which depend on the blending ratios. It can be seen that as the hydrogen fuel mass ratio increases, maximum pressure increase reaching its maximum value at 12% hydrogen, and start to decrease with further hydrogen addition. The increase of hydrogen addition from (0-12) %, by mass, causes an increase in the maximum pressure from 92.876 bar to 115.982 bar. The maximum pressure was observed to occur earlier in the cycle when hydrogen was inducted along with the intake air. This is due to increase in the rate of mass burning. Therefore, the time required to complete combustion is reduced. This produced lower heat transfer to the cylinder walls, lower exhaust emissions and higher engine efficiency. It is observed from **Fig.(2)** the pressure decrease with hydrogen percent higher than 12%, due to the decrease in the volumetric efficiency. **Figs.(3-6)** present the temperature of typical zones which show dramatic changes during combustion stage. Some of these zones are stoichiometric, and the others are lean and rich. These figures show the effect of hydrogen addition on the zonal temperature histories. The results show that the tip and the tail zones have higher peak temperature than the global gas temperature. It is clearly shown that the maximum zonal temperature increases, as the hydrogen fuel mass ratio increases reaching its maximum value at 12%. This is due to the reduction in the combustion duration, which produced lower heat transfer to the cylinder walls, lower exhaust emissions and higher engine efficiency. Higher percentage, more than 12% of hydrogen addition reduces zonal temperature, due to the decrease in the volumetric efficiency. **Fig.(7)** presents the zonal rate of heat release with pure diesel and **Figs.(8-10)** show the zonal rate of heat release with hydrogen blending (4-16) % by mass respectively. The result shows that the addition of hydrogen improves the combustion efficiency and increases the maximum rate of heat release which moves closer to TDC. The increase of hydrogen addition from (4-12) %, by mass leads to increase the maximum rate of heat release. This is attributed to the increase in the flame propagation speed which increases the rate of heat release and enhances the turbulence inside the cylinder. **Figs.(11-15)** present performance parameter for three global equivalence ratios. The performance parameter is made dimensionless relative to their values when pure diesel is used. It can be seen from **Figs.(11)** that combustion duration deceases as the percentage of hydrogen addition increases. This due to high burning speed of hydrogen (2.7-3.3) m/sec compared to that of diesel (0.32-0.38) m/sec **(Claus,1997)**. Therefore the time required to complete the combustion becomes shorter. The reduction in the combustion duration is very important, since it leads directly to lower heat transfer loss to the cylinder walls, higher thermal efficiency, lower specific fuel consumption, and lower HC emissions. Similar results are observed by **(Hacohen,1987)**. They showed that the increase of hydrogen enrichment will reduce the combustion duration, and the decrease of equivalence ratio increases the combustion duration. **Fig.(12)** it shows the effect of hydrogen addition on the indicated mean effective pressure. It is observed that there is an increase in the indicated mean effective pressure corresponding to the increase in the hydrogen percent reaching its peak value at 12%, then with further increase, it starts to decrease in all types of mixtures. This is due to the decrease in the maximum pressure that occurs with hydrogen ratios greater than 12%. **Fig.(13)** shows that specific fuel consumption decreases as the percentage of hydrogen addition increases. This is expected because hydrogen is characterized by high energy-mass coefficient of all the fuels and in terms of mass energy consumption it exceeds the conventional diesel fuel by about three times (see **Appendix (A)**). Therefore the results clearly establish that the supplement of 10% hydrogen in the diesel–air mixture reduces S.F.C by 60%. Above 10% hydrogen addition, the S.F.C. increases due to the reduction in the volumetric efficiency which causes a clear reduction in the indicated power and efficiency. These results agree well with the results obtained by **(Harold,1985)**. He showed that a significant reduction in S.F.C was achieved with hydrogen addition to the diesel fuel engine. From **Fig.(14)** it was found that indicated power increase initially as the percentage of hydrogen addition increases, up to 10% hydrogen addition. This increase in the indicated power is due to high rate of mass burning and higher heating value of hydrogen, which causes an improvement in the combustion efficiency. With further increase in the percentage of hydrogen addition, the power decreases, since the effect of the reduction in the mixture density and volumetric efficiency becomes dominant. **Fig.(15)** explains the effect of hydrogen blending on the thermal efficiency of the diesel engine. This result showed that with hydrogen addition, the thermal efficiency was observed to increase and reaches a maximum value at a hydrogen addition of 10 %, then with further increase in the hydrogen addition, the thermal efficiency decreases. The increase in the thermal efficiency is attributed to high diffusion rates of hydrogen and faster energy release due to the increase flame propagation speed, improvement of the combustion process and the reduction in the combustion duration. In spite of the continuing reduction in the combustion duration as the hydrogen addition is increased, the reduction in the volumetric efficiency will affect considerably the thermal efficiency causing a clear reduction. Similar results are observed by **(Gopal,1982)** and **(Harold,1985)**. They showed that as burning speed increases, the combustion duration is shortened and the actual indicator diagram seems closer to TDC, therefore a higher efficiency is obtained.

CONCLUSIONS

The blending ratio of 12% hydrogen by mass is the optimum ratio which gives maximum pressure, zonal temperature, and zonal heat release rate. Maximum improvement in indicated power and engine efficiency takes place at 10% hydrogen blending. A blending ratio of 12% hydrogen gives the maximum reduction in S.F.C. which is about 60%.

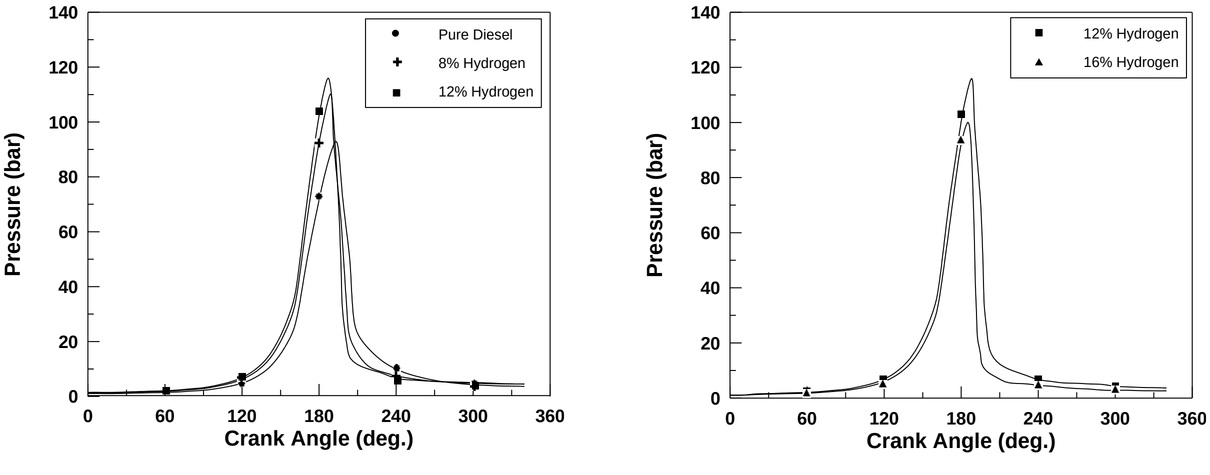
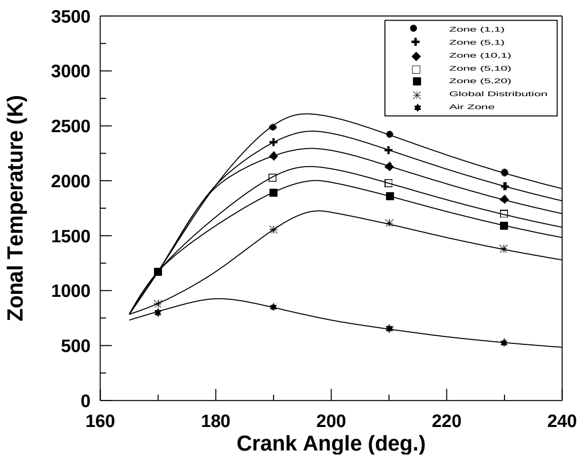


Fig.(1) &Fig.(2) Variation of Cylinder Pressure with Crank Angle at Different Hydrogen Additions.



Fig(3) Effect of Hydrogen Addition on the Zonal Temperature Distribution Pure Diesel.

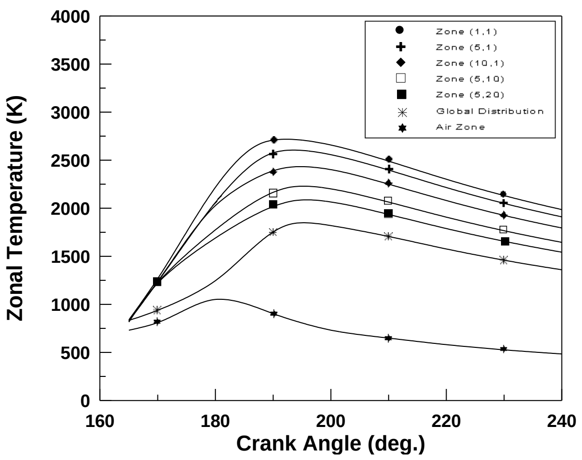


Fig.(4) Effect of Hydrogen Addition on the Zonal Temperature Distribution Fuel Mass Ratio: 92% Diesel, 8% Hydrogen.

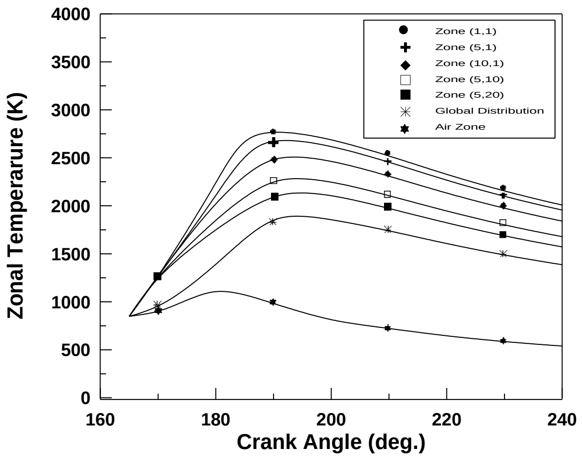


Fig.(5) Effect of Hydrogen Addition on the Zonal Temperature Distribution Fuel Mass Ratio: 88% Diesel, 12% Hydrogen.

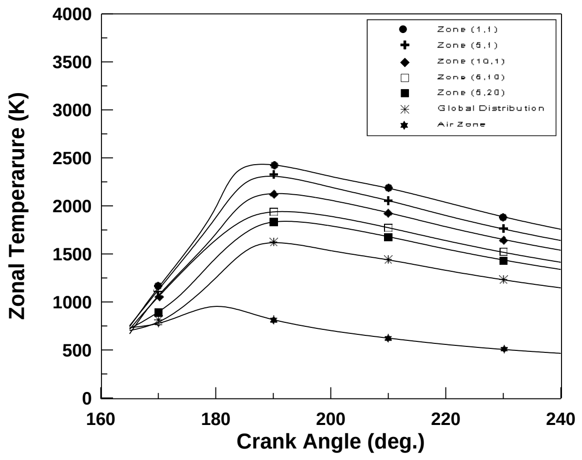
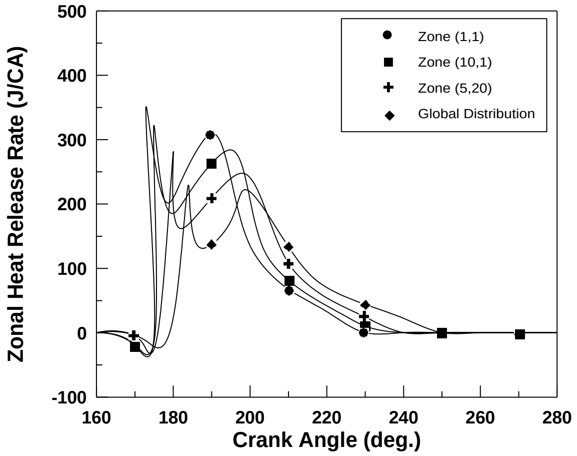


Fig.(6) Effect of Hydrogen Addition on the Zonal Temperature Distribution Fuel Mass Ratio: 84% Diesel, 16% Hydrogen.



Fig(7) Effect of Hydrogen Addition on the Zonal Heat Release Rate Pure Diesel.

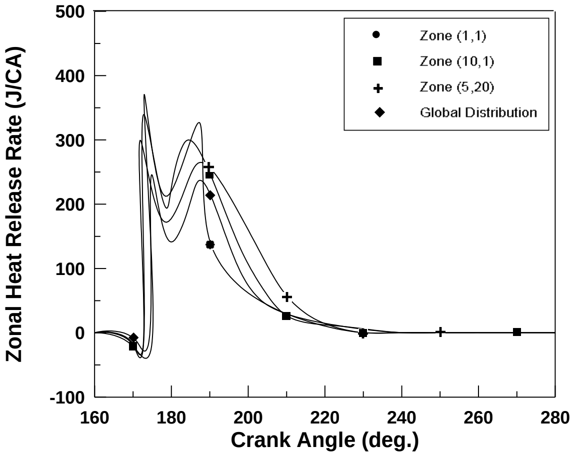


Fig.(8) Effect of Hydrogen Addition on the Zonal Heat Release Rate Fuel Mass Ratio: 92% Diesel, 8% Hydrogen.

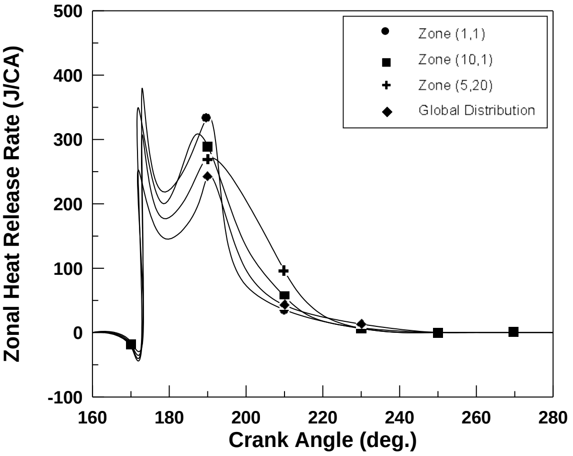


Fig.(9) Effect of Hydrogen Addition on the Zonal Heat Release Rate Fuel Mass Ratio: 88% Diesel, 12% Hydrogen.

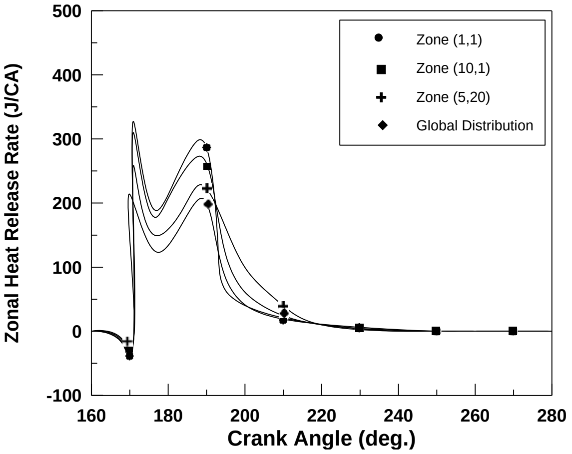
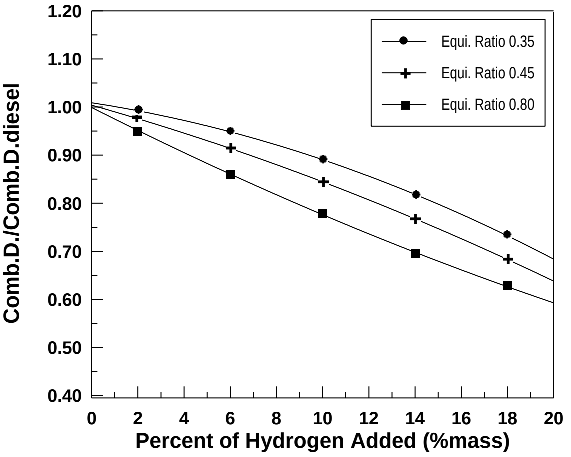
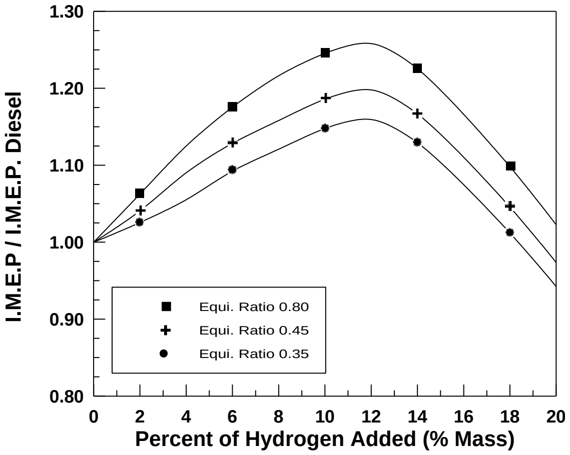


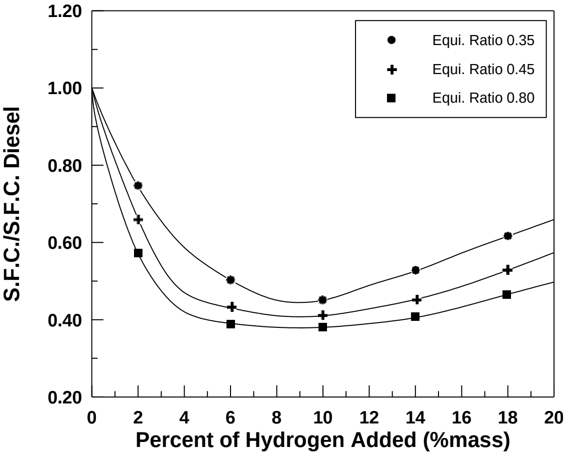
Fig.(10) Effect of Hydrogen Addition on the Zonal Heat Release Rate Fuel Mass Ratio: 84% Diesel, 16% Hydrogen.



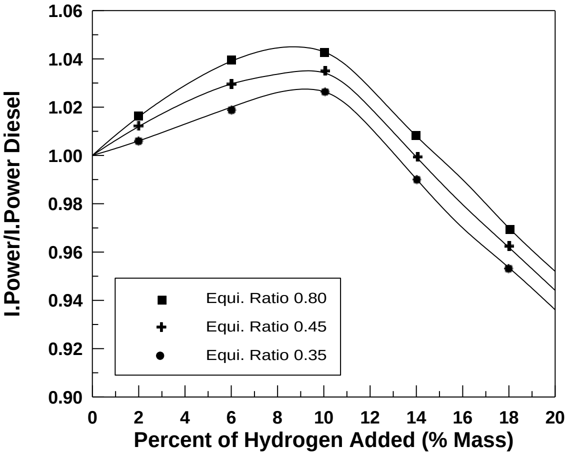
Fig(11) Effect of Hydrogen blending on the Combustion Duration of Diesel Engine with Different Equivalence Ratios.



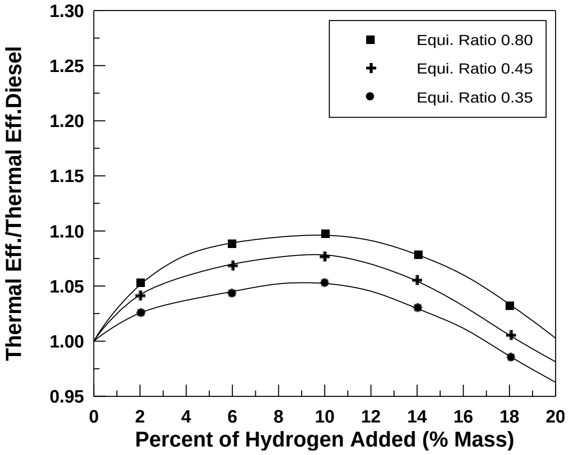
Fig(12) Effect of Hydrogen blending on the Indicated Mean Effective Pressure of Diesel Engine with Different Equivalence Ratios.



Fig(13) Effect of Hydrogen blending on the Specific Fuel Consumption of Diesel Engine with Different Equivalence Ratios.



Fig(14) Effect of Hydrogen blending on the Indicated Power of Diesel Engine with Different Equivalence Ratios.



Fig(15) Effect of Hydrogen blending on the Thermal Efficiency of Diesel Engine with Different Equivalence Ratios.

NOMENCLATURE

The following symbols are used generally throughout the text. Others are defined as when used.

Symbol	Description	Units
A	Total Area of Cylinder	m^2
dS/dt	Velocity of Zone	m/sec
dE_{comp}	Step Change of Internal Energy during Compression stage	kJ
dW	Step Change of Work in the system during Time Step	kJ
h_C	Heat Transfer Coefficient	$w/m^2.^oK$
Lr	Radiation Path Length	mm
$m_{f,v}$	Mass of Fuel Vapor	kg
N_d	Number of Droplets Contained in the zone	
P	Cylinder Pressure	bar
P_{ox}	Partial Pressure of Oxygen	atm
ΔP	Pressure Drop across the Injector Nozzle	Mpa
Q_w	Heat Transfer Rate to the Cylinder Wall	kW
Q_{fg}	Total Heat Transfer Rate to the Liquid Fuel Droplets	kW
$Q_{radiation}$	Radiation Heat Transfer Rate	kW
$Q_{convection}$	Convection Heat Transfer Rate	kW
r_Z	Zonal Radius	mm
RRp	Heat Release during Premixed Combustion Phase	J/degree
RRm	Heat Release during Diffusion Combustion Phase	J/degree

T_l	Liquid Fuel Temperature	^oK
T_Z	Zonal Temperature	^oK
X_{fv}	Mass Fraction of Fuel Vapor	-

GREEK SYMBOLS

Symbol	**Description**	**Units**
λ_f	Specific Heat of Vaporization of Liquid Fuel.	*kJ/Kmol*
ϕ	Equivalence Ratio.	-
$\partial\phi/\partial t$	Equivalence Ratio Gradient.	-
w	Local average gas velocity in the cylinder.	*[m/sec]*
θ	Crank Angle.	*degree*

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APPENDIX A

Table (1) Properties of Diesel and Hydrogen Fuel (Claus,1997).

Property	Hydrogen	Diesel Fuel
Chemical Formula	H_2	$C_{14} H_{30}$
Molecular Weight (kg/Kmol)	2.02	198
Density [kg/m^3]	83.7640	820
Stoi. air/fuel, (weight)	34.4	102.34
Maximum Flame Speed [m/s]	2.92	0.38
Specific Heat [kJ/kg.k^o]	1.44	≈1.7
Lower Heating Value [kJ/kg]	120000	42800

Table (2) Engine Specification (Jung,2001).

Engine Type	Caterpillar 3406**,** Diesel Engine
Charging	Natural Aspiration
Number of Cylinder	1
Bore [mm]	137.1
Stroke [mm]	165.1
Connected Rod Length [mm]	261.62
Capacity [Litters]	2.44