A Numerical Investigation of the Effect of Hydrogen Blended on the Temperature Field and Performance of a Spark Gnition Engine

Samir M. Abdul Haleem, and Haroun A.K. Shahad

***Abstract*—** In the present work, theoretical analysis on the effects of hydrogen blending with gasoline fuel when used in a four stroke spark ignition engine on the performance has been performed.

Simulation of the: power cycle for naturally aspirated engine was performed. A multi-zone quasi dimensional method with the semi empirical flame speed correlation was used to model the combustion process. Zonal heat transfer by radiation and convection was developed. The effect of dissociation of combustion products and rate kinetics was account for in this model. The nitric oxides formation was modeled.

A computer program written in Fortran language has been developed to calculate the performance in a spark ignition engine working with either pure gasoline or hydrogen-gasoline mixture. The model was used to predict the following parameter: Zonal temperature history, Overall cylinder temperature, Cylinder pressure history, Heat loss by radiation and convection, Flame front speed, Engine performance, Pollutant concentration in exhaust gases

***Keywords*—** hydrogen, blended fuel, spark ignition engine

I. Introduction

The various engine combustion models that have been developed to date may be grouped into three categories [1]:

1. Zero dimensional models 2. Quasi-dimensional models

3. Multi-dimensional models

In the above classification, although the level of detail and proximity to physical reality increases as one precedes downward, so does the complexity of creating and using those models. Multi-zone models (which considered in the present study) take this form of analysis one step further by considering energy and mass balances over several zones, thus obtaining results that are closer to reality.

The aim of the present work is to study the performance of a single cylinder air cooled spark ignition engine operated with gasoline enriched hydrogen. However many researches had been directed toward engines operated with hydrogen blended fuel, performance and components.

Prabhu et al, [2], 1985, developed an analytical model to study the performance, fuel economy and nitric oxide emission of a spark ignition hydrogen engine. A quasi one dimensional model was developed. The semi empirical turbulent flame speed expression suggested by Fagelson was used. $U_T = A^* \mathrm{Re}^B * U_L$

Samir M. Abdul Haleem, Mechanical Engineering Department, Babylon University Babylon, Iraq, email: samir76samir76@yahoo.com

Haroun A.K. Shahad, Mechanical Engineering Department, Babylon University Babylon, Iraq, email: hakshahad@yahoo.com

In which A and B are empirically determined constants, Re is the Reynolds numbers based on the piston diameter, mean piston speed, and the burnt gas properties, and UL is the laminar flame speed. They found that: By supercharging the hydrogen engine the knock limit was set in at an equivalence ratio leaner than the naturally aspirated hydrogen engine; The indicated power increases for supercharging pressure when compared with the naturally aspirated engine.

Shahad and Sadiq, [3], 1999, developed a quasi–dimensional model to study the effect of hydrogen blending on fuel consumption and pollutant concentrations. Investigations had been concentrated on decreasing fuel consumption by using alternative fuels and on lowering the concentration of toxic components in combustion products. They concluded that The hydrogen added to gasoline engine improves the combustion process, especially in the later combustion period, reduces the ignition delay, speed up the flame front propagation, reduces the combustion duration and retards the spark timing, and The thermal efficiency of the engine is increased until a hydrogen–fuel mass ratio of 8% of stoichiometric mixture and 10% for 0.8 equivalence ratio.

Das, et al, [4], 2000, evaluated the potential of using a clean–burning fuel such as hydrogen for a small horsepower spark ignition engines since (CNG) has already been used as an alternative fuel for internal combustion. An effort had been made in the present work to compare hydrogen selling with CNG operation. The engine was operated separately either with hydrogen or compressed natural gas using an electronically–controlled, solenoid–actuated injection system developed in the engine and unconventional laboratory of the Indian Institute of Technology, Delhi. It had been observed that the brake specific fuel consumption was reduced and the brake thermal efficiency improved with hydrogen operation compared to systems running on compressed natural gas.

Hailin and Karim, [5], 2003, employed a quasi–dimensional two–zone model for the operation of spark ignition engine when fueled with hydrogen. The effects of changes in operation conditions include a very wide range of variations in equivalence ratio on the onset of knock and its intensity, combustion duration, power, efficiency and operational limits were investigated. They found that the compression ratio and intake temperature are the main parameters that affect the knock limited equivalence ratio while the effect of spark timing tend to be in comparison less effective. The knock free operational mixture region tends to narrow significantly with increasing compression ratio and intake temperature. This represents a practical limitation to the improvement of power and efficiency of hydrogen engines.

However many research programs were recently carried out to study the performance of hydrogen engine [6], [7], [8], [9], [10], [11].

It is found that pure hydrogen can be used as a fuel for gasoline engines with slight increase in compression ratio due to the high self-ignition temperature of hydrogen. The only pollutant in this case is the nitrogen oxide if air is used.

It must be stressed that in this work the mixing process is based on energy basses rather than on mass basses, where the amount of hydrogen energy added is equal to the amount of hydrocarbon energy removed. It is thought that this mixing process given better assessment of the blending process.

II. MATHEMATICAL MODEL

The reaction equation of the complete combustion of hydrogen blended gasoline fuel is written as follows.

$$C_8H_{18} + \alpha H_2 + \beta(O_2 + 3.76N_2) \rightarrow r_1CO_2 + r_2O_2 + r_3N_2 + r_4H_2O \quad (2)$$

2.1 Zone Generation

The whole charge in the cylinder is divided into layers of equal height. Each layer is divided into nn (10) circumferential ring elements, each ring element is divided into 360 zones (each one degree) as shown in figure (1). So the element thickness (d) is

$$d = \frac{b/2}{10} \quad (3)$$

Assuming the zone has a square profile, i.e. the height equal the thickness, thus the number of rows of zones in the z direction (mm) is equal to

$$mm = \left(s + \frac{4V_c}{\pi b^2}\right) \Big/ d \quad (4)$$

The volume of each element is a function of its location as

$$V(ii, jj, kk) = \pi * 0.1 * ii * b * d^2 / 360 \quad (5)$$

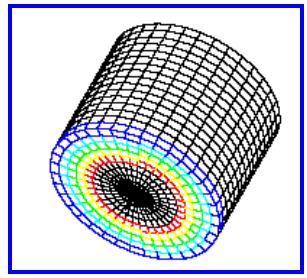
Where $ii = 1, nn(10)$ $\quad jj = 1, mm$ $\quad kk = 1,360$

The summation of volumes of all zones at each time (crank angle) step must equal to the total cylinder volume

$$V_{cyl}(\theta) = \sum_{ii=1}^{nn}\sum_{jj=1}^{mm}\sum_{kk=1}^{360} V(ii, jj, kk) \quad (6)$$

Also the summation of masses of all zones equal to the total mass of the cylinder, thus [12]

$$m_{cyl} = \sum_{ii=1}^{nn}\sum_{jj=1}^{mm}\sum_{kk=1}^{360} m(ii, jj, kk) \quad (7)$$



Incylinder Zonal Distribution [12]

2.2 Heat Transfer Model:

Heat flux into the walls varies through the engine cycles from negative to positive values reaching up to several megawatts per square meter. Also varies spatially due to differences in local gas temperature and velocity and doubles also due to differences in orientation of the wall to the radiating flames.

2.2.1 Convection Heat Transfer

Woschni developed the power of 0.8 empirical equation for forced convection heat transfer as

$$Nu = 0.035\mathrm{Re}^{0.8} \quad (8)$$

Woschni proposed an empirically based in which the characteristic velocity term (local average gas velocity) is expressed as [13]

$$w = C_3 Sp + C_4\left(\frac{V_d T_{ref}}{P_{ref} V_{ref}}\right)(P - P_{mot}) \text{ [m/s]} \quad (9)$$

Where Pr, Vr, Tr working fluid pressure, volume, temperature at some reference condition (inlet valve closing)

C3 and C4 are model constant, which are specified as [14]

For compression period C3=2.28 C4=0

For combustion and expansion period C3=6.18 C4=3.24*10-3

With the cylinder bore taken as characteristic length

2.2.2 Radiation Heat Transfer

2.2.2.1 Total Gas Emissivity

The non-luminous radiation from the combustion gases is primarily due to emission contributions from the tri-atomic molecules of carbon dioxide, water vapor. Thus the total gas emissivity calculated by the following approximation [15].

$$\varepsilon_g = \varepsilon_{CO_2} + \varepsilon_{H_2O} - \varepsilon_{CO_2}\varepsilon_{H_2O} \quad (10)$$

The emissivity of carbon dioxide and water vapor can be calculated by the following empirical relations:

$$\varepsilon_{CO_2} = 0.711(p_{CO_2} l_{rad})^{1/3} \Big/ (T_{ij}/100)^{1/2} \quad (11)$$

$$\varepsilon_{H_2O} = 0.707(p_{H_2O} l_{rad})^{1/3} \Big/ (T_{ij}/100)^{1/2} \quad (12)$$

Where pCO2 partial pressure of (CO2) [bar]

pH2O partial pressure of (H2O) [bar]

Tij temperature of zone (i, j) [K]

The mean path length of a volume V and a surface area A is given with sufficient accuracy by [14]

$$l_{rad} = 0.9\frac{4V}{A} \tag{13}$$

2.2.2.2 Direct and Total Exchange Area Factors

Assume the surface of the equivalent cylinder includes the cylinder linear, the head of the cylinder, and the piston crown. Eight types of direct exchange area factors are calculated during each time step and they are

1. Cylinder linear to cylinder head direct exchange area.
2. Cylinder linear to piston crown direct exchange area.
3. Cylinder linear to itself direct exchange area.
4. Piston crown to cylinder head direct exchange area.
5. Gas element to cylinder linear direct exchange area.
6. Gas element to piston crown direct exchange area.
7. Gas element to cylinder head direct exchange area.
8. Gas element to gas element direct exchange area.

The surface is broken into N isothermal elements while the medium is broken into k isothermal volumes.

Where the subscripts s and g are used to distinguish between the emissive power and irradiation of surface element and volume zone, respectively.

Making an energy balance over a volume zone, have [16].

$$Q_{gi} = \kappa V_i (4Eb_{gi} - G_i) \tag{14}$$

$$Q_{gi} = \sum_{j=1}^{N} \overline{s_j g_i}(Eb_{gi} - J_j) + \sum_{k=1}^{K} \overline{g_k g_i}(Eb_{gi} - Eb_{gk}) \tag{15}$$

$$Q_{gi} = 4\kappa V_i Eb_{gi} - \sum_{j=1}^{N} \overline{s_j g_i} J_j - \sum_{k=1}^{K} \overline{g_k g_i} Eb_{gk}, i = 1,2.....K \tag{16}$$

Looking to an isothermal enclosure, found that

$$\sum_{j=1}^{N} \overline{s_j g_i} - \sum_{k=1}^{K} \overline{g_k g_i} = 4\kappa V_i, i = 1,2,.....K \tag{17}$$

For a volume zone the Qgi represent the net radiative source within volume Vi and is, therefore,

$$Q_{gi} = \int_{V_i} A.q dV_i \tag{18}$$

$$Q_{gi} = 4\kappa V_i Eb_{gi} - \sum_{j=1}^{N} \overline{s_j g_i} J_j - \sum_{k=1}^{K} \overline{g_k g_i} Eb_{gk}, i = 1,2..K \tag{19}$$

2.3 Mechanism of Flame

The combustion process in the spark ignition engine takes place in a turbulent flow field [13]. Unlike a laminar flame, which has a propagation velocity that depend unequally on the thermal and chemical properties of the mixture, a turbulent flame has a propagation velocity that depends on the characteristics of the flow, as well as the mixture properties. Turbulent flame speed (Ut) can be defined as the velocity at which the unburned mixture enters the flame front in a direction normal to the flame [17].

The essential features of the flame development and propagation processes are described as

1. The flame development angle: the crank angle interval between the spark discharge and the time when a small fraction of the cylinder mass has burned. Usually this fraction equal to (0.1 percent) of cylinder volume. Hence the development period is [13, 17]

$$Dp = \frac{6RPM}{St}\sqrt[3]{\frac{0.001Vcyl}{\pi}} \tag{20}$$

2. Rapid–burned angle: the crank angle interval required to burn the bulk of the charge. Its defined as the interval between the end of the flame development and the end of flame propagation process

The laminar flame velocity for mixture of gasoline and hydrogen fuel in spark ignition is modeled as [6]

$$Ul = 9656\left(\frac{P}{P_o}\right)^{-0.623} e^{\frac{-2145}{Tu}} + 0.83Y_{H_2} \tag{21}$$

Po reference pressure [bar]

$$Y_{H_2} = \left[\frac{[H_2] + \frac{[H_2]}{[H_2/air]_{st}}}{[G] + [air] - \frac{[H_2]}{[H_2/air]_{st}}}\right] \tag{22}$$

where $[H_2]$ is the molecular concentration of hydrogen

$[G]$ is the molecular concentration of gasoline

$[air]$ is the molecular concentration of air

$[H_2/air]_{st}$ is the stoichiometric ratio of hydrogen concentration to air concentration

Laminar flame speed is affected by the residual gas. Heywood [14] show that

$$Ul = Ul(1 - 2.06Xb^{0.77}) \tag{23}$$

Where Xb is the mole fraction of diluents of the burned gas. Therefore the turbulent flame front speed is described as

$$Ut = Ul * ff \tag{24}$$

Where ff is the turbulization factor which can be taken proportional to engine speed [14, 19]

$$ff = 1 + 0.0018RPM \tag{25}$$

2.5 Overall cylinder Temperature and Pressure:

The overall cylinder temperature is obtained as follows

$$T_{overall} = \frac{\sum_{ii=1}^{nn}\sum_{jj=1}^{mm}\sum_{kk=1}^{360} m(ii, jj, kk)T(ii, jj, kk)}{m_{cyl}} \tag{26}$$

$$\frac{P_{cyl}V_{cyl}}{RT_{overall}} = \sum_{ii=1}^{nn}\sum_{jj=1}^{mm}\sum_{kk=1}^{360}\frac{P(ii, jj, kk)V(ii, jj, kk)}{RT(ii, jj, kk)} \tag{27}$$

or the new cylinder pressure becomes

$$P_{cyl} = \frac{RT_{overall}}{V_{cyl}}\sum_{ii=1}^{nn}\sum_{jj=1}^{mm}\sum_{kk=1}^{360}\frac{P(ii, jj, kk)V(ii, jj, kk)}{RT(ii, jj, kk)} \tag{28}$$

6 Combustion process

The combustion process starts as the spark plug fired. The flame expands and travels across the chamber until finally the whole of the mixture is engulfed by flame. The combustion chamber is divided into many layers of ring elements; each layer contains ten rings as shown in section (3.4). Each ring element is broken into 360 elements. The flame front travels in a surface of a hemispherical shape assumed of thickness equal to the mesh size. The combustion process occurs only in the zones that lie on the flame front except the dissociation processes which continuous in the back flame zones.

The convection heat transfer between each ring element and its neighbors is calculated. The radiation heat transfer between each zone and all other zones, cylinder linear, cylinder head and piston crown are calculated at each time step.

The time step or crank angle step (in the combustion process) is the time required for the flame to travel a zonal thickness. Thus the time increment is

$$\Delta t = \frac{\sqrt{2}.d}{St} \tag{28}$$

So the crank angle increment becomes

$$\Delta\theta = 6.RPM\Delta t \tag{29}$$

III. RESULTS AND DISCUSSION

Figure (1) shows the flame speed as a function of air to fuel ratio for pure gasoline and 10 % hydrogen energy blending ratio for engine compression ratio of 11 and of 2100 rpm which represent the conditions used by J. F. Cassidy [10]. The results of J. F. Cassidy [10] are also shown in the same figure for pure gasoline and 7% mass replacement ratio.

The effect of hydrogen blending on the mean cylinder temperature is also studied. Figure (2) shows the effect at an engine speed at 2000 rpm, a compression ratio of 6.5 and 14 air fuel ratio. The results show that, although the energy input is the same, for all cases, the peak mean cylinder temperature is increased and advanced. This is due to the increase of flame propagation speed. However the increase in the mean cylinder temperature is more pronounced with stoichiometric mixture.

Figures (3), (4), and (5) show the effect of air to fuel ratio at different hydrogen blending ratio on the thermal efficiency, the brake mean indicated pressure, and the specific fuel consumption, respectively. The figures are drawn for a naturally aspirated engine with 6.5 compression ratio, and 2000 rpm engine speed.

The thermal efficiency for pure gasoline operation is at maximum value at air to fuel ratio about 16 (lean mixture). It's noticeably that the blended hydrogen improve efficiency, that's due to the increase in the cylinder content temperature. The maximum value of the thermal efficiency slid to the rich side as blended hydrogen increase, the reason behind that is the effect of blended hydrogen on flame speed is greater in the rich mixture than that for lean mixture. The engine thermal efficiency is also improved as the percentage of hydrogen blending is increased reaching maximum at about 20% blending. With further increase in hydrogen blending the thermal efficiency decrease is due to the drop in the volumetric efficiency. The reason behind the drop in the volumetric efficiency as hydrogen blending ratio increase is the low density of hydrogen compared to gasoline that cause a reduction in the mixture density which reduce the volumetric efficiency.

The brake mean indicated pressure is increase as increase mixture richness, that due to increase in amount of fuel bunt. Enlarge the ratio of hydrogen blended increase the indicated mean effective pressure, the reason behind that is the reduction of the time of combustion process due to rate the flame propagation speed, so the pressure diagram becomes close to the ideal diagram. The high blended ratio (about 20%) gives small increased in the brake mean indicated pressure that because of the low density of hydrogen that drop off the volumetric efficiency.

The specific fuel consumption is at minimum value at lean mixture (air to fuel ratio of 16) and decreases as increasing the hydrogen blending ratio, that's due to the increase in engine isothermal efficiency. The minimum specific fuel consumption shift to the hydrogen blending ratio, that's because of the high increase in flame speed for rich mixture rather than the lean mixture.

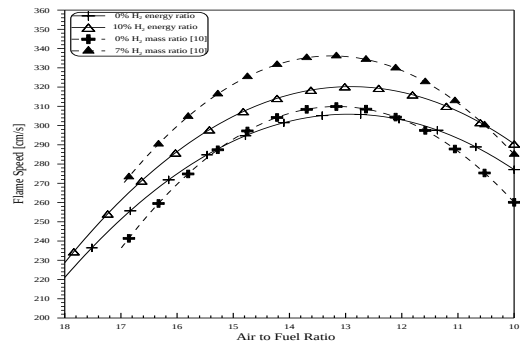


Fig. 1 The Effect of air to fuel ratio on flame speed

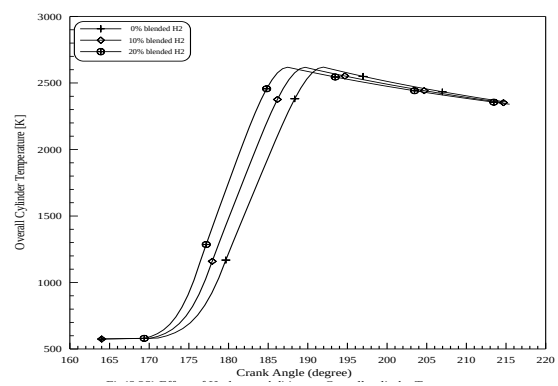


Fig. 2 The Effect of hydrogen addition on overall cylinder temperature

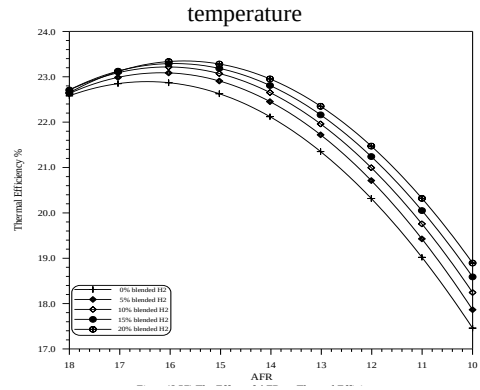


Fig. 3 The Effect of air to fuel ratio on thermal efficiency

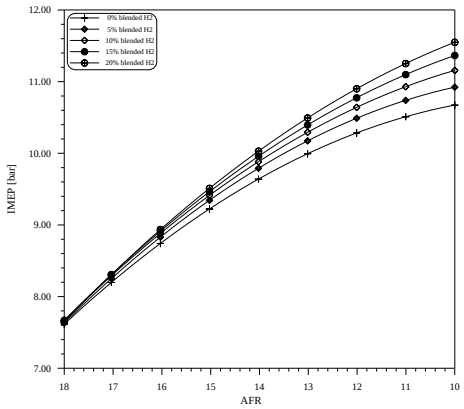


Fig. 4The Effect of air to fuel ratio on brake mean indicated pressure

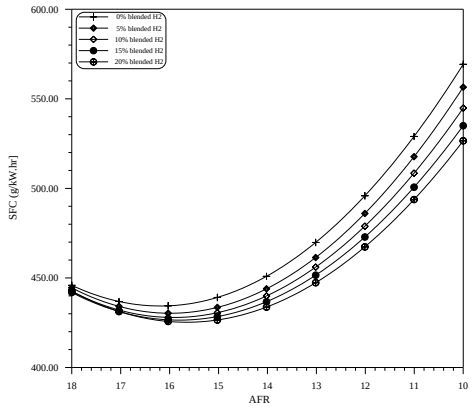


Fig. 5 The Effect of air to fuel ratio on specific fuel consumption

IV. CONCLUSIONS

Based on the results presented and discussed in the present work following conclusions can be drawn: The hydrogen added to gasoline acts as a burning promoter, and expands the range of combustibility of the air fuel mixture and hence leaner mixture can be burnt. The combustion process is improved, and the flame propagation speed enhanced. The blended hydrogen increases the maximum cylinder pressure, zonal temperature up to a blending ratio of 20%. Hydrogen blending duration improves engine efficiency. Hydrogen blending advance the point of maximum cylinder pressure, so that the actual pressure diagram gets closer to the ideal diagram (Otto cycle).

REFERENCES

[1] Y. Liu, K. C. Midkiff, and S. R. Bell, "Development of a Multizone Model for Direct Injection Diesel Combustion", Int. Journal of Engine Res., vol. 5, No. 1, pp. 71-81, 2003.

[2] G. P. Parbhu, B. Nagalingam and K. V. Gopalakrishnan, "Theoretical Study of a Spark Ignited Supercharged Hydrogen Engine", International Journal of Hydrogen Energy, vol. 10, No. 6 pp. 389-397, 1985. http://www.elsevier.com/locate/ijhydene

[3] Haroun A. K. Shahad and Maher S. AL-Baghdadi, "A Prediction Study of the Effect of Hydrogen Blending on the Performance and Pollutants Emission of a Four Stroke Spark Ignition Engine", International Journal of Hydrogen Energy, vol. 24, pp. 783-793, 1999. http://www.elsevier.com/locate/ijhydene

[4] L.M. Das, R. Gulati, and P.K. Gupta, "A Comparative of the Performance Characteristics of a Spark Ignition Engine Using Hydrogen and Compressed Natural Gas as Alternative Fuels", International Journal of Hydrogen Energy, vol. 25, pp. 363-375, 2000. http://www.elsevier.com/locate/ijhydene

[5] Ghazi A. Karim, "Hydrogen as a Spark Ignition Engine Fuel", International Journal of Hydrogen Energy, vol. 28, pp. 569-577, 2003. http://www.elsevier.com/locate/ijhydene

[6] Le Corre Olivier, and Pirotais Frederic, "NOx Emission Reduction of a Natural Gas SI Engine Under Lean Conditions: Comparison of the EGR and RGR Concepts", Proceedings of ICES, 2006 Spring Technical Conference, of the ASME Internal Combustion Engine Division, Aachen, Germany, May 8-10, 2006.

[7] J. F. Cassidy, "Emission and Total Energy Consumption of a Multi cylinder Piston Engine Running on Gasoline and a Hydrogen – Gasoline Mixture", Nasa Technical Note, Nasa TN D-8487, National Aeronautics and Space Administration, Washington, D. C. May, 1977.

[8] S. Furuhama, and Y. Kobayashi, "A Liquid Hydrogen Car With A Two – Stroke Direct Injection Engine and LH2 – Pump", International Journal of Hydrogen Energy, vol. 7, No. 10 pp. 809-820, 1982. http://www.elsevier.com/locate/ijhydene

[9] F. E. Lynch, "Parallel Induction: A Simple Fuel Control Method for Hydrogen Engines", International Journal of Hydrogen Energy, vol. 8, No. 9 pp. 721-730, 1983. http://www.elsevier.com/locate/ijhydene

[10] B. Haragopala, K. N. Shrivastava and H. N. Bhakta, "Hydrogen for Dual Fuel Engine Operation", International Journal of Hydrogen Energy, vol. 8, No. 5 pp. 381-384, 1983. http://www.elsevier.com/locate/ijhydene

[11] K. S. Varde, and G. A. Frame, "Hydrogen Aspiration in a Direct Injection Type Diesel Engine – Its Effect on Smoke and Other Engine Performance Parameters", International Journal of Hydrogen Energy, vol. 8, No. 7 pp. 549-555, 1983. http://www.elsevier.com/locate/ijhydene

[12] Samir M. Abdulhaleem, "Theoretical and experimental investigation of engine performance and emissions of a four stroke spark ignition engine operated with hydrogen blended gasoline". Ph.D. thesis, Al-Mustansiriya University, Baghdad, Iraq, 2007.

[13] M. F. Modest, "Radiative Heat Transfer" Copyright by McGraw-Hill, 1993.

[14] M. N. Ozisik, "Heat Transfer, a Basic Approach" Copyright by McGraw-Hill, 1988.

[15] M. A., Al-Baghdadi, "The Effect of Hydrogen Addition on the Performance and Emissions of a Four Stroke Spark Ignition Engine" M.SC. Thesis, University of Babylon, 1996.

[16] H., Hayami, et al, "Application of a Low Solidity Cascade Diffuser to Transonic Centrifugal Compressor" Tran. ASME, vol.112, pp.25-29, January 1990.

[17] J. D., Mattingly, "Element of Gas Turbine Propulsion" Copyright by McGraw-Hill, 1996.

[18] J. B., Heywood, "Internal Combustion Engine Fundamentals" Copyright by McGraw-Hill, 1989.

[19] S. M. Abdul-Haleem, "Modeling of a Turbocharged Four Stroke Diesel Engine" M.SC. Thesis, University of Babylon, 2001.

[20] G. H. Choi, Y. J. Chung, and S. B. Han, "Effect of Hydrogen Enriched LPG Fuelled Engine with Converted from a Diesel Engine", Proceedings