IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 13, Issue 4 Ver. III (Jul. - Aug. 2016), PP 104-114
www.iosrjournals.org

Energy analysis and the comparison of gasoline direct injection engines performance parameters with port fuel injection engines

Esmaeil Tavanaei Reza Bahoosh Kazerooni
M.A student, Shahid Chamran University of Ahvaz
stavanaee@gmail.com
Assistant professor, Shahid Chamran University of Ahvaz
reaz360@gmail.com

Abstract: *The Analysis of the first law done by thermodynamic or fluid models is a suitable method for evaluating the motor performance and sensitivity of its performance various parameters. This article is aimed at two-regional thermodynamic surveying of gasoline direct injection engines as well as comparing their pressure, fuel consumption, and the work produced in compression and expansion phases in these motors with port fuel injection engines. It would be concluded in this study that taking into account the fuel utilization reduction in direct injection engines, they also produce considerable work compared to port fuel injection motors.*
***Keywords**: Internal combustion engines, gasoline direct injection, two regional, thermodynamic simulation*

I. Introduction

The cycle in which heat is released at a constant volume is called Otto cycle. It is supposed in this research that the combustion has happened quickly and the piston has not moved during the combustion which in this state the combustion starts with a spark and considering this fact, the engine which works on Otto cycle basis is called spark ignition engine. Otto cycle is used to analyze internal combustion engines in which ignition occurs very fast. The cycle in port fuel injection is similar to Otto cycle. Gasoline direct injection engines are a solution for using petrol motors with higher compression ratio in comparison to petrol port fuel injection engines. This paper thermodynamically analyzes the GDI engine and compares the pressure, fuel consumption, and the produced work in compression and expansion stages of these motors with PFI engines. Numerous researches have been done about internal combustion motors using experimental, numerical, and theoretical methods. Ebrahimi (EBRAHIMI, 2010) studied the effects of pressure change and cut-off on the dual cycle. Lin et al. (Lin, 1999) examined the effect of heat transfer from the cylinder wall on the output work in the dual cycle. Rashidi et al. (Rashidi, 2014) surveyed the analysis of the first and second thermodynamic laws on a standard dual cycle while considering the heat dissipation. Mostafa Atamaka et al. (Atamaka, 2011) conducted the limited-time thermodynamic analysis for a modern diesel cycle in 2011. Bahoosh and Sedeh (Bahoosh, 2013) improved the performance parameters of direct injection motor using exergy analysis and examined the effect of biodiesel fuel made of sunflower oil. Jazayeri et al. (Jazayeri, 2008) presented an approach for modelling air-fuel mixture by combining the mixture formation components including fuel pressure, time of injection start, injection velocity, and the place of spray. In another research, the way of air-fuel mixture formation for the injected fuel through the injector with two diffusers in a GDI engine has been surveyed which regarding this issue, Nishida et al. (Nishida, 2012) have examined characteristics of liquid fuel, the fuel injector penetration vapor, and fuel injection angle by 3-D simulation and laser imaging. In the field of experimental studies, Parlak et al. (Parlak, 2005) surveyed the performance optimization of an irreversible dual cycle. They compared the performance of the dual cycle with the diesel cycle at maximum output work. Zhao and Chen (Chen, 2006) determined the internal irreversibility using the expansion and compression efficiency and examined the performance of a dual cycle. The purpose of this article is two-regional thermodynamic analysis of gasoline direct injection engine and the comparison of pressure, fuel consumption, and the work produced in expansion and compression phases for these motors with port fuel injection engines.

II. Research method

Introduction of the thermodynamic model:

One of the most accurate heat transfer calculation models, two-regional modeling has been used in this study. I this method. The cylinder volume is divided to separate controls volume. It is assumed in this type of model that the cylinder volume is divided into two regions of burned and unburned in each moment of the time. The unburned area contains air-fuel mixture and remained gases and the burned part encompasses the products caused by combustion. Thermal range of 300 to 1000° Kelvin is for computing the unburned mixture features and the range of 1000 to 5000° Kelvin is utilized for calculating the burned mixture traits. The thermodynamic cycle presented in this research has three phases of compression, expansion, and combustion during 360-degree movement of the crankshaft.

The governing equations

The law of mass and energy conservation for the contents in the cylinder in this study is written as the following. (Heywood, JB., 1988).

$$\frac{dm}{d\theta} = \frac{dm_u}{d\theta} + \frac{dm_b}{d\theta} \tag{1}$$

$$\frac{d(m_u u_u)}{d\theta} = h_u \frac{dm_u}{d\theta} + \frac{\delta Q_u}{d\theta} - P\frac{dV_u}{d\theta} \tag{2}$$

$$\frac{d(m_b u_b)}{d\theta} = h_b \frac{dm_b}{d\theta} + \frac{\delta Q_b}{d\theta} - P\frac{dV_b}{d\theta} \tag{3}$$

Index u is related to the unburned state of the mixture and index b is related to the burned state of the blend in the cylinder. Furthermore, m, u, h, θ, V, P, and Q are the mass of the cylinder contents, special energy, special enthalpy, the crankshaft angle during rotation, cylinder volume, pressure in the cylinder, and the heat transfer between the combustion area and the cylinder wall, respectively.

The air properties

It is assumed that mixture of the surrounding air contains 21% of oxygen and 79% of nitrogen. Each of the polynomials should be determined in a specified temperature in order to find the air thermodynamic attributes. These thermodynamic features are processed as some polynomials in two thermal states of below 1000° Kelvin for computing the traits of the unburned mixture and over 1000° Kelvin for the burned blend. Ferguson (CR, 198).

$$\frac{C_P}{R} = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4 \tag{4}$$

$$\frac{h}{RT} = a_1 + \frac{a_2}{2}T + \frac{a_3}{3}T^2 + \frac{a_4}{4}T^3 + \frac{a_5}{5}T^4 + \frac{a_6}{T} \tag{5}$$

$$\frac{S}{RT} = a_1 lnT + a_2T + \frac{a_3}{2}T^2 + \frac{a_4}{3}T^3 + \frac{a_5}{4}T^4 + a_7 \tag{6}$$

The values of ai are accessible in the two separate tables (Ferguson, CR, 198). In these relations, R is the universal gas constant and it is equal to 8/314 J/mol.K.

The fuel thermodynamic properties

Due to the fact that the cylinder input mixture contains air and fuel, the fuel characteristics should be surveyed.Below polynomials have been introduced for calculating the fuel thermodynamic traits in which the temperature is in terms of Kelvin and coefficients of a1 to a7 are determined through table 1.

$$\frac{C_P}{R} = a_1 + a_2T + a_3T^2 + a_4T^3 + a_5\frac{1}{T^2} \tag{7}$$

$$\frac{h}{RT} = a_1 + \frac{a_2}{2}T + \frac{a_3}{3}T^2 + \frac{a_4}{4}T^3 - a_5\frac{1}{T^2} + \frac{a_6}{T} \tag{8}$$

$$\frac{S}{RT} = a_1 \ln T + a_2T + \frac{a_3}{2}T^2 + \frac{a_4}{3}T^3 - \frac{a_5}{2}\frac{1}{T^2} + a_7 \tag{9}$$

The ai values are accessible in table 1 for temperatures below and over 1000° Kelvin.

Table 1: ai values for different types of fuels

a_7	a_6	a_3	a_2	a_1	Fuel
8.873728	-9.930422E+03	-1.048592E-06	7.871586E-03	1.971324	Methane
1.545E+01	-3.5880E+04	-1.8801E-05	6.0977E-02	4.0652	Gasoline
-1.7879	-1.9385E+04	-3.6858E-05	1.1954E-01	7.9710	Diesel
1.50884+01	-2.525420E+04	-3.624890E-06	1.262503E-02	1.779819	Methanol
1.917126E+01	-1.026351E+04	-8.142134E-06	2.087101E-02	1.412633	Nitromethane

Combustion products

In this study, the combustion products should be separately furnished for two states with temperatures below and over 1000° Kelvin.

Low temperature combustion products

In temperatures less than 1000 Kelvin, the products are supposed to have six types which are as the followings:

$$C_\alpha H_\beta O_\gamma N_\delta + \frac{a_s}{\phi}\left(O_2 + 3.76N_2\right) \rightarrow n_1CO_2 + n_2H_2O + n_3N_2 + n_4O_2 + n_5CO + n_6H_2 \quad (10)$$

High temperature combustion products

In 1975, Olikara and Borman presented a balance based on the equilibrium constant method for determining the combustion products. (Olikara, C. and G. Borman, 2008)

$$C_\alpha H_\beta O_\gamma N_\delta + \frac{a_s}{\phi}\left(O_2 + 3.76N_2\right) \rightarrow n_1CO_2 + n_2H_2O + n_3N_2 + n_4O_2 + n_5CO + n_6H_2 + n_7H + n_8O + n_9HO + n_{10}NO \quad (11)$$

It should be mentioned that in the equation above Φ is the equivalence ratio of fuel to air which is defined as the following.

$$\phi = \frac{\left(\frac{F}{A}\right)_{actual}}{\left(\frac{F}{A}\right)_s} \quad (12)$$

Mass of the burned gases

Two models of Current study and Weibe Function (Soma 2008) are used in the surveys related to spark ignition engines for simulating the burning ratio of the mixture in the cylinder.

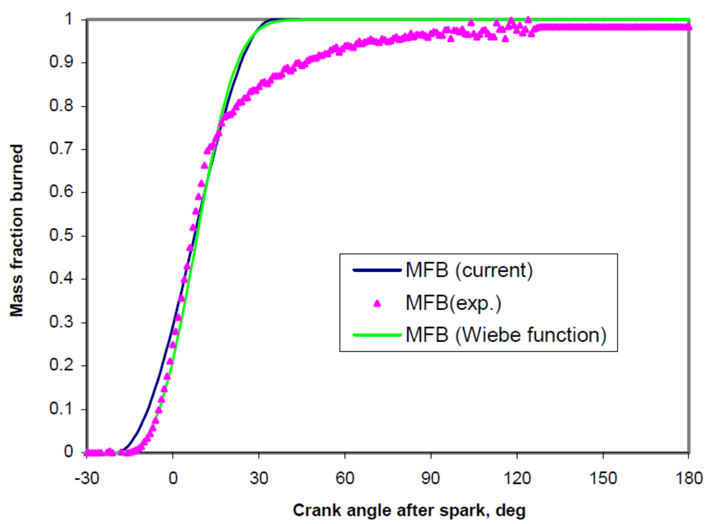

Figure 1: The comparison of experimental combustion ratio with combustion coefficient in analytical methods (Ferguson, Cr., 198)

The equation below shows the calculation of the burned gas mass.

$$x = \frac{m_b}{m} \quad (13)$$

By differentiating equation 13, the combustion ratio in each degree of crankshaft would be furnished.

$$\frac{dx}{d\theta} = \frac{\pi}{2\theta_b} \sin\left[\frac{\pi(\theta - \theta_s)}{\theta_b}\right] \quad (14)$$

In above relations, sθ is the sparkling start angle and bθ is the sparking duration.

Heat transfer

Woschni model has been used in this study. At the end, the heat released is furnished by equation 15.

$$Q_i = hA_i(T_i - T_w) \qquad i=(u,b) \quad (15)$$

T_w is the cylinder wall temperature which is considered 420° K in most of the researches.

Woschni approach is used to compute the h in which p is pressure, u is the piston speed, B is the piston diameter, and T is the cycle temperature in each moment. Coefficients of C_1 and C_2 are chosen from table 2 according to the engine work phases (Woschni,1967).

Table 2: Coefficients of the heat transfer process triple stages in Woschni method.

Period	C_1	C_2
Compression	2.28	0
Combustion and expansion	2.28	0/00324

$$h = 130P^{0.8}.u^{0.8}.B^{-0.2}.T^{-0.55} \quad (16)$$

$$P_m = P_r\left[\frac{V_r}{V}\right]^K \quad (17)$$

$$u = C_1.v_p + C_2\frac{V_s.T_r}{P_r.V_r}(P - P_m) \quad (18)$$

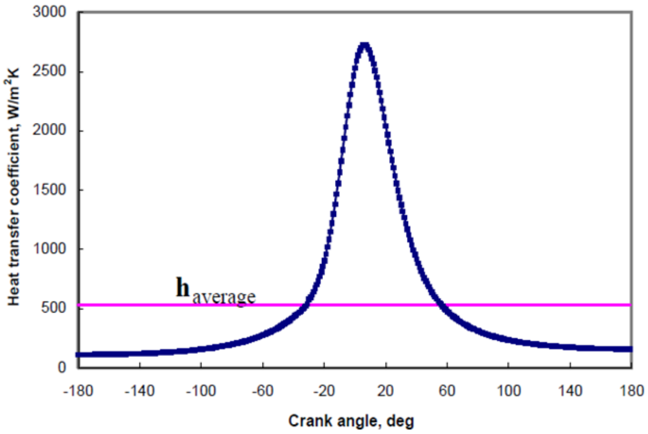

Figure 2: The changes of heat transfer coefficient in terms of the crankshaft degree (Ferguson, CR., 198)

Computing the work done based on the crankshaft degree

One of the engine performance parameters is the work presented by it. The work is generated by the pressure of the gases in the combustion area of the cylinder:

$$\frac{dW}{d\theta} = P\frac{dV}{d\theta} \quad (19)$$

DOI: 10.9790/1684-130403104114

Calculating the cylinder volume

The cylinder volume in each degree of the crankshaft rotation is furnished as the following.

$$V = V_0\left[1+\frac{r-1}{2}\left\{1-\cos\theta+\frac{1}{\varepsilon}\left[1-(1-\varepsilon^2\sin^2\theta)^{1/2}\right]\right\}\right] \quad (20)$$

In addition, ε is obtained through the equation below.

$$\varepsilon = \frac{S}{2l} \quad (21)$$

In the above relations, r, S, and V0 are compression ratio, stroke, and dead volume, respectively. The stroke and crankshaft have the below relation with each other.

$$L = 2a \quad (22)$$

Differentiating equation 20 in terms of θ, the following equations would be achieved.

$$\frac{dV}{d\theta} = \frac{V_0(r-1)}{2}\left[\sin\theta+\frac{\varepsilon}{2}\sin 2\theta\left(1-\varepsilon^2\sin^2\theta\right)^{-1/2}\right] \quad (23)$$

$$V = V_0\left[1+\frac{r-1}{2}\left\{1-\cos\theta+\frac{1}{\varepsilon}\left[1-(1-\varepsilon^2\sin^2\theta)^{1/2}\right]\right\}\right] \quad (24)$$

9.1. The pressure and temperature in the cylinder

For such modellings, Ferguson (Ferguson, CR., 1986) could mathematically state the pressure in the cylinder and the burned and unburned mixtures temperatures.

$$\frac{dP}{d\theta} = \frac{f_1+f_2+f_3}{f_4+f_5} \quad (25)$$

(26)

$$f_2 = \frac{h}{m\,\omega}\left[\frac{v_b}{C_{P_b}}\frac{\partial \ln v_b}{\partial \ln T_b}\frac{\sum_{i=h,p,l} A_{bi}\left(T_b - T_{wi}\right)}{T_b} + \frac{v_u}{C_{P_u}}\frac{\partial \ln v_u}{\partial \ln T_u}\frac{\sum_{i=h,p,l} A_{ui}\left(T_u - T_{wi}\right)}{T_u}\right] \quad (27)$$

$$f_3 = -\left(v_b - v_u\right)\frac{dx}{d\theta} - v_b\frac{\partial 1n v_b}{\partial 1n T_b}\frac{h_u - h_b}{c_{pb}T_b}\left[\frac{dx}{d\theta} - (x - x^2)\frac{C_b}{\omega}\right] \quad (28)$$

$$f_5 = x\left[\frac{v_b^2}{C_{p_b T_b}}\left(\frac{\partial \ln v_b}{\partial \ln T_b}\right)^2 + \frac{v_b}{P}\frac{\partial \ln v_b}{\partial \ln P}\right] \quad ,$$

(30و29)

(31)
$$\frac{dT_b}{d\theta} = \frac{-h\sum_{i=h,p,1} A_{bi}\left(T_b - T_{wi}\right)}{m\,\omega C_{pb}\,x} + \frac{v_b}{C_{pb}}\frac{\partial \ln v_b}{\partial \ln T_b}\frac{dP}{d\theta} + \frac{h_u - h_b}{x C_{pb}}\left[\frac{dx}{d\theta} - (x - x^2)\frac{C_b}{\omega}\right]$$

(32)
$$\frac{dT_u}{d\theta} = \frac{-h\sum_{i=h,p,1} A_{ui}\left(T_u - T_{wi}\right)}{m\,\omega C_{pu}\left(1-x\right)} + \frac{v_u}{C_{pu}}\frac{\partial \ln v_u}{\partial \ln T_u}\frac{dP}{d\theta}$$

Furthermore, the following relations were presented for computing the temperatures of the burned and unburned mixtures.

In the above relations, u and b indexes are related to the unburned and burned states of the mixture. In addition, A parameter is the burning cross section, **ω** is piston velocity, TW is the cylinder wall temperature and $\frac{dx}{d\theta}$ is the burning rate in terms of the crankshaft rotation degree which would be explained later.

Combustion process in gasoline direct injection engines:

Three states are examined in this paper which ignition occurs in GDI engines between the rotations of less than 3000 RPM to 4500 RPM in these three situations.

Layered filling state

In the range of rotation speed of 3000 RPM with low output torque, the engine performs in the layered charge state.In this situation, the injector injects the fuel during the compression stroke and a bit before the spark plug ignition.

Homogenous and lean burning

During the changing time between layered and homogenous states, the motors can function with the diluted homogenous air-fuel ratio.

Homogenous and layered filling

In this state, whole combustion chamber is filled with diluted homogenous air-fuel blend. This mixture is made by a little fuel injection in the intake stroke. The remaining fuel is injected in the compression stroke (two phases of injection). In this case, a richer mixture is created around the spark plug.

Findings

Validation

For validating, the code of the pressure curve in the cylinder in 3000 RPM was compared with experimental findings obtained fromIran Khodro designing, research, and production company (APCO). As it can be seen in figure 3, the simulation and experimental results have an acceptable compliance.

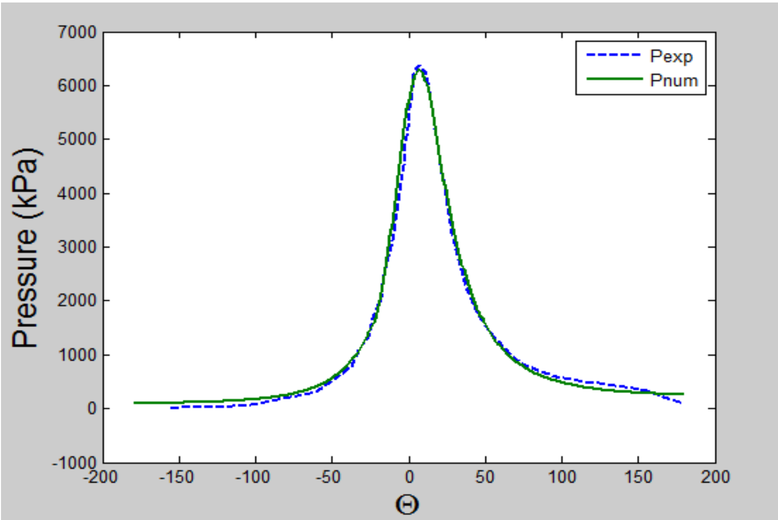

Figure 3: The validation of the diagram obtained from thermodynamic modelling with IPCO experimental results

The analysis of the first law of thermodynamic

Forecasting capability of gas pressure in the combustion chamber Ignition in layered state:

In motor rotation of 600 to 3000 RPM, the burning in GDI engines is layered. In this case, the ignition is considered with 50 to 70 percent of additional air. As it is clear in figure 4, in layered condition, the port fuel injection engine which works with equivalence ratio of 9 close to stoichiometricstate have an equal maximum pressure in comparison to the state in which GDI motors function with equivalence ratio of 5, thus this work not only reduces the fuel consumption considerably (approximately by half) but also significantly decreases the production of Knox pollutants.

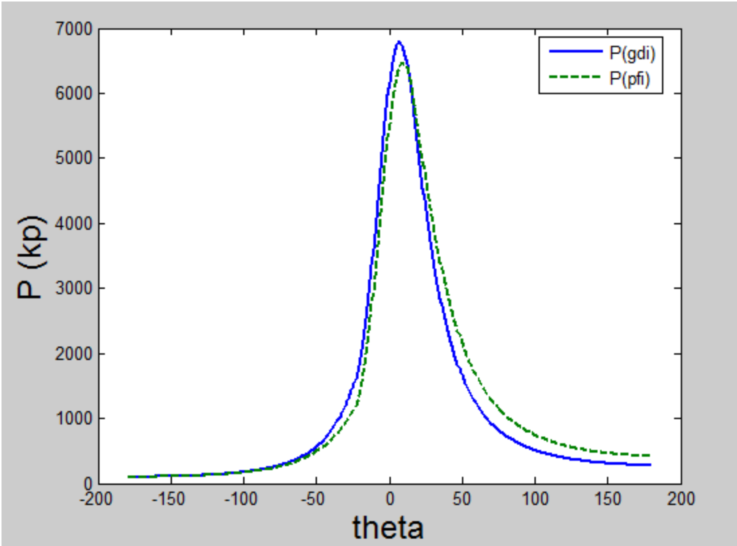

Figure 4: The comparison of GDI and PFI engines in layered situation

Homogenous and lean combustion mode:

In this case, the burning is considered with 40 percent of additional air. In this mode, the equivalence ratio is roughly 7 while in GDI engines, it is near to stoichiometric state.

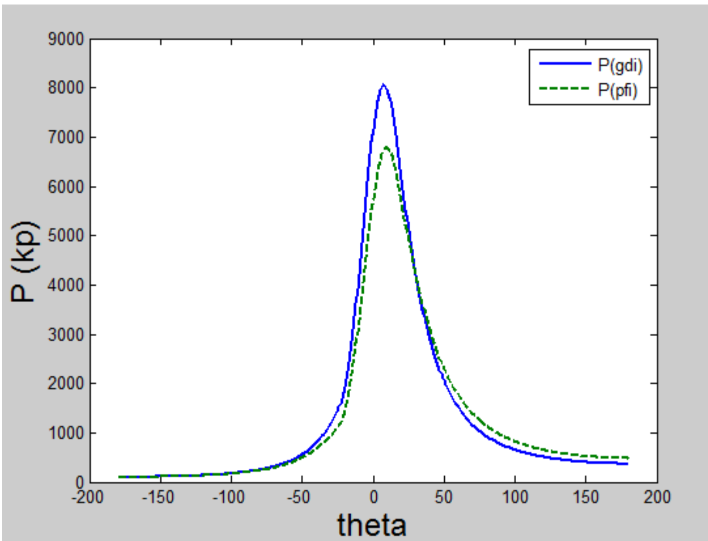

Figure 5: The comparison of GDI and PFI engines in homogenous-diluted mode

The homogenous layered state:

In this survey, equivalence ratio of 7 has been used in this state. In this mode also, as it is depicted in figure 6, the maximum pressure in direct injection is higher.

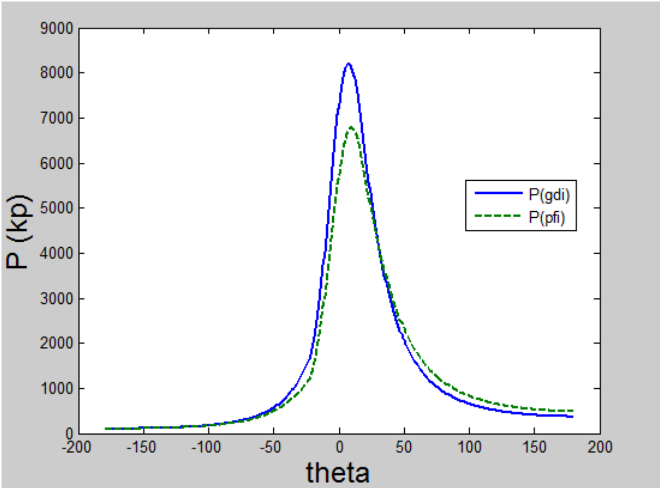

Figure 6: The comparison of GDI and PFI engines in homogenous layered mode

The analysis of the work produced in GDI motors in comparison to PFI engines:

Layered mode:

As it is demonstrated in figures 7 and 8, the work produced in direct injection mode is 380J while the produced work in port fuel injection is 550J. It indicates that GDI motors consume gasoline 5 times more than PFI engines and produce work 69 time more than them which is a great advantage.

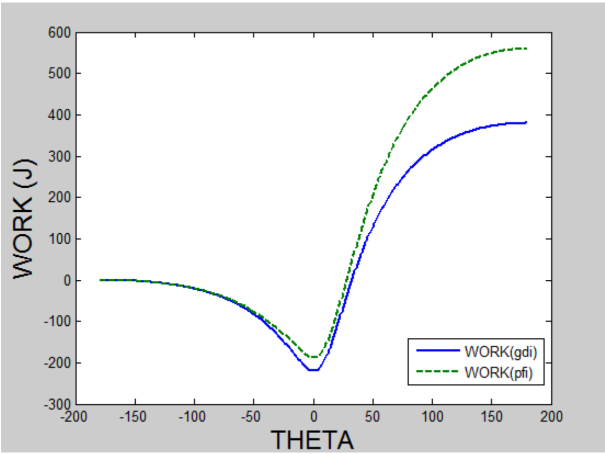

Figure 7: The comparison of GDI motor workin layered mode with PFI engine

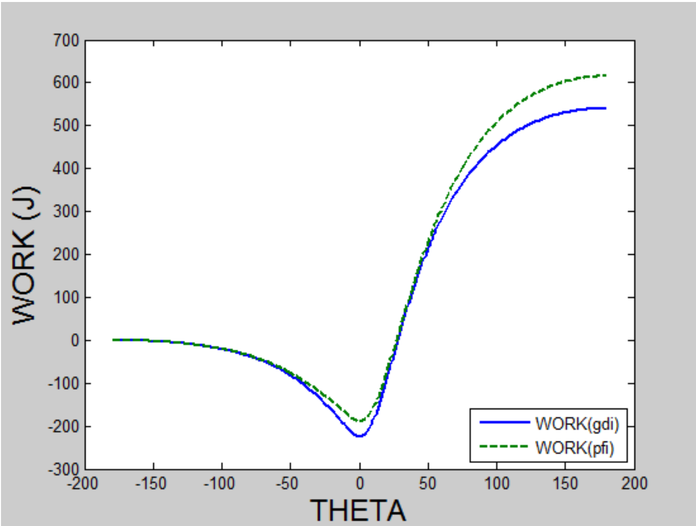

Figure 8: The comparison of the work changes in GDI engines in layered mode with PFI motors

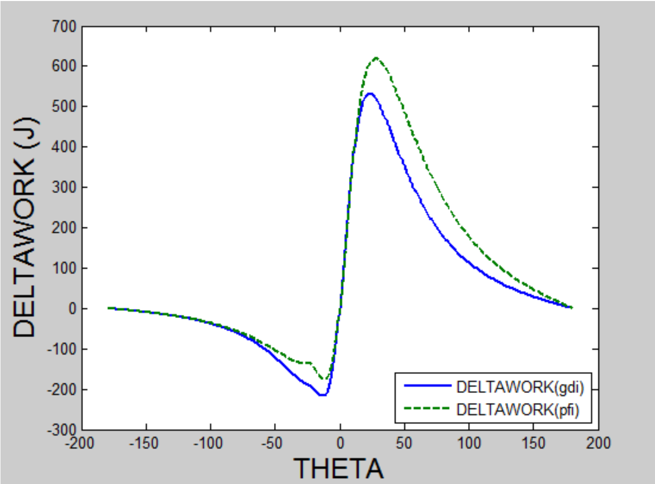

Figure 9: The work comparison of GDI engines with PFI engines in diluted homogenous mode

Diluted homogenous mode:

As presented in figures 9 and 10, similar to the layered mode, in diluted homogenous situation the work produced in GDI condition is less than PFI mode. The produced work in PFI and GDI states are 600J and 540.3J, respectively. However, the GDI engine fuel consumption ratio in diluted homogenous mode to the PFI engine is 7 and the ratio of the GDI produced work to the PFI motor is 87. As a result, this comparison shows the GDI engine predominance.

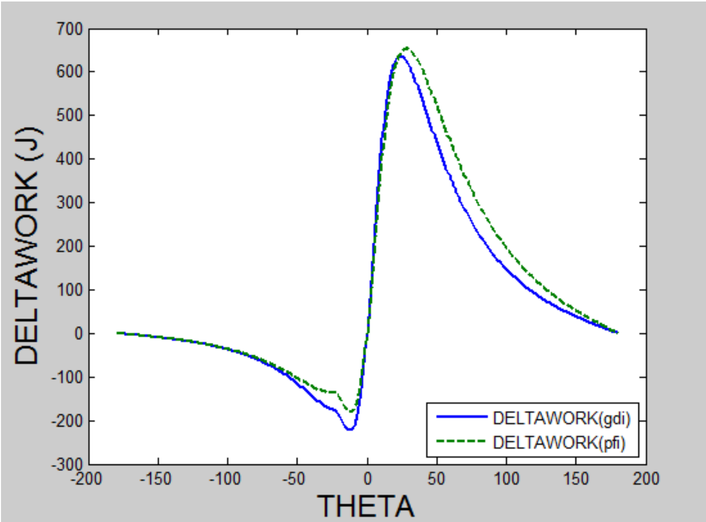

Figure 10: The comparison of the work changes in GDI engines with PFI motors in diluted homogenous mode

Layered homogenous mode:

The work produced in PFI and GDI states are 2.64KJ and 2.42KJ, respectively. However, the ratio of GDI engines fuel consumption in the layered homogenous mode is ------. Furthermore, the produced work ratio of GDI motors to PFI motors is 0.92.

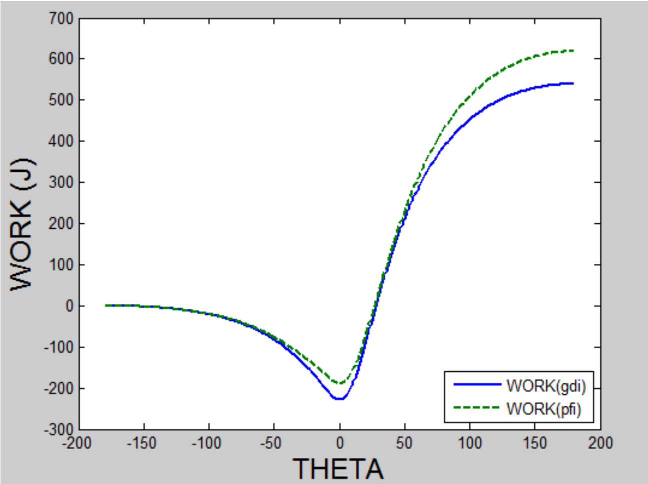

Figure 11: The comparison of the work produced in GDI engines in layered homogenous mode with PFI motors

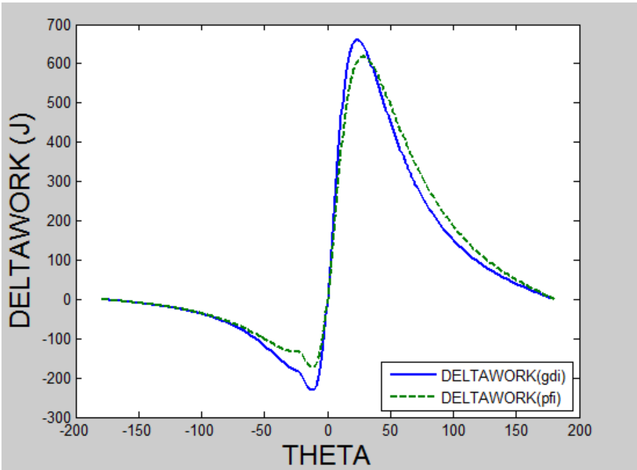

Figure 12: The comparison of the work changes in GDI engines in layered homogenous mode with PFI motors

III. Discussion and conclusion

Presented in this research are thermodynamic modelling of gasoline direct injection engines and the comparison of parameters including pressure, fuel consumption, and the produced work with port fuel injection engines. The diagram of the pressure in terms of the crankshaft degree has been drawn and the pressure comparison shows that the maximum pressure is higher in direct injection mode. Results signify that GDI engines fuel consumption is considerably lower than PFI motors. The difference in the work produced in both modes is not very significant while the fuel utilization of GDI state is lower. Regarding the fact that this survey has been conducted in the field of GDI engines thermodynamic modelling, in order to continue this study, it is recommended to the researchers to model other combustion modes in GDI engines and evaluate more performance parameters.

References

[1]. R. Ebrahimi, "Preformance Analysis of a Dual Cycle Engine with Consideration of Pressure Ratio and Cut-Off Ratio", ACTA PHYSICA POLONICA A, Vol. 118 (2010)
[2]. mostafa atmaka, metin gumus, Abdullah demir."Comparative Thermodynamic Analysis of Dual Cycle Under Alternative Conditions(2011)
[3]. J. Lin, L.Chen, C.Wu, F.Sun, Int.J.Energy Res, 23, 765(1999)
[4]. M.M. Rashidi, A. Hajipour and A. Fahimirad, "First and Second-Laws Analysis of an Air-Standard Dual Cycle With Heat LossConsideration"International Journal of Mechatronics Electrical and Computer Technology, Vol. 4(11), Apr. 2014
[5]. Ferguson, CR.,1986, Internal combustion engines. New York, Wiley
[6]. Nishida, K., Tian, J., Sumoto, Y., Long,W Sato, K and Yamakawa, M., "An experimental andnumerical study on sprays injected from two-hole nozzles for DISI engines", Contents Lists Availableat ScienceDirect, Fuel 88, pp. 1634–1642.
[7]. Jazayeri, S. A., Zia Basharhagh M., BavandporM., and Keshavarz M., "Analysis of air fuel mixture,formation in gasoline direct injection SI engine,The Journal of Engine Research, vol. 13, pp. 45-52,2008.
[8]. Hongqing, F and Li. Huijie, 2010, Second-law analyses applied to a spark ignition engine under surrogate fuels for gasoline, 3551-

3556.
[9]. Heywood, JB., 1988, Internal combustion engine fundamentals NewYork: McGraw-Hill.
[10]. Olikara, C. and G. Borman, 2008, A Computer Program for Calculating Properties of equilibrium Combustion Products with Some Applications to I.C. Engines. SAE Technical Paper Series 750468.
[11]. Ferguson, CR.,1986, Internal combustion engines. New York, Wiley.
[12]. Soma, S and A. Datta, 2008, Thermodynamic irreversibilities and exergy balance in combustion processes, Progress in Energy and Combustion Science, 351-376.
[13]. Annand, WJD., 1963, Heat transfer in the cylinders of reciprocating internal combustion engines.
[14]. Woschni, G., 1967, A universally applicable equation for theinstantaneous heat transfer coefficient in the internalcombustion engine, Society of Automotive Engineers Inc.
[15]. *Jun Harada, Tsutomu Tomita, Hiroyuki Mizuno, Zenichiro Mashiki, and Yasushi Ito –"Development of Direct Injection Gasoline Engine".1997*