Amirkabir Journal of Mechanical Engineering

Amirkabir J. Mech. Eng., 51(4) (2019) 1-4
DOI:

Energy and Exergy Analyses of the Performance of a Spark Ignition Engine Using Two-Zone Combustion Model

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ABSTRACT: Relying on numerical methods, it is possible to estimate the performance of an engine precisely, and to evaluate the optimization methods of the engine. Almost 30% of the fuel energy is dissipated through the cylinder walls, therefore, accurate calculation of heat transfer rate is necessary to yield reliable results from the simulation model. The thermodynamic model in this paper, that is developed to optimize the performance of a spark ignition engine, utilizes the two-zone combustion model that is suitable for this type of engine. The results indicated that increasing the compression ratio by 18% leads to almost 2% lower exergy destruction due to the engine exhaust and heat transfer to the engine body, however, the engine power output increased by about 4%. Also, it is concluded that the maximum energy and exergy efficiencies are obtained when combustion takes place 5 degree before top dead center.

Review History:

Received:
Revised:
Accepted:
Available Online:

Keywords:

Thermodynamic model
Spark ignition engine
Energy and Exergy analysis
Heat transfer and burning rate
Two-zone model

1- Introduction

Computer analyses of power generation cycles have been developed at various levels from thermodynamic (zero-dimensional) models to three-dimensional models based on numerical analysis. Thermodynamic methods are the simplest method for modeling these cycles that provide accurate and timely solutions if approved by empirical models [1]. Among the thermodynamic methods, the two-zone combustion model, especially in spark-ignition engines, predicts the consistent results with experimental data that can be used to evaluate the engine performance from the energy and exergy points of view.

Razmara et al. [2] developed a control model based on exergy in which a Homogeneous Charge Compression Ignition (HCCI) model is used. The results showed that using an optimization control model the fuel consumption and exergy reduces by 7.7% and 8.3%, respectively. Nemati et al. [3] modeled the effects of various combustion biofuels in a compression ignition engine with the multi-zone combustion model. They showed that the volumetric composition of 20% bio-diesel fuel with diesel has the highest exergy efficiency. Bridjesh et al. [4] investigated the effect of compression ratio in a compression ignition engine. The results indicate that by increasing the compression ratio from 13.3 to 17.5, energy efficiency and exergy efficiency increase by 5% and 4.24%, respectively.

The purpose of this study is to study the thermodynamic simulation of different models of heat transfer in cylinders using a two-zone combustion engine model of a spark ignition engine with natural gas fuel. The simulation results are compared with experimental results, and the effects of compression ratio and the starting angle of combustion on the energy efficiency and exergy are investigated. A thermodynamic model is developed using a computer code in MATLAB software. In fact, the innovation of this research is based on the energy and exergy analyses of a spark-ignition engine with natural gas and the evaluation of the effect of the compression ratio and the ignition start time on engine performance.

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2- Methodology

Fig. 1 shows a schematic view of the model. The hypotheses used in this thermodynamic model are [5]:

- During the suction process, the compression, expansion and discharging of the cylinder contents are uniform in terms of composition and thermodynamic properties.
- During combustion, the contents of the cylinder are divided into two parts; burnt and unburned, separated by the flame front.
- The calculation of fluid properties was performed using the fitting of a nine-degree function of the datasheet of the properties [6].
- Pressure is considered the same for both zones.
- The heat transfer between the burned and unburned zones is ignored.

In the suction process, regardless of viscous loss, the pressure of the inner cylinder is assumed to be equal to the ambient pressure, and the operating fluid temperature changes due to the heat transfer with the cylinder walls, piston and cylinder heads. Since the temperature of the gases entering the suction process is less than the temperature of the inner walls of the cylinder, the cooling effect of the new fluid is taken into

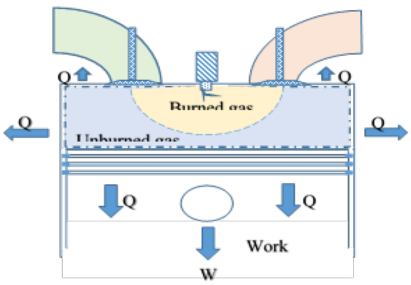

Fig. 1. The schematic view of the two-zone combustion model

account at each interval and the new temperature is calculated using the first law of thermodynamic.

$$dE = dQ - dW \tag{1}$$

In the compression cycle, the calculation of the temperature and pressure of the gases in the cylinder as an isentropic compression process was performed at each step assuming a complete gas and taking into account the properties of each molar component of the burned and unburned products. Then the effect of heat transfer with the walls on the operating fluid was considered. In order to model the combustion, the temperature of the compression is modified according to the amount of heat that is predicted from the amount of fuel injected at each time interval by the Wibe function, in fixed volume, and ultimately, the effect of heat transfer is considered.

The Newtonian cooling method is used, which is an accepted method in the zero-dimensional models [7]:

$$Q_{wi} = h_g A_w \left(T_g - T_w\right) \tag{2}$$

Many studies about heat transfer have been published, which provide several relationships for determining heat transfer coefficient [8].

Exergy of a system is the most accessible amount of the work which is obtained during the interaction with the system to reach thermal, mechanical, and chemical equilibrium conditions. In general, the exergy of a closed system, with the exception of kinetic and potential energy changes, is divided into two thermal and chemical parts as follows:

$$\mathrm{Ex}_{\mathrm{system}}^{\mathrm{th}} = \left(\mathrm{U}-\mathrm{U}^0\right) + \mathrm{P}_0\left(\mathrm{V}-\mathrm{V}^0\right) - \mathrm{T}_0\left(\mathrm{S}-\mathrm{S}^0\right) \tag{3}$$

$$Ex_{system}^{ch} = T_0 \sum_{i=1}^{k} \bar{R} y_i Ln\left(\frac{y_i}{y_i^0}\right) \tag{4}$$

The exergy loss in the internal combustion engine is generally found in two parts. A large part of the exergy is lost due to the heat loss of the engine exhaust. Another part of the exergy is lost through chemical mixing. Exergy of combustion products in the exhaust is calculated from the following equation:

$$Ex_{exhaust} = \sum_{i=1}^{k} H_i(T_e) - H_i(T_o) - T_0\left[S_i(T_e) - S_i(T_o) - RLn\frac{P_e}{P_0}\right] + T_0 \bar{R} y_i Ln\left(\frac{y_i}{y_i^0}\right) \tag{5}$$

The rate of exergy changes is written in terms of the crank angle α, as follows [9]:

$$\frac{dEx_{tot}}{d\alpha} = \left(1-\frac{T_0}{T}\right)\frac{dQ}{d\alpha} - \left(\frac{dW}{d\alpha} - P_0\frac{dV}{d\alpha}\right) + \frac{m_f}{m_{tot}}\frac{dX_b}{d\theta} ex_{f,ch} - \frac{dI_{comb}}{d\alpha} \tag{6}$$

The irreversibility of the engine is equal to the difference between the exergy input and the exergy output.

$$I_{engine} = Ex_{fuel} - Ex_{work} - \left(Ex_{exhaust} - Ex_{intake}\right) - Ex_{heat} - Ex_{ub} \tag{7}$$

Ultimately, the exergy efficiency is equal to the exergy ratio of the output work to the chemical exergy of the input fuel, which is almost equal to the lower heating value of the fuel:

$$\eta_{ex} = \frac{Ex_{work}}{m_f \times ex_{f,ch}} \tag{8}$$

3- Discussion and Results

Fig. 2 presents the effect of temperature on specific heat at the constant pressure of the fluid. It is observed that the presence of different molar components in the unburned and burnt

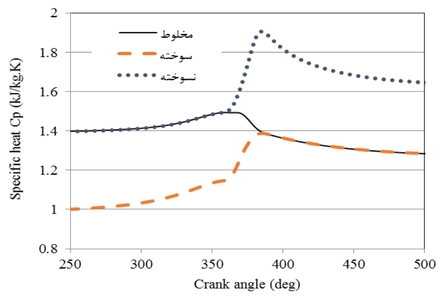

Fig.2. The specific heat of the operating fluid at constant pressure in terms of crank angle

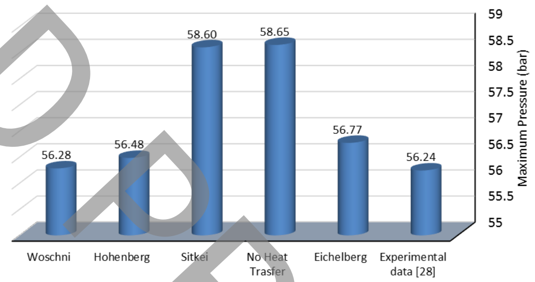

Fig. 3. Maximum operating temperature of the fluid

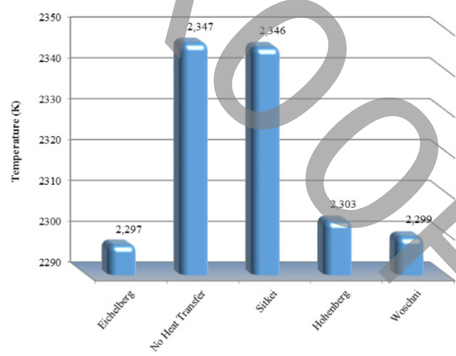

Fig. 4. Maximum operating pressure of the cycle

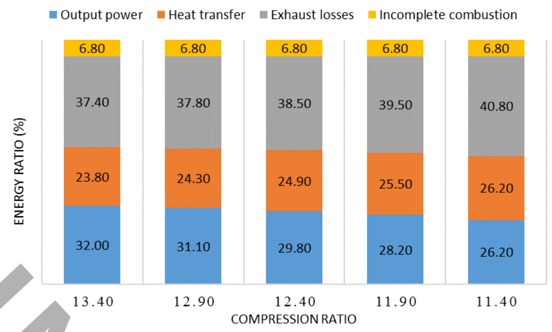

Fig. 5. Distribution of exergy ratio versus compression ratio

fluid causes a difference in the heat of the two zones.

In Figs3 . and 4, the temperature and pressure of the operating fluid are compared according to various heat transfer models. Considering the heat transfer, maximum temperature and maximum pressure decrease comparing with the non-heat transfer state in Woschni model by 2% and 4%, in the Hohenberg model by 1.9 and 3.7%, in the model Sitkei 0.043 and 0.058% and in the Eichelberg model by 2.13% and 3.2%, respectively. As shown in Figs. 3 and 4, one of the weaknesses of the Sitkei model is the lack of accurate calculation of the heat transfer rate compared with other models.

Fig5 . shows the distribution of the exergy ratio of the output to the total exergy input terms of the compression ratio. Exergy analysis of engine with a compression ratio of 12.9 indicates that 30.5% of the input exergy was converted to work, 18.8% of the exergy was destructed due to heat loss through exhaust and the exiting smoke, 14% is wasted through the heat transfer of the gas and the operating fluid to the wall of the closure, the liners and the waste pine, about 6% is wasted due to incomplete combustion of the fuel and 30.7% of the exergy of the input fuel was destroyed and lost. Increase of the compression ratio in the engine leads to increase of the exergy efficiency, slightly.

4- Conclusions

In this study, the use of two-zone combustion model provides the possibility of accurately simulating the properties of the operating fluid in the combustion cycle. The results of thermodynamic analysis showed that energy and exergy efficiency increase with increasing the compression ratio. The highest exergy and energy efficiency occurs when the combustion starts at 5°C before top dead center. The deviation to the positive and negative numbers for the start of ignition timing reduces energy efficiency and exergy, but deviations toward positive numbers, with a greater intensity, reduces efficiency. Maximum energy and exergy efficiency is 31.10 and 30.50%, respectively, when the ignition is carried out at 5 degrees before the top dead center. With increasing the compression ratio, energy efficiency and exergy efficiency increase, but the rate of increase in energy efficiency and exergy decreases. Increasing the compression ratio from 11.4 to 11.9 increases the exergy efficiency by 0.5%, but increasing the compression ratio from 12.9 to 13.4 only results in 0.2% increase in exergy efficiency.

References

[1] Farahat, S., Naserian, M. M., Sarhadi, F., 2017. “Performance optimization of an irreversible Brayton cycle, and proposing new definitions for 2nd law efficiency and exergy”. Amirkabir Journal of Mechanical Engineering, 10.22060/mej.2016.713.

[2] Razmara, M., Bidarvatan, M., Shahbakhti, M., Robinett, R., 2016. “Optimal exergy-based control of internal combustion engines”, Applied Energy, 183, pp. 1389-1403

[3] Nemati P., Jafarmadar S., Taghavifar H., 2016. “Exergy analysis of biodiesel combustion in a direct injection compression ignition (CI) engine using quasi-dimensional multi-zone model”. Energy, 115, pp. 528-538.

[4] Bridjesh, P., Arunkumar, G., 2015. “Energy and Exergy Analysis of Diesel Engine by Varying Compression Ratio”, International Journal of Chemical Sciences, 13(2).

[5] Feng, H., Liu, D., Yang, X., An, M., Zhang, X., Zhang W., 2016. “Availability analysis of using iso-octane/n-butanol blends in spark-ignition engines”, Renewable Energy, 96, pp. 281-294.

[6] Ibrahim, A., Bari, S.,2008.”Optimization of a natural gas SI engine employing EGR strategy using a two-zone combustion model”, Fuel, 87(10), pp.1824-1834.

[7] Ph. D. thesis, Ollivier, E., 2006 “Contribution to the characterization of heat transfer in spark ignition engines, Application to knock detection”, ENSTIM de Nantes, France.

[8] Chang, J., Güralp, O., Filipi, Z., Assanis, D.N., Kuo, T. W., Najt, P., Rask, R., 2004. “New heat transfer correlation for an HCCI engine derived from measurements of instantaneous surface heat flux”, 0148-7191, SAE Technical Paper.

[9] Sezer, I., Altin, I., Bilgin, A., 2009. “Exergetic Analysis of Using Oxygenated Fuels in Spark-Ignition (SI) Engines”. Energy & Fuels, 23, pp. 1801-1807.

Please cite this article using:

E. Baniasadi, H. Abbasi Varzaneh, Energy and Exergy Analyses of the Performance of a Spark Ignition Engine Using Two-Zone Combustion Model, *Amirkabir J. Mech. Eng.*, 51(4) (2019) 1-4.

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