Performance of an Otto engine with volumetric efficiency

Rahim Ebrahimi[1], Davood Ghanbarian[1], Mahmoud Reza Tadayon[2]

[1.] Department of Agriculture Machine Mechanics, Shahrekord University, P.O. Box 115, Shahrekord, Iran
[2] Department of Agronomy and plant breeding, Shahrekord University, P.O. Box 115, Shahrekord, Iran
Rahim.Ebrahimi@gmail.com

Abstract: In this paper, the performance of an Otto engine is evaluated under variable volumetric efficiency. Finite-time thermodynamics is used to derive the relations between power output and thermal efficiency at different compression ratio and volumetric efficiency for an air-standard Otto cycle. The effect of the volumetric efficiency on the irreversible cycle performance is significant. It was found that the effect of volumetric efficiency on the cycle performance is obvious, and they should be considered in practice cycle analysis. The conclusions of this investigation are of importance when considering the designs of actual Otto engines. [Journal of American Science 2010;6(3):27-31]. (ISSN: 1545-1003).

Keywords: Volumetric efficiency; Irreversibility; Analysis; Performance; Otto cycle

Nomenclature

B	constants related to heat transfer
c_p	specific heat at constant pressure
c_v	specific heat at constant volume
$fmep$	friction mean effective pressure
Q_{in}	heat added to the working fluid
Q_{out}	heat rejected by the working fluid
Q_{LHV}	lower heating value of the fuel
R_{air}	air constant of the working fluid
$\dot{m}_f$	mass flow rate of the fuel
$\dot{m}_t$	mass flow rate of the air–fuel mixture
m_a/m_f	air–fuel ratio
N	engine speed
r_c	compression ratio
T	temperature
P_{out}	power output of the cycle
V	volume of the gas in the cylinder

Greek symbols

η_c	compression efficiency
η_{com}	combustion efficiency
η_e	expansion efficiency
η_{th}	thermal efficiency of the cycle
η_v	volumetric efficiency
γ	specific heat ratio
ρ_{air}	inlet air density
ϕ	equivalence ratio

Subscripts

1, 2, 3, 4	state points
s	stoichiometric condition

1. Introduction

A study of gas cycles as the models of internal combustion engines is useful for illustrating some of the important parameters influencing engine performance. In the last two decades, by using finite time thermodynamics theory, many optimization studies based on various performance criteria have been carried out for endoreversible and irreversible heat engine models [Ge et al., 2008; Ebrahimi, 2009a]. Lior and Rudy (1988) conducted an availability analysis of an ideal Otto cycle with instantaneous burning of the fuel. Orlov and Berry (1993) deduced the power and efficiency upper limits for internal-combustion engines. They derived the maximum work or power and the corresponding efficiency bounds. Angulo-Brown et al. (1996) examined the performance of an irreversible air-standard Otto cycle by taking into account the finite-time evolution of the cycle's compression and power strokes and gathering the global losses in a dissipation term represented by a friction force linear with the piston mean velocity. Chen et al. (1998) derived the relations between the net power and the efficiency of the Otto-cycle with heat-transfer loss. Fischer and Hoffman (2004) concluded that a quantitative simulation of an Otto-engine's behavior can be accurately achieved by a simple Novikov model with heat leaks. Ozsoysal (2006) gave the valid ranges of the heat transfer loss parameters of the Otto and diesel cycles with consideration of the heat loss as a percentage of the fuel's energy. Rocha-Matinez et al. (2006) presented a simplified irreversible Otto-cycle model with fluctuations in the combustion heat. Parlak and Sahin (2006) defined the internal irreversibility by using entropy production and analyzed the effect of the internal irreversibility on the performance of the irreversible reciprocating heat engine cycle. Abu-Nada et al. (2007) studied on thermodynamic analysis of spark ignition engine. They implemented a theoretical model of Otto cycle, with a working fluid consisting of various gas mixtures. Ge et al., (2008) analyzed the performance of an air standard Otto cycle. In the irreversible cycle model, the non-linear relation between the specific heat of the working fluid and its temperature, the friction loss computed according to the mean velocity of the piston, the internal irreversibility described by using the compression and expansion efficiencies, and the heat transfer loss are considered. Ebrahimi (2010a, 2010b, 2010c) determined the characteristics of power and efficiency for Otto cycle with engine speed, equivalence ratio and combustion efficiency.

As can be seen in the relevant literature, the investigation of the effect of volumetric efficiency on performance of Otto cycle does not appear to have been published. Therefore, the objective of this study is to examine the effect of the volumetric efficiency on performance of air standard Otto cycle.

2. Cycle model

An air-standard Otto cycle model is shown in figure 1. Process $1 \to 2s$ is a reversible adiabatic compression, while process $1 \to 2$ is an irreversible adiabatic process that takes into account the internal irreversibility in the real compression process. The heat addition is an isochoric process $2 \to 3$. Process $3 \to 4s$ is a reversible adiabatic expansion, while $3 \to 4$ is an irreversible adiabatic process that takes into account the internal irreversibility in the real expansion process. The heat rejection is an isochoric process $4 \to 1$. From the constant volume process $2 \to 3$, the heat added to the working fluid is

$$Q_{in} = \dot{m}_t c_v (T_3 - T_2) \quad (1)$$

The total energy of the fuel per second input into the engine can be given by: (Heywood, 1988)

$$Q_{fuel} = \eta_{com} \dot{m}_f Q_{LHV} \quad (2)$$

The heat loss through the cylinder wall is given in the following linear expression (Chen et al., 2008, Ebrahimi, 2009b)

$$Q_{ht} = \dot{m}_t B (T_2 + T_3) \quad (3)$$

Since the total energy of the delivered fuel Q_{fuel} is assumed to be the sum of the heat added to the working fluid Q_{in} and the heat leakage Q_{ht},

$$Q_{in} = Q_{fuel} - Q_{ht} = \eta_{com} \dot{m}_f Q_{LHV} - \dot{m}_t B (T_2 + T_3) \quad (4)$$

The intake system -the air filter, carburetor, and throttle plate, intake manifold, intake port, intake valve- restricts the amount of air which an engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the volumetric efficiency. It is defined as the volume flow rate of air into the intake system divided by the rate at which volume is displaced by the piston:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{air} V_d N} \quad (5)$$

The relations between η_v and $\dot{m}_f$, between η_v and $\dot{m}_t$ are defined as (Heywood, 1988):

$$\dot{m}_f = \frac{\eta_v \rho_{air} V_d N \phi}{2 (m_a/m_f)_s} \quad (6)$$

and

$$\dot{m}_t = \frac{\eta_v \rho_{air} V_d N}{2} \left(1 + \frac{\phi}{(m_a/m_f)_s} \right) \quad (7)$$

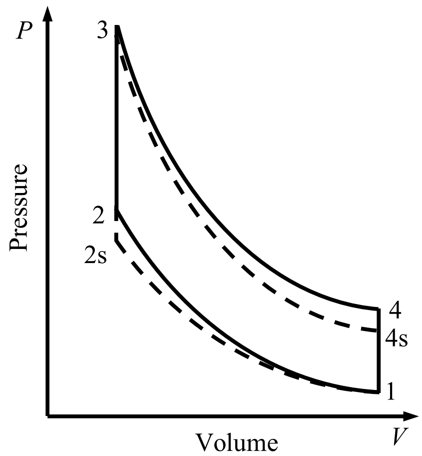

Figure 1. $P-V$ diagram for the air standard Otto cycle

The compression ratio, r_c, is defined as:

$$r_c = V_1/V_2 \quad (8)$$

For the processes $1 \to 2s$ and $3 \to 4s$, one has

$$T_{2s} = T_1 r_c^{\gamma-1} \quad (9)$$

$$T_{4s} = T_3 r_c^{1-\gamma} \quad (10)$$

For the two reversible adiabatic processes $1 \to 2s$ and $3 \to 4s$, the compression and expansion efficiencies can be defined as (Chen, 2004, Ebrahimi, 2010c):

$$\eta_c = (T_{2s} - T_1)/(T_2 - T_1) \quad (11)$$

$$\eta_e = (T_4 - T_3)/(T_{4s} - T_3) \quad (12)$$

Substituting equation (11) into equation (9) yields:

$$T_2 = \frac{T_1 \left(r_c^{\gamma-1} + \eta_c - 1 \right)}{\eta_c} \quad (13)$$

Combining equations(1) , (4), (6) and (13) gives:

$$T_3 = \frac{\eta_c \eta_{com} Q_{LHV} \phi (m_a/m_f)_s \left[1+\phi/(m_a/m_f)_s\right] + T_1\left(r_c^{\gamma-1}+\eta_c-1\right)\left(R_{air}/(\gamma-1)-B\right)}{\eta_c\left(R_{air}/(\gamma-1)+B\right)} \quad (14)$$

Substituting equations (10) and (14) into equation (12) yields:

$$T_4 = \left[\frac{\eta_c \eta_{com} Q_{LHV} \phi (m_a/m_f)_s \left[1+\phi/(m_a/m_f)_s\right] + T_1\left(r_c^{\gamma-1}+\eta_c-1\right)\left(R_{air}/(\gamma-1)-B\right)}{\eta_c\left(R_{air}/(\gamma-1)+B\right)}\right] \times \left(\eta_e r_c^{1-\gamma}-\eta_e+1\right) \quad (15)$$

The data of total motored friction mean effective pressure for several four stroke cycle, four cylinder spark ignition engines between 845 and 2000 cm^3 displacement, at wide open throttle, as a function of engine speed (Abd Alla, 2002) are well correlated by an equation of the form:

$$fmep = 97 + 0.9N + 0.18N^2 \quad (16)$$

Therefore, the lost power due to friction is

$$p_{fri} = \frac{fmep V_d N}{2} = V_d N\left(45.5+0.45N+0.09N^2\right) \quad (17)$$

Assuming an ideal, non-reacting gas with specific heat, the net actual power output of the Otto cycle engine can be written as:

$$P_{otto} = Q_{in} - Q_{out} - p_{fri} = \frac{\eta_v \rho_{air} V_d N R_{air}}{2(\gamma-1)}\left(1+\frac{\phi}{(m_a/m_f)_s}\right) \times \left[T_1\left(1-\frac{r_c^{\gamma-1}+\eta_c-1}{\eta_c}\right) - \left[\frac{\eta_c \eta_{com} Q_{LHV} \phi (m_a/m_f)_s \left[1+\phi/(m_a/m_f)_s\right] + T_1\left(r_c^{\gamma-1}+\eta_c-1\right)\left(R_{air}/(\gamma-1)-B\right)}{\eta_c\left(R_{air}/(\gamma-1)+B\right)}\right]\left(\eta_e r_c^{1-\gamma}+\eta_e\right)\right] - V_d N\left(45.5+0.45N+0.09N^2\right) \quad (18)$$

The thermal efficiency of the Otto cycle engine is expressed by

$$\eta_{th} = P_{otto}/Q_{in} \quad (19)$$

where Q_{in} is:

$$Q_{in} = \dot{m}_t c_v \left(T_3 - T_2\right) = \frac{\eta_v \rho_{air} V_d N R_{air}}{2\eta_c(\gamma-1)}\left(1+\frac{\phi}{(m_a/m_f)_s}\right) \times \left(\frac{\eta_c \eta_{com} Q_{LHV} \phi (m_a/m_f)_s \left[1+\phi/(m_a/m_f)_s\right] + T_1\left(r_c^{\gamma-1}+\eta_c-1\right)\left(R_{air}/(\gamma-1)-B\right)}{R_{air}/(\gamma-1)+B} - T_1\left(r_c^{\gamma-1}+\eta_c-1\right)\right) \quad (20)$$

Notice that both power and efficiency are convex functions of the compression ratio.

3. Numerical examples and discussions

As it can be clearly seen from Eqs. (18) and (19), the thermal efficiency and the power output of the Otto cycle are dependent on the volumetric efficiency. In order to illustrate the effect of this parameter, the relations between the power output and the compression ratio, between the thermal efficiency and the compression ratio, and the optimal relation between power output and the efficiency of the cycles presented in figures 2–4. The values of the constants and the parameters used in this example are summarized in Table 1.

Table 1. Constants and parameters used in the numerical example [Heywood, 1988; Ebrahimi, 2009b; Chen et al., 2008; Ge et al., 2008]

bore×stroke	76.7×78 mm
constant related to heat transfer	0.71 $kJ\,kg^{-1}K^{-1}$
lower heating value of the fuel	44000 $kJ\,kg^{-1}$
air–fuel ratio at stoichiometric conditions	0.0685
engine speed	$3000\,rpm$
compression ratio	$1 \to 70$
intake temperature	$300\,K$
compression efficiency	0.97
expansion efficiency	0.97
combustion efficiency	100%
volumetric efficiency	$60 \to 100\%$
specific heat ratio	1.4
equivalence ratio	1

Figures 2-4 show the effect of the volumetric efficiency on the cycle performance with heat resistance, internal irreversibility and friction losses. From these figures, it can be found that the volumetric efficiency plays important roles on the performance of the Otto engine. It is clearly seen that the effects of volumetric efficiency on the performance of the cycle is related to compression ratio. They reflect the performance characteristics of a real irreversible Otto cycle engine. The power output versus compression ratio characteristic and the thermal efficiency versus compression ratio characteristic are approximately parabolic like curves. It should be noted that the heat added and the heat rejected by the working fluid increase as the volumetric efficiency increases. It can also be seen that the curves of power output versus thermal efficiency are loop shaped as is common to almost all real heat engines (Ebrahimi, 2009b).

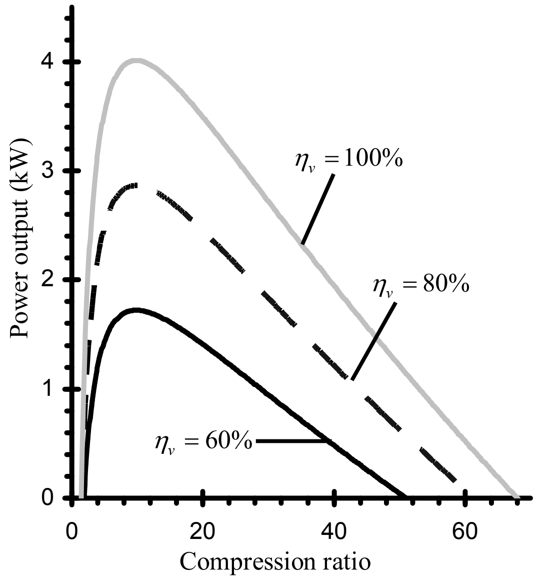

Figure 2. Effect of volumetric efficiency on the variation of the power output with compression ratio

Referring to Figs. 2 and 3, it can be concluded that, throughout the compression ratio range, the power output and thermal efficiency increase with the increasing volumetric efficiency. This can be attributed to the fact that the difference between heat added and heat rejected increase with the increasing volumetric efficiency. This is consistent with the practical working volumetric efficiency of spark ignition engines (Heywood, 1988). From these figures, it can be resulted that the maximum power output and the maximum thermal efficiency increase about 69.6% and 31% when volumetric efficiency increases 40%. The results also revealed that the optimal compression ratio corresponding to maximum thermal efficiency point and the working range of the cycle increase as the volumetric efficiency increases. In other words, the optimal compression ratio corresponding to maximum thermal efficiency point and the working range of the cycle increase about 25% and 25.1% when volumetric efficiency increases 40%. While the optimal compression ratio corresponding to maximum power output point remains constant with the increasing volumetric efficiency.

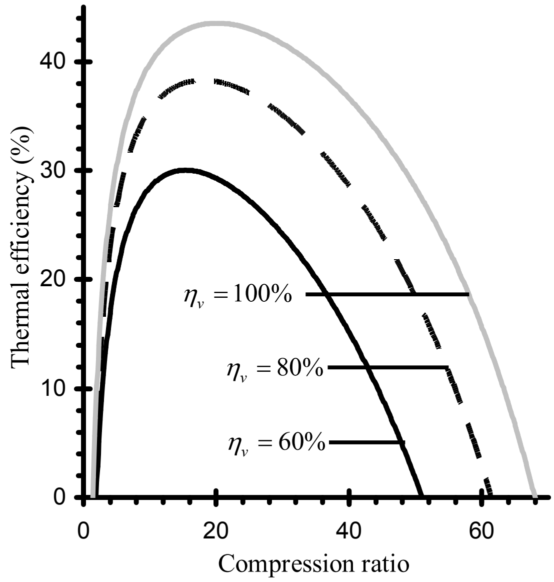

Figure 3. Effect of volumetric efficiency on the variation of the thermal efficiency with compression ratio

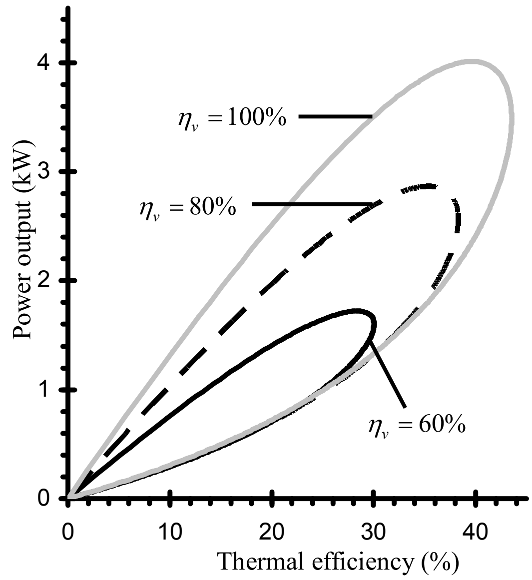

Figure 4. Effect of volumetric efficiency on the variation of the power output with thermal efficiency

Referring to Fig. 4, it can be seen that the power output at maximum thermal efficiency and the thermal efficiency at maximum power output improve when the volumetric efficiency increased. In other words, the power output at maximum thermal efficiency and the thermal efficiency at maximum power output increase about 54.3% and 28.7% when volumetric efficiency increases 40%.

4-Conclusion

In this paper, the effects of volumetric efficiency on the performance of an Otto cycle during the finite time are investigated. The general conclusions drawn from the results of this work are as follows:

- Throughout the compression ratio range, the power output and thermal efficiency increase with the increasing volumetric efficiency.
- The optimal compression ratio corresponding to maximum thermal efficiency point and the working range of the cycle increase as the volumetric efficiency increases. While the optimal compression ratio corresponding to maximum power output point remains constant with the increasing volumetric efficiency.
- The power output at maximum thermal efficiency and the thermal efficiency at maximum power output improve when the volumetric efficiency increased.

The results of this investigation are of importance when considering the designs of actual Otto engines.

Correspondence to:
Rahim Ebrahimi
Department of Agriculture Machine Mechanics
Shahrekord University, PO Box 115, Shahrekord, Iran
Tel/Fax: 0098-381-4424412
Email: Rahim.Ebrahimi@gmail.com

References

1. Abu-Nada E, Al-Hinti I, Akash B, Al-Sarkhi A. Thermodynamic analysis of spark-ignition engine using a gas mixture model for the working fluid. International Journal of Energy Research 2007;31:1031–1046.
2. Abd Alla GH. Computer simulation of a four stroke spark ignition engine. Energy Conversion & Management 2002;43:1043-1061.
3. Angulo-Brown F, Rocha-Martinez JA, Navarrete-Gonzalez TD. A non-endoreversible Otto cycle model: improving power output and efficiency. Journal of Physics D: Applied Physics, 1996;29:80–83.
4. Chen L, Wu C, Sun F, Wu C. Heat transfer effects on the net work output and efficiency characteristics for an air standard Otto-cycle. Energy Convers Manage 1998;39(7):643–8.
5. Chen L, Ge Y, Sun F. Unified thermodynamic description and optimization for a class of irreversible reciprocating heat engine cycles. Proc IMechE Part D: J Automobile Engineering 2008;222:1489-1500.
6. Chen L, Sun F, Wu C. The optimal performance of an irreversible Dual-cycle. Applied Energy 2004;79(1):3–14.
7. Ebrahimi R. Effects of cut-off ratio on performance of an irreversible Dual cycle. Journal of American Science 2009a;5(3):83-90.
8. Ebrahimi R. Thermodynamic simulation of performance of an endoreversible Dual cycle with variable specific heat ratio of working fluid. Journal of American Science 2009b;5(5):175-180.
9. Ebrahimi R. Effects of gasoline-air equivalence ratio on performance of an Otto engine, Journal of American Science 2010a;6(2):131-135.
10. Ebrahimi R. Theoretical study of combustion efficiency in an Otto engine, Journal of American Science 2010b;6(2):113-116.
11. Ebrahimi R. Engine speed effects on the characteristic performance of Otto engines, Journal of American Science 2010c;6(1):123-128.
12. Fischer A, Hoffman KH. Can a quantitative simulation of an Otto engine be accurately rendered by a simple Novikov model with a heat leak? J Non-Equil Thermody 2004;29(1):9–28.
13. Ge Y, Chen L, Sun F. Finite time thermodynamic modeling and analysis of an irreversible Otto cycle. Applied Energy 2008;85(7):618-624.
14. Heywood JB. Internal combustion engine fundamentals. New York: McGraw-Hill; 1988.
15. Lior N, Rudy GL. Second-law analysis of an ideal Otto cycle. Energy Convers Manage 1988;28:327–34.
16. Orlov VN, Berry RS. Power and efficiency limits for internal-combustion engines via methods of finite-time thermodynamics. J Appl Phys 1993;74(10):4317–22.
17. Ozsoysal OA. Heat loss as percentage of fuel's energy in air standard Otto and diesel cycles. Energy Conv Manage 2006;47(7–8):1051–1062.
18. Parlak A, Sahin B. Performance optimization of reciprocating heat engine cycles with internal irreversibility. J. Energy Inst., 2006;79(4):241–245.
19. Rocha-Matinez JA, Navarrete-Gonzalez TD, Pavia-Miller CG, et al.. A simplified irreversible Otto-engine model with fluctuations in the combustion heat. Int J Ambient Energy 2006;27(4):181–92.

10/18/2009