Performance of an irreversible Diesel cycle under variable stroke length and compression ratio

Rahim Ebrahimi *

Department of Agriculture Machine Mechanics, Shahrekord University, P.O. Box 115, Shahrekord, Iran
Rahim.Ebrahimi@gmail.com

Abstract: Finite-time thermodynamic analysis of an air-standard internal-combustion Diesel cycle is performed in this paper. The relations between the power output and the compression ratio, between the power output and the thermal efficiency are derived by detailed. The maximum power output and the corresponding efficiency limit of the cycle with considerations of heat transfer and friction like term losses are also found. Detailed numerical examples are given. The results shows that if compression ratio is less than certain value, the power output increases with increasing stroke length, while if compression ratio exceeds certain value, the power output first increases and then starts to decrease with increasing stroke length. With further increase in compression ratio, the increase of stroke length results in decreasing the power output. The results obtained in the present study are of importance to provide good guidance for performance evaluation and improvement of practical Diesel engines. [Journal of American Science 2009;5(7):58-64]. (ISSN: 1545-1003).

Keywords: Diesel cycle; power output; thermal efficiency; irreversible; friction; stroke length

1. Introduction

Significant achievements have ensued since finite-time thermodynamics was developed in order to analyze and optimize the performances of real heat-engines (Mozurkewich and Berry, 1982; Andersen and Band, 1984; Sieniutycz and Shiner, 1994; Chen et al., 1998; Aragon-Gonzalez et al., 2000; Chen et al., 2004]. Hoffman et al. (1985) used mathematical techniques, developed for optimal-control theory, to reveal the optimal motions of the pistons in Diesel cycle engine. Klein (1991) studied the effect of heat transfer on the performance of the Otto and Diesel cycles. Blank and Wu (1993) examined the effect of combustion on the work or power optimised Otto, Diesel and Dual cycles. They derived the maximum work or power and the corresponding efficiency bounds. Orlov et al. (1993) obtained the power and efficiency limits for internal combustion engines. Chen et al. (1996) derived the relations between the power output and the thermal efficiency for the Diesel cycle with the consideration of the heat-transfer losses. Bhattacharyya (2000) proposed a simplified irreversible model for an air-standard Diesel cycle by using the finite time thermodynamic approach. In his study, global thermal and friction losses are lumped into an equivalent friction term, which is linear in the piston velocity. Chen et al. (2002) modelled the diesel cycle with friction-like term loss during a finite time and studied the effect of friction-like term loss on the cycle performance. Burzler (2002) determined the optimum piston trajectory that yielded maximum power output and efficiency of the diesel cycle with different losses when the cycle period, fuel intake per cycle, fuel–air mixture composition, and compression ratio are taken as constants. Rocha-Martinez et al. (2002) investigated the effect of variable specific-heats on the Otto and Diesel cycle performance. Chen et al. (2002) modeled the behaviors of Diesel cycle, with friction losses, over a finite period. Qin et al. (2003) and Ge et al. (2005a) derived the performance characteristics of the diesel cycle with heat transfer and friction like term losses when the maximum temperature of the cycle was not fixed. Wang and Hou (2005) studied the performance analysis and comparison of an Atkinson cycle coupled to variable temperature heat reservoirs under maximum power and maximum power density conditions, assuming a constant specific heat, too. Their results showed an engine design based on maximum power density is better than that based on maximum power conditions, from the view points of engine size and thermal efficiency. Ge et al. (2005b) considered the effect of variable specific heats on the cycle process and studied the performance characteristics of endoreversible and irreversible Otto cycles when variable specific heats of working fluid are linear functions of the temperature. 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. Zhao and Chen (2006) performed analysis and parametric optimum criteria of an irreversible Atkinson heat engine using finite time thermodynamic. Al-Sarkhi et al. (2006) studied the effects of variable specific heats of the working fluid on the performances of the Diesel and Miller cycles. Ge et al. (2007) studied the effects of variable specific heats of the working fluid on the performances of the Diesel cycle. Ge et al. (2008a, 2008b) analyzed the performance of an air standard Otto and Diesel cycles. 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 (2009) studied the effects of the temperature dependent specific heat ratio of the working fluid on the performance of the diesel cycle.

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

2. Thermodynamic analysis

The Diesel cycle shown in Fig. 1 approximates the compression stroke up to ignition with the adiabatic reversible (isentropic) process $1 \to 2$; it assumes that the combustion process is represented by the reversible constant pressure process $2 \to 3$; it approximates the power stroke with the isentropic expansion process $3 \to 4$; and it assumes that the heat-removing process is the reversible constant volume process $5 \to 1$.

In a real cycle, the specific heat ratio is generally modeled as the first and second order equation of mean charge temperature (Gatowski et al., 1984; Brunt and Emtage, 1997; Ebrahimi, 2006). Thus, it can be supposed that the specific heat ratio of the working fluid is function of temperature alone and has the second order equation forms:

$$\gamma = aT^2 + bT + c \tag{1}$$

Where γ is the specific heat ratio and T is the absolute temperature. a, b and c are constants.

The heat added per second in the isobaric ($2 \to 3$) heat addition process may be written as

$$Q_{in} = M_{nl}\int_{T_2}^{T_3} c_p dT = M_{nl}\int_{T_2}^{T_3}\left(\frac{R\left(aT^2+bT+c\right)}{aT^2+bT+c-1}\right)dT = M_{nl}R\left(T_3 - T_2 + \frac{2}{D}\left[arctg\left(\frac{2aT_3+b}{D}\right) - arctg\left(\frac{2aT_2+b}{D}\right)\right]\right) \tag{2}$$

Where D is defined as $D = \sqrt{4ac - 4a - b^2}$. R and c_p are gas constant and specific heat at constant pressure for the working fluid, respectively. M is the molar number of the working fluid per second.

The heat rejected per second in the isochoric heat rejection process ($4 \to 1$) may be written as

$$Q_{out} = M_{nl}\int_{T_1}^{T_4} c_v dT = M_{nl}\int_{T_1}^{T_4}\left(\frac{R}{aT^2+bT+c-1}\right)dT = \frac{2M_{nl}R}{D}\left[arctg\left(\frac{2aT_4+b}{D}\right) - arctg\left(\frac{2aT_1+b}{D}\right)\right] \tag{3}$$

where c_v is the molar specific heat at constant volume for the working fluid.

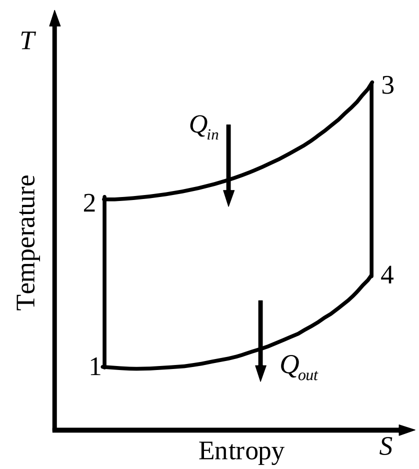

Figure 1. Temperature-entropy $(T - S)$ diagram for the air standard Diesel cycle

Since c_p and c_v are dependent on temperature, the adiabatic exponent $\gamma = c_p / c_v$ will vary with temperature as well. Therefore, the equation often used in a reversible adiabatic process with constant γ cannot be used in a reversible adiabatic process with variable γ. However, according to Refs. (Ebrahimi, 2009a; Chen et al., 2008), the equation for a reversible adiabatic process with variable can be written as follows:

$$TV^{\gamma-1} = \left(T + \mathrm{d}T\right)\left(V + \mathrm{d}V\right)^{\gamma-1} \tag{4}$$

From Eq. (4), one gets

$$\frac{1}{2(c-1)}\ln\left(\frac{aT_j^2+bT_j+c-1}{aT_i^2+bT_i+c-1}\right)+\frac{b}{(c-1)D}\left[arctg\left(\frac{2aT_j+b}{D}\right)-arctg\left(\frac{2aT_i+b}{D}\right)\right]-\frac{\ln\left(T_j/T_i\right)}{c-1}=\ln\left(V_j/V_i\right) \quad (5)$$

The compression, r_c, and cut-off, β, ratios are defined as

$$r_c=V_1/V_2 \quad \text{and} \quad \beta=V_4/V_3=T_4/T_3 \quad (6)$$

Therefore, the equations for processes ($1\to 2$) and ($3\to 4$) are shown, respectively, by the following:

$$\frac{1}{2(c-1)}\ln\left(\frac{aT_2^2+bT_2+c-1}{aT_1^2+bT_1+c-1}\right)+\frac{b}{(c-1)D}\times\left[arctg\left(\frac{2aT_2+b}{D}\right)-arctg\left(\frac{2aT_1+b}{D}\right)\right]-\frac{\ln\left(T_2/T_1\right)}{c-1}=\ln\left(1/r_c\right) \quad (7)$$

and

$$\frac{1}{2(c-1)}\ln\left(\frac{aT_4^2+bT_4+c-1}{aT_5^2+bT_3+c-1}\right)+\frac{b}{(c-1)D}\times\left[arctg\left(\frac{2aT_4+b}{D}\right)-arctg\left(\frac{2aT_3+b}{D}\right)\right]-\frac{\ln\left(T_4/T_3\right)}{c-1}=\ln\left(\frac{r_c}{\beta}\right) \quad (8)$$

For an ideal Diesel cycle model, there are no losses. However, for a real internal combustion engine cycle, heat transfer irreversibility between the working fluid and the cylinder wall is not negligible. One can assume that the heat loss through the cylinder wall is proportional to the average temperature of both the working fluid and the cylinder wall and that the wall temperature is constant at T_0. If the released heat by combustion for one molar working fluid is A_1, and the heat leakage coefficient of the cylinder wall is B_1, one has the heat added to the working fluid per second by combustion as the following linear relation (Chen et al. 2006).

$$Q_{in}=M_{nl}\left(A_1-B_1\frac{T_2+T_3}{2}-T_0\right)=M_{nl}\left(A_1+B_1T_0-\frac{T_2+T_3}{2}\right)=M_{nl}\left[A+B\left(T_2+T_3\right)\right] \quad (9)$$

where $A=A_1+B_1T_0$ and $B=B_1/2$ are two constants related to combustion and heat transfer.

Taking into account the friction loss of the piston, as deduced by Al-Sarkhi et al. (2006) for the Diesel cycle, and a dissipation term represented by a friction force which in a linear function of the velocity gives

$$f_\mu=-\mu v=-\mu\frac{dx}{dt} \quad (10)$$

where μ is the coefficient of friction, which takes into account the global losses, x is the piston's displacement and v is the piston's velocity. Therefore, the lost power due to friction is

$$P_\mu=\frac{dW_\mu}{dt}=-\mu\left(\frac{dx}{dt}\right)^2=-\mu v^2 \quad (11)$$

Running at N cycles per second, the mean velocity of the piston is

$$\bar{v}=4LN \quad (12)$$

where L is the total distance the piston travels per cycle.

Thus, the power output of the Diesel cycle engine can be written as

$$P_{out}=Q_{in}-Q_{out}-P_\mu=\frac{2M_{nl}R}{D}\left[\frac{D}{2}\left(T_3-T_2\right)+arctg\left(\frac{2aT_1+b}{D}\right)+arctg\left(\frac{2aT_3+b}{D}\right)-arctg\left(\frac{2aT_2+b}{D}\right)-arctg\left(\frac{2aT_4+b}{D}\right)\right]-16\mu\left(LN\right)^2 \quad (13)$$

The efficiency of the Diesel cycle engine is expressed by

$$\eta_{ot}=\frac{Q_{in}-Q_{out}-P_\mu}{Q_{in}}=\frac{\frac{2}{D}\left[arctg\left(\frac{2aT_1+b}{D}\right)+arctg\left(\frac{2aT_3+b}{D}\right)-arctg\left(\frac{2aT_2+b}{D}\right)-arctg\left(\frac{2aT_4+b}{D}\right)\right]+\left(T_3-T_2\right)-16\mu\left(LN\right)^2}{T_3-T_2+\frac{2}{D}\left[arctg\left(\frac{2aT_3+b}{D}\right)-arctg\left(\frac{2aT_2+b}{D}\right)\right]} \quad (14)$$

When r_c and T_1 are given, T_2 can be obtained from equation (7), then, substituting equation (2) into equation (9) yields T_3, and the last, T_4 can be worked out by equation (8). Substituting T_1, T_2, T_3 and T_4 into equations (13) and (14), respectively, the power output and thermal efficiency of the Diesel cycle engine can be obtained. Therefore, the relations between the power output, the thermal efficiency and the compression ratio can be derived.

3. Results and discussion

The following constants and parameters have been used in this exercise: $T_1 = 300\,K$, $A = 60000\,J.mol^{-1}$, $\beta = 1.5$, $B = 28\,J.mol^{-1}K^{-1}$, $a = 1.6928\times10^{-8}\,K^{-1}$, $b = -9.7617\times10^{-5}\,K^{-1}$, $c = 1.4235$, $r_c = 1.5 - 56.6$, $L = 40 - 120\ mm$, $M_{nl} = 2.24\times10^{-4}\,NL\,kmols^{-1}$, $\mu = 12.9\,Nsm^{-1}$ and $N = 4500\,rpm$ (Chen et al., 2006; Ghatak and Chakraborty, 2007; Ge et al., 2007; Ebrahimi, 2009b). Using the above constants and range of parameters, the power output versus compression ratio characteristic and the power output versus efficiency characteristic with varying the stroke length can be plotted. Numerical examples are shown as follows.

The variations in the temperatures T_2, T_3 and T_4 with the compression ratio are shown in figure 2. It is found that T_2 and T_3 increase with the increase of compression ratio, and T_4 decreases with the increase of compression ratio.

Figures 3-4 show the effects of the variable stroke length on the cycle performance with heat resistance and irreversible friction losses. From these figures, it can be found that the stroke length plays important roles on the power output. It is clearly seen that the effects of stroke length on the power output is related to compression ratio. They reflect the performance characteristics of a real irreversible Diesel cycle engine. It should be noted that the heat added and the heat rejected by the working fluid increase with increasing stroke length. (see equations. (2) and (3)).

Figure 3 indicates the effects of the stroke length on the power output of the cycle for different values of the compression ratio. It can be seen that the power output versus compression ratio characteristic is approximately parabolic like curves. In other word, the power output increases with increasing compression ratio, reach their maximum values and then decreases with further increase in compression ratio. The maximum power output increases with increasing stroke length up to about 80 mm where it reaches its peak value then starts to decline as the stroke length increases. This is consistent with the practical working stroke length of engines, which are between 70 and 90 mm in general. The optimal compression ratio corresponding to maximum power output point remains constant with increase of engine speed. The working range of the cycle decreases as the stroke length increases. The results shows that if compression ratio is less than certain value, the power output increases with increasing stroke length, while if compression ratio exceeds certain value, the power output first increases

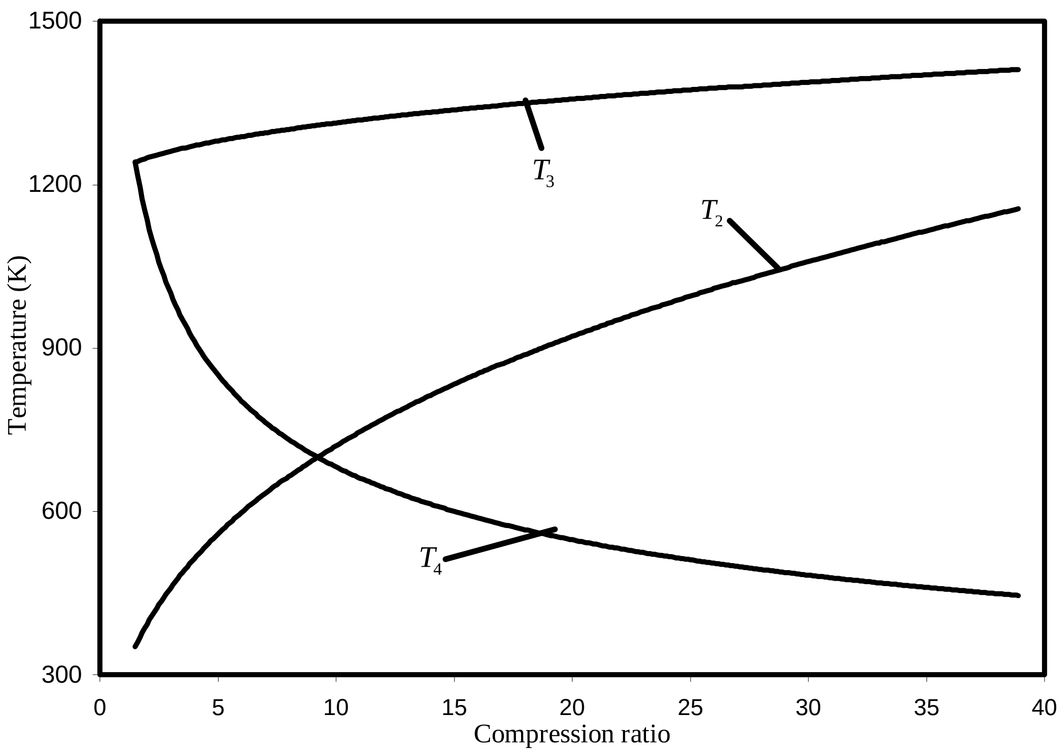

Figure 2. The temperature versus compression ratio ($L = 70\,mm$)

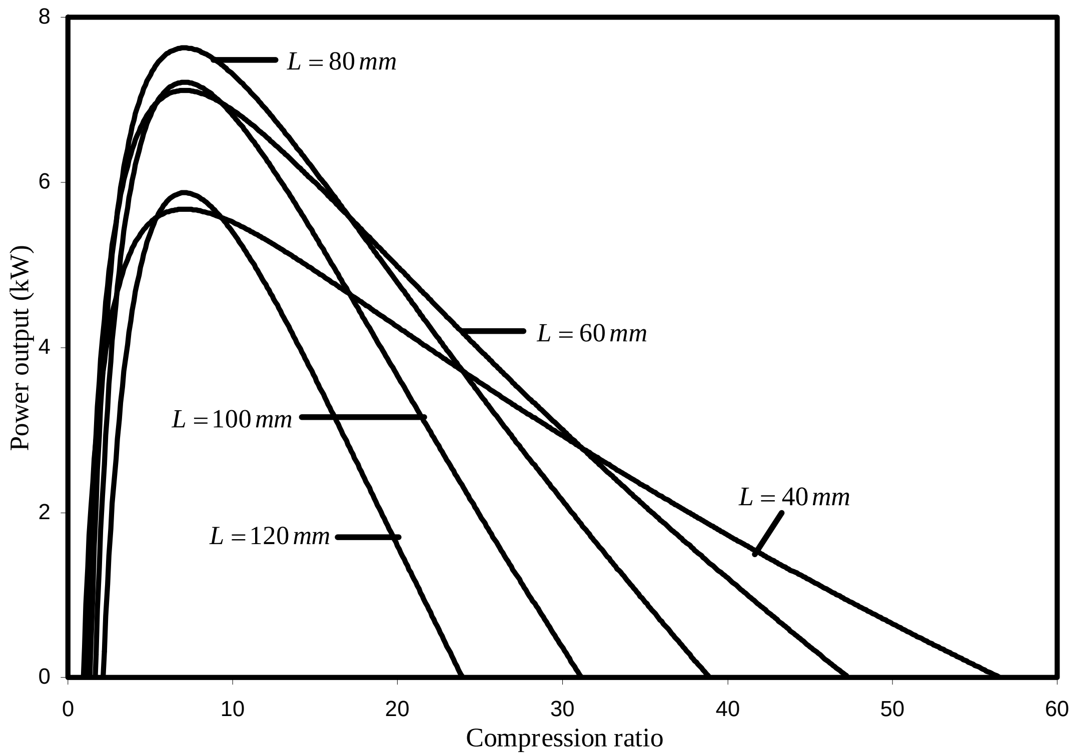

Figure 3. Effect of L on the $P_{out} - r_c$ characteristic

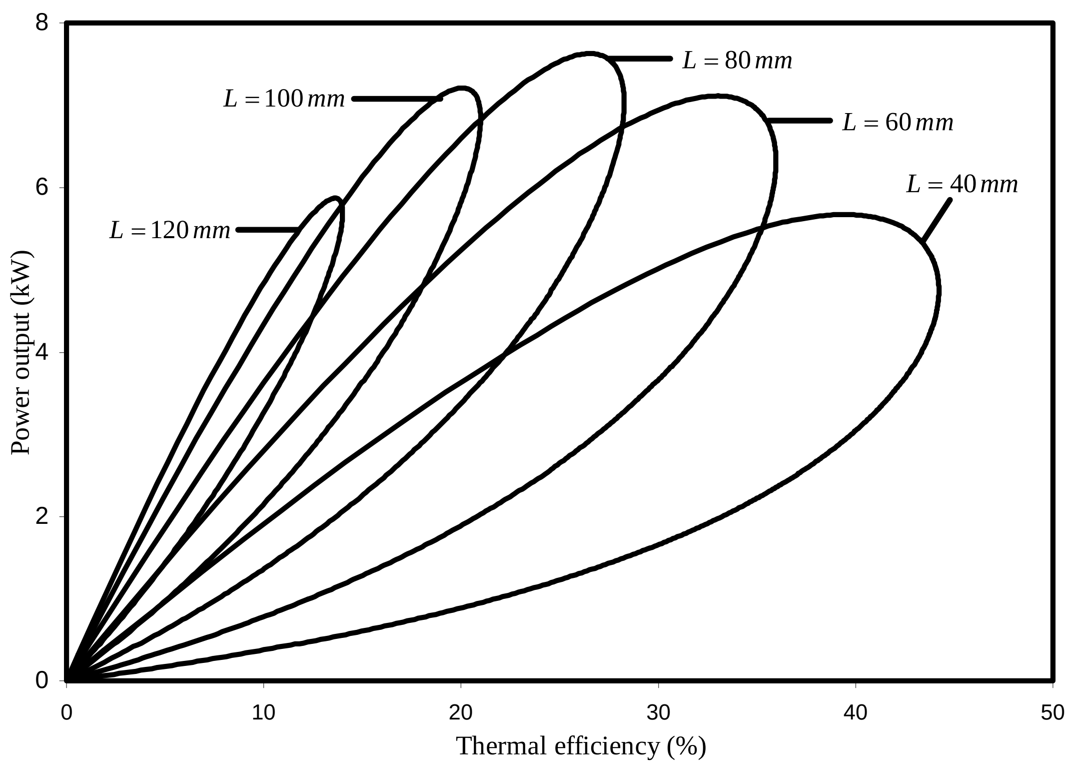

Figure 5. Effect of L on the $P_{out} - \eta_{at}$ characteristic

and then starts to decrease with increasing stroke length. With further increase in compression ratio, the increase of stroke length results in decreasing the power output. Numerical calculation shows that for any same compression ratio, the smallest power output is for $L = 120\,mm$ when $r_c \leq 5.4$ or $r_c > 9.3$ and is for $L = 40\,mm$ when $5.4 < r_c \leq 9.3$ and also the largest power output is for $L = 40\,mm$ when $r_c \leq 1.65$ or $31.2 < r_c$, is for $L = 60\,mm$ when $1.65 < r_c < 3$ or $17.1 \leq r_c \leq 31.2$ and is for $L = 70\,mm$ when $3 \leq r_c < 17.1$.

The influence of the stroke length on the power output versus thermal efficiency is displayed in figure 4. As can be seen from this figure, the power output versus thermal efficiency is loop shaped one. It can be seen that the power output at maximum thermal efficiency improves with increasing stroke length from 40 to around $70\,mm$. With further increase in stroke length, the power output at maximum thermal efficiency decreases. The thermal efficiency at maximum power decreases with increase of stroke length from 40 to $120\,mm$.

According to above analysis, it can be found that the effects of the stroke length on the cycle performance are obvious, and they should be considered in practice cycle analysis in order to make the cycle model be more close to practice.

4. Conclusion

An air-standard Diesel-cycle model, assuming a temperature-dependent specific heat ratio of the working fluid, and heat resistance and frictional irreversible losses, has been investigated numerically. The performance characteristics of the cycle with varying stroke lengths and compression ratios were obtained by numerical examples. The results show that if compression ratio is less than certain value, the power output increases with increasing stroke length, while if compression ratio exceeds certain value, the power output first increases and then starts to decrease with increasing stroke length. With further increase in compression ratio, the increase of stroke length results in decreasing the power output. The results also show that the maximum power output increase with increasing stroke length. With further increase in stroke length, the increase of stroke length results in decreasing the maximum power output. The analysis helps us to understand the strong effect of stroke length on the performance of the Diesel cycle. Therefore, the results are of great significance to provide good guidance for the performance evaluation and improvement of real Diesel engines.

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

Reference

[1] Aizenbud BM, Band YB. Optimization of model internal combustion engine. Journal of Applied Physics 1982;63:1277-1282.

[2] Al-Sarkhi A, Jaber JO, Abu-Qudais M, Probert SD. Effects of friction and temperature dependent specific-heat of the working fluid on the performance of a diesel engine. Applied Energy 2006;83(2):153–65.

[3] Aragon-Gonzalez G, Ganales-Palma A, Leon-Galicia A. Maximum irreversible work and efficiency in power cycles. Journal of Physics D: Applied Physics. 2000;33(11):1403–09.

[4] Burzler MJ. Performance optimal for endoreversible systems. PhD Thesis, University of Chemnitz, Germany, 2002.

[5] Bhattacharyya S. Optimising an irreversible Diesel cycle-Fine tuning of compression ratio and cut-off ratio. Energ Convers Manage 2000;41:847–54.

[6] Blank, DA, Wu C. The effects of combustion on a power optimised endoreversible Diesel cycle. Energy Conver & Manage 1993;34:493–8.

[7] Brunt MFJ, Rai H, Emtage AL. The calculation of heat release energy from engine cylinder pressure data. SAE Paper No. 981052. 1998.

[8] 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.

[9] Chen L, Ge Y, Sun F, Wu C. Effects of heat transfer, friction and variable specific heats of working fluid on performance of an irreversible dual cycle, Energy Conversion and Management 2006;47:3224–34

[10] Chen L, Sun F, Wu C. Optimal performance of an irreversible Dual cycle, Applied Energy 2004;79(1):3-14.

[11] Chen L, Lin J, Luo J, Sun F, Wu C. Friction effects on the characteristic performance of Diesel engines. Int J Energ Res 2002;26(10):965–71.

[12] Chen L, Wu C, Sun F, Cao S. Heat transfer effects on the net work output and efficiency characteristics for an airstandard

Otto cycle. Energy Conversion and Management 1998;39:643–48.

[13] Chen L, Zeng F, Sun F, Wu C. Heat-transfer effects on net work and/or power as functions of efficiency for air-standard diesel cycles. Energy 1996;21(12):1201-1205.

[14] Ebrahimi R. Experimental study on the auto ignition in HCCI engine, Ph.D. Thesis, Valenciennes et du Hainaut-Cambrésis, France, 2006 (In French).

[15] 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.

[16] Ebrahimi R. Effects of cut-off ratio on performance of an irreversible Dual cycle, Journal of American Science 2009a;5(3):83-90

[17] Ebrahimi R. Effects of variable specific heat ratio of working fluid on performance of an endoreversible Diesel cycle, J. Energy Inst. [accepted]

[18] Gatowski JA, Balles EN, Chun KM, Nelson F, Ekchian JA, Heywood FB. A heat release analysis of engine pressure data. SAE Paper No, 841359. 1984

[19] Ge Y, Chen L, Sun F. Finite time thermodynamic modeling and analysis of an irreversible Otto cycle. Applied Energy 2008a;85(7):618-624.

[20] Ge Y, Chen L, Sun F. Finite time thermodynamic modeling and analysis for an irreversible Diesel cycle. Proceedings IMechE, Part D: Journal of Automobile Engineering 2008b;222(D5):887-94.

[21] Ge Y, Chen L, Sun F, Wu C. Reciprocating heat-engine cycles. Applied Energy 2005a;81(4):397-408.

[22] Ge Y, Chen L, Sun, F, Wu C. The effects of variable specific-heats of the working fluid on the performance of an irreversible Ottocycle. Int J Exergy 2005b;2(3):274–83.

[23] Ge Y, Chen L, Sun F, Wu C. Performance of Diesel cycle with heat transfer, friction and variable specific heats of working fluid. J Energy Inst 2007;80(4):239–42.

[24] Ghatak A, Chakraborty S. Effect of external irreversibilities and variable thermal properties of working fluid on thermal performance of a Dual internal combustion engine cycle. Strojnicky Casopis (Journal Mechanical Energy) 2007;58:1–12.

[25] Hoffman KH, Watowich SJ, Berry RS. Optimal paths for thermodynamic systems: the ideal Diesel cycle. J Appl Phys 1985;58(6):2125–34.

[26] Klein SA An explanation for observed compression ratios in internal combustion engines, Trans. ASME J. Engrg. Gas Turbine Pow 1991;113(4):511–13.

[27] Mozurkewich M, Berry RS. Optimal paths for thermodynamic systems. The ideal Otto cycle. J Appl Phys 1982;63:34-42.

[28] Ozsoysal OA. Heat loss as percentage of fuel's energy in air standard Otto and diesel cycles. Energy Conversion Managmt 2006;47(7–8):1051–1062.

[29] 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.

[30] Qin X, Chen L, Sun F. The universal power and efficiency characteristics for irreversible reciprocating heat engine cycles. Eur J Physics 2003;24 (4):359–66.

[31] Rocha-Martinez JA, Navarrete-Gonzalez TD, Pavia-Miller C G, Paez-Hemandez R. Otto and Diesel engine models with cyclic variability. Revista Mexciana de Fisica 2002;48(3):228–34.

[32] Sieniutycz S, Shiner JS. Thermodynamics of irreversible processes and its relation to chemical engineering: Second law analyses and fnite-time thermodynamics. J Non Equilib thermodyn 1994;19:303-48.

[33] Wang P, Hou SS. Performance analysis and comparison of an Atkinson cycle coupled to variable temperature heat reservoirs under maximum power and maximum power density conditions. Energy Conversion & Management 2005;46:2637–55.

[34] Zhao Y, Chen J. Performance analysis and parametric optimum criteria of an irreversible Atkinson heat-engine. Applied Energy 2006;83:789–800.

8/5/2009