International Conference of Recent Advances in Engineering-ICRAE2013
Murch 2013, Islamic Azad University, Khomeinishahr Branch, Iran

Effects of mean piston speed and cut-off ratio on performance of Diesel engine

Mohsen Nasiri [1]

Department of Mechanical Engineering, Shahrekord University, Shahrekord, Iran

nasiri_mohsen@yahoo.com

Mohamad Hashemi Gahruei

Department of Mechanical Engineering, Shahrekord University, Shahrekord, Iran

mohamadhashemi879@gmail.com

Saeed Vahidi

Department of Mechanical Engineering, Shahrekord University, Shahrekord, Iran

savasoal@yahoo.com

Abstract

The performance of an air standard Diesel cycle is analyzed using finite-time thermodynamics. In the model, the linear relation between the specific heat ratio of the working fluid and its temperature, the friction loss computed according to the mean velocity of the piston and the heat transfer loss are considered. The relations between the power output and compression ratio and between the power output and the thermal efficiency are derived by detailed numerical examples. The results show that the power output and the thermal efficiency first increases and then start to decrease with increasing of compression ratio. Also, the point of maximum power output and thermal decreases with increasing mean piston speed and cut-off ratio. The conclusions of this investigation are of importance when considering the designs of actual Diesel engines.

Keywords: Diesel heat engine, Cut-off ratio, Mean piston speed, Finite-time Thermodynamic.

1-Corresponding Author

Introduction

The Finite-time thermodynamics is an extension of conventional thermodynamics relevant in principle across the entire span of the subject, from the most abstract level to the most applied. The approach is based on the construction of generalized thermodynamic potentials [1] for processes containing time or rate conditions among the constraints on the system [2] and on the determination of optimal paths that yield the extreme corresponding to those generalized potentials [3]. Many significant achievements have been made since introduction of finite-time thermodynamics in order to analyze and optimize the performances of real heat-engines [4,6]. Hoffman et al. [7] and Mozurkewich and Berry [8] used mathematical techniques, developed in optimal-control theory, to reveal the optimal motions of the pistons in Diesel cycle engine respectively. Aizenbud et al. [9] and Chen et al. [10] evaluated internal-combustion engine cycles using the optimal motion of a piston fitted in a cylinder containing a gas pumped at a specified heating-rate. Curzon and Ahlborn [11] extended the usefulness of the Carnot cycle by accounting for the irreversibility of finite-time heat-transfers; some authors have examined the finite-time thermodynamic performance of the Atkinson cycle. Parlaka [12] studied a comparative performance analysis of irreversible Dual and Diesel cycles under maximum power conditions. In this study, a comparative performance analysis and optimization based on maximum power and maximum thermal efficiency criteria have been performed for irreversible Dual and Diesel cycles. Optimal performance and design parameters, such as pressure ratio, cut-off ratio and extreme temperature ratio, of the cycles has been derived analytically and compared with each other based on maximum power and the corresponding thermal efficiency criteria. Al-Sarkhi et al. [13] investigated effects of friction and temperature-dependent specific-heat of the working fluid on the performance of a Diesel-engine. In this research, using finite-time thermodynamics, the relations between the power output, thermal efficiency and compression ratio have been derived. The effect of the specific heat of the working fluid, being temperature dependent, on the irreversible cycle performance, is significant. Akash [14] investigated effect of heat transfer on the performance of an air-standard diesel cycle. In this paper, it presents the effect of heat transfer on the net work output and the indicated thermal efficiency of the cycle. The heat losses through cylinder walls are considered to be proportional to the average temperature during heat addition process. The effects of other parameters, in conjunction with heat transfer, such as cutoff ratio and intake air temperature were also reported. Chen et al. [15] studied a friction effect on the characteristic performance of Diesel engines. In this model takes into account the finite-time evolution of the cycle's compression and power strokes and it considers global losses lumped in a friction-like term. The relations between the power output and the compression ratio, as well as between the thermal efficiency and the compression ratio are derived. The maximum power output with the corresponding efficiency, and the maximum efficiency with the corresponding power output are calculated versus compression ratio. Livanos et al. [16] performed a friction model of a marine diesel engine piston assembly. In modern marine diesel engines, power output and in-cylinder firing pressures are constantly increasing, leading to higher friction in engine components and especially in the piston assembly. A good understanding of the friction contributions of the various engine components is needed, if mechanical efficiency is to be improved. A friction model for the engine piston assembly has been developed and is presented in this paper. The model, based on lubrication theory, considers the detailed engine geometry and the complete lubricant action, and thus can be applied to a wide range of engines. In detail, the analysis takes into account the friction components of compression rings, oil control rings, piston skirt and gudgeon pin of the engine piston assembly.

Bhattacharyya [17] Optimized an irreversible Diesel cycle with fine tuning of compression ratio and cut-off ratio. A simplified irreversible model has been proposed for the air standard Diesel cycle. Global thermal and friction losses have been lumped into an equivalent friction term. Optimization of the cycle has been performed for power output as well as for thermal efficiency with respect to compression ratio and cut-off ratio. The optimum values of these ratios compare well with standard values used in real Diesel engines. The cycle also demonstrates a loop-shaped power versus efficiency curve as is exhibited by real heat engines. An irreversible cycle model of a Diesel heat engine is established by Zhao et al. [18] and used to investigate the influence of multi-irreversibilities, which mainly result from the non-isentropic compression and expansion processes, finite rate heat transfer processes and heat leak loss through the cylinder wall, on the performance of the cycle.

As can be seen in the relevant literature, the investigation of the effects of mean piston speed and cut-off ratio on performance of Diesel cycle with considering the variable specific heat ratio of the working fluid does not appear to have been published. Therefore, the objective of this study is to examine the effects of mean piston speed and cut-off ratio on performance of air standard Diesel cycle.

Thermodynamics simulation of air standard Diesel cycle

Fig. 1, show the pressure-volume diagram of thermodynamic process of an air standard Diesel cycle which all four phases of these irreversible cycles are considered. Diesel cycle is first introduced by Rudolf Diesel based his engine which is known as Diesel engine today.

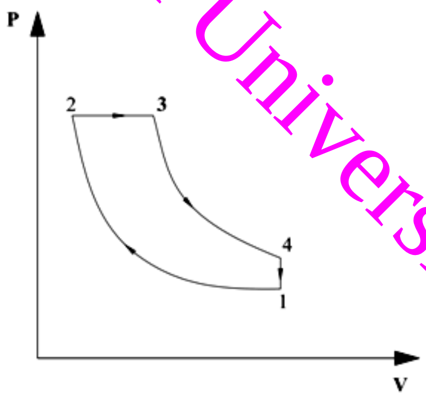

Figure (1) P–V diagram of air standard Diesel cycle

In an air standard Diesel, the compression process 1→2 is isentropic, heat is added to the cycle during process 2→3 which is an isobaric process, throughout isentropic process 3→4 the expansion is occurred, and the heat rejection process 4→1 is an isochoric process. As is usual in finite-time thermodynamic heat-engine cycle models, there are two instantaneous adiabatic-processes, namely 1→2 and 3→4. For the heat addition and heat rejection (2→3 and 4→1 stages, respectively), it is assumed that heating occurs from state 2 to state 3 and cooling ensues from state 4 to state 1 and proceed according to isothermal rates, as shown in Eq. (1):

$$\frac{dT}{dt} = \frac{1}{c_1} \text{ for}(2\rightarrow 3) \quad , \quad \frac{dT}{dt} = \frac{1}{c_2} \text{ for}(4\rightarrow 1) \qquad (1)$$

Where T is the absolute temperature and t is the time; C_1 and C_2 are constants. Integrating Eqs. (1) yield:

$$t_1 = c_1(T_3 - T_2) \; , \; t_2 = c_2(T_4 - T_1) \qquad (2)$$

Where t_1 and t_2 are heating and cooling periods, respectively. Then, the cycle period is

$$\tau = t_1 + t_2 = c_1(T_3 - T_2) + c_2(T_4 - T_1) \qquad (3)$$

As already mentioned in the previous section, it can be supposed that the specific heat ratio of the working fluid is a function of temperature alone and has the linear forms:

$$\gamma = \gamma_0 - k_1 T \qquad (4)$$

where γ is the specific heat ratio ($\gamma = c_p/c_v$) and T is the absolute temperature. γ_0 and k1 are constants.
During the process of 2→3 which is constant pressure process, the fluid is heated which could be calculated from the following equation:

$$Q_{in} = M\int_{T_2}^{T_3} C_p\,dT = M\int_{T_2}^{T_3}\left(\frac{(\gamma_0 - k_1 T)R}{\gamma_0 - k_1 T - 1}\right)dT$$
$$= \frac{MR}{k_1}\left[\ln\left(\frac{\gamma_0 - k_1 T_2 - 1}{\gamma_0 - k_1 T_3 - 1}\right) + k_1(T_3 - T_2)\right] \qquad (5)$$

And during the 4→1 process which is a constant volume process, the fluid inside the cylinder will release an amount of heat which output heat from this system can be determined from the following equation:

$$Q_{out} = M\int_{T_1}^{T_4} C_v\,dT = \int_{T_1}^{T_4}\left(\frac{R}{\gamma_0 - K_1 T - 1}\right)dT =$$
$$= \frac{MR}{k_1}\ln\left(\frac{\gamma_0 - K_1 T_1 - 1}{\gamma_0 - K_1 T_4 - 1}\right) \qquad (6)$$

Where, M is the mole number of the working fluid.
where cp is the molar specific heat at constant pressure for the working fluid.
According to Refs. [19,20], the equation for a reversible adiabatic process with variable specific heat ratio can be written as follows:

$$TV^{\gamma-1} = (T + dT)(V + dV)^{\gamma-1} \qquad (7)$$

Re-arranging Eqs. (1) and (4), we get the following equation:

$$T_i(\gamma_0 - k_1 T_j - 1) = T_j(\gamma_0 - k_1 T_i - 1)(V_j / V_i)^{\gamma_0 - 1} \qquad (8)$$

So, the isentropic process of 1→2 and 3→4 can be rewritten based on Eq. (11) as follows:

$$T_1(\gamma_0 - k_1 T_2 - 1) = T_2(\gamma_0 - k_1 T_1 - 1) r_c^{1-\gamma_0} \qquad (9)$$
$$T_3(\gamma_0 - k_1 T_4 - 1) = T_4(\gamma_0 - k_1 T_3 - 1)(r_c / \beta)^{\gamma_0 - 1} \qquad (10)$$

Where, in these equations rc and β are compression and cut-off ratios, which defined as:

$$r_c = \frac{V_1}{V_2} \qquad (11)$$
$$\beta = \frac{V_3}{V_2} \qquad (12)$$

In this study, in order to make conditions closer to the reality, the heat losses from heat transfer to outside of the cycle were noted. It could be presumed that heat loss from the cylinder wall is proportionate to the average temperature of fluid inside the cylinder and chamber wall. Therefore, the heat given to a fluid during an actual cycle can be calculated from the following equation [13]:

$$Q_{in} = M[A - B(T_2 + T_3)] \quad (13)$$

Where, in this equation A and B are constant amounts which are related to the heat transferred to walls and combustion.

Taking into account the friction loss of the piston and assuming a dissipation term represented by a friction force that is a linear function of the piston velocity gives [21, 22, 23]:

$$f_\mu = \mu \bar{S}_p = \mu \frac{dx}{xt} \quad (14)$$

where μ is a coefficient of friction that takes into account the global losses, $\bar{S}_p$ is the mean piston speed and x is the piston displacement. Then, the lost power is:

$$P_\mu = \frac{dW_\mu}{dt} = \mu \frac{dx}{dt}\frac{dx}{dt} = \mu \bar{S}_p^2 \quad (15)$$

with given amount of T_1 from Eq. (9), T_2 could be calculated. T_3 can be deduced by substituting Eq. (5) into Eq. (13). Then, by placing T_3 in Eq. (10), T_4 is also achieved.

Finally, given T_1, T_2, T_3, and T_4, the amount of power output could be calculated from the following equations and Thus, the power output $P_{output} = (W/\tau) - P_\mu$ can be written as:

$$P_{output} = \frac{Q_{in} - Q_{out}}{\tau} - P_\mu = \frac{MR}{k_1\tau}\left[ln\left(\frac{(\gamma_0 - k_1T_2 - 1)(\gamma_0 - k_1T_4 - 1)}{(\gamma_0 - k_1T_3 - 1)(\gamma_0 - k_1T_1 - 1)}\right) + k_1(T_3 - T_2)\right] - \mu\bar{S}_p^2 \quad (16)$$

And the efficiency of the cycle is:

$$\eta_{th} = \frac{P_{output}}{(Q_{in}/\tau)} = \frac{\frac{MR}{k_1}\left[ln\left(\frac{(\gamma_0 - k_1T_2 - 1)(\gamma_0 - k_1T_4 - 1)}{(\gamma_0 - k_1T_3 - 1)(\gamma_0 - k_1T_1 - 1)}\right) + k_1(T_3 - T_2)\right] - \mu\tau\bar{S}_p^2}{\frac{MR}{k_1}\left[ln\left(\frac{(\gamma_0 - k_1T_2 - 1)}{(\gamma_0 - k_1T_3 - 1)}\right) + k_1(T_3 - T_2)\right]} \quad (17)$$

Numerical Simulation and discussion

The following constants and parameters have been used in this exercise: A=60,000 J/mol, B = 25 J/mol K, M=1.57×10-5 kmol, T_1=300 K, μ = 12.9 NSm-1, $\bar{S}_p$= 7-15 ms-1, γ_0=1.4, k_1=0.00002-0.00011 K^{-1}, β=1.5–3. Taking equal heating and cooling periods, i.e., t_1=t_2=s/2=16.6 ms (s=33.33 ms), the isothermal rates C_1 and C_2 are estimated as C_1=8.128×10^{-6} s/K and C_2=18.67×10^{-6} s/K [18-25]. Substituting T_1, T_2, T_3, T_4 and T_5 into Eqs. (5) and (6) yields the heat in and heat out. Then, substituting heat in, heat out and lost power into Eqs. (16) and (17) yields the power output and thermal efficiency. Therefore, the relations between the power output, the thermal efficiency and the compression ratio can be derived.

Figs. 2–5 show the effects of the variable specific heat ratio of the working fluid on the performance of the cycle with heat resistance and friction irreversible-losses. The power output versus compression ratio characteristic is approximately parabolic-like and the power output versus thermal efficiency is looped shaped. From these figures, it can be found that mean piston speed and cut-off ratio play a key role on the power output and the thermal efficiency. It is clearly seen that the effects of β, and $\bar{S}_p$ on the power output and thermal efficiency are related to the compression ratio. They reflect the performance characteristics of a real irreversible Diesel cycle engine.

Figs. 2 and 3 show the effects of cut-off ratio on the performance of the Diesel cycle. Figs. 2 and 3 show that the points of maximum thermal efficiency and power output of cycle will decreases with an increase of cut-off ratio. The maximum power output of Diesel cycle increases sharply with decrease of β: When β decreases from 3 to 1.5, the maximum power output of Diesel cycle will increase from 1.93 KW to 4.40 KW (about 128%). Fig. 2 shows that with increase of β the points of maximum power output occur at the higher compression ratio. The influence of the cut-off ratio on the power output versus thermal efficiency is displayed in Fig. 3. As can be seen from this figure, the power output versus thermal efficiency is a loop shaped one. It can be seen that the power output at maximum thermal efficiency improves with decreasing cut-off ratio from 3 to 1.5.

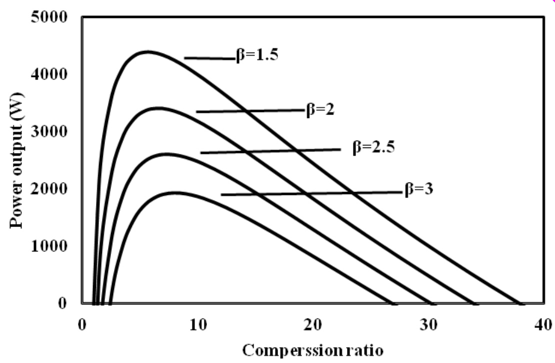

Figure (2) Effect of cut-off ratio on P_{Diesel}-r_c characteristic for $\bar{S}_p$=10 m/S, k_1=0.00008 K^{-1}.

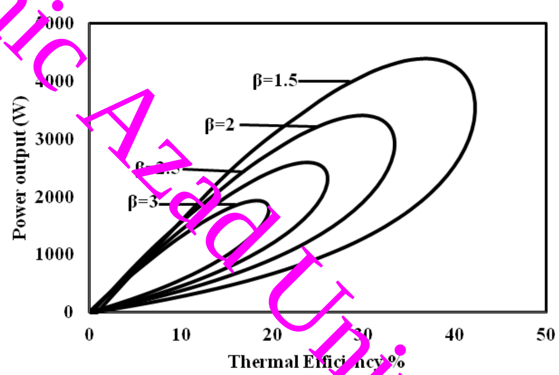

Figure (3) Effect of cut-off ratio on P_{Diesel}- η_{Diesel} characteristic for $\bar{S}_p$=10 m/S, k_1=0.00008 K^{-1}.

Figs. 4 and 5 show the effects of mean piston speed on the performance of the Diesel cycle. Figs 4 and 5 show that the points of maximum thermal efficiency and power output of cycle will decreases with an increase of mean piston speed. This happens due to the increase of the lost power by the mean piston speed (see Eq.(15)). The maximum power output of Diesel cycle increases sharply with decrease of $\bar{S}_p$: When $\bar{S}_p$ decreases from 15 m/s to 7 m/s, the maximum power output of Diesel cycle will increase from 1.79 KW to 4.06 KW (about 126.8%). The influence of the mean piston speed on the power output versus thermal efficiency is displayed in Fig. 5. As can be seen from this figure, the power output versus thermal efficiency is a loop shaped one. It can be seen that the power output at maximum thermal efficiency improves with decreasing mean piston speed from 15 m/s to 7 m/s.

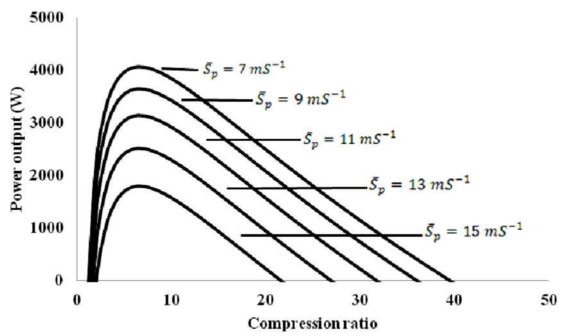

Figure (4) Effect of mean piston speed on P_{Diesel}-r_c characteristic for β=2, k_1=0.00008 K^{-1}.

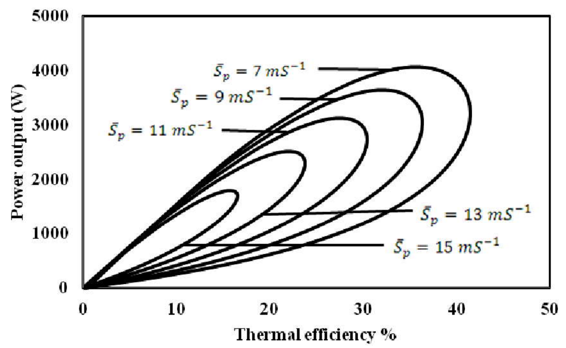

Figure (5) Effect of mean piston speed on P_{Diesel}-η_{Diesel} characteristic for β=2, k_1=0.00008 K^{-1}.

Conclusion

This study is aimed at investigating the effects of mean piston speed, cut-off ratio and variable specific heat ratio of the working fluid on the Diesel cycle's performance. By using finite time thermodynamics theory, the characteristic curves of the power output versus compression ratio and the power output versus thermal efficiency are obtained. In the model, the linear relation between the specific heats ratio of working fluid and its temperature, the frictional loss computed according to the mean velocity of the piston, and heat transfer loss are considered. The results show that the mean piston speed and cut-off ratio plays a significant role on the Diesel cycle performance. And, the points of maximum power output and thermal efficiency of Diesel cycle will decreases with an increase of mean piston speed and cut-off ratio. Also, with increase of cut-off ratio the points of maximum power output occur at the higher compression ratio. The results obtained from this research may be used with assurance to provide guidance for the analysis of the behavior and design of practical Diesel engines.

References

[1] Hermann, R.,1973, *Geometry, Physics and Systems, Dekker*, New York.

[2] Salamon, P.; Andresen, B.; Berry, R.S.,1977, *Thermodynamics in finite time*. II. Potentials for finite-time processes. Phys. Rev. A. 14, 2094-2102.

[3] Mozurkewich, M.; Berry, R.S., 1981, "Finite -time thermodynamics, Engine performance improved by optimized piston motion". *J. Appl. Phys*, 78, 1986–1998.

[4] Chen, L.; Wu, C.; Sun, F., 1999, "Finite-time thermodynamic optimization or entropy-generation minimization of energy systems". *J. Non-Equil. Thermody*, 24, 327–339.

[5] Bejan, A., 1996, "Entropy-generation minimization, the new thermodynamics of finite-size devices and finite time processes", *J. Appl. Phys*, 79, 1191–1218.

[6] Chen, L.; Sun, F., 2004, "Advances in finite-time thermodynamics, analysis and optimization", NewYork, Nova Science Publishers.

[7] Hoffman, K.H.; Watowich, S.J.; Berry, R.S., 1985, "Optimal paths for thermodynamic systems, the ideal Diesel cycle", *J. Appl. Phys*, 58, 2125 –2134.

[8] Mozurkewich, M.; Berry, R.S., 1982, "Optimal paths for thermodynamic systems, the ideal Otto–cycle", *J. Appl. Phys*, 53, 34–42.

[9] Aizenbud, B.M.; Band, Y.B.; Kafri, O., 1982, "Optimization of a model internal-combustion engine", *J. Appl. Phys*, 53, 1277 –1282.

[10] Chen, L.; Sun, F.; Wu, C., 1998, "Optimal expansion of a heated working-fluid with linear phenomenological heat-transfer", *Energy Conversion and Management*, 39, 149–156.

[11] Curzon, F.L.; Ahlborn, B., 1975, "Efficiency of a Carnot engine at maximum power-output", *Am. J. Phys*, 43, 22–24.

[12] Parlak, A., 2005, "Comparative performance analysis of irreversible Dual and Diesel cycles under maximum power conditions". *Energy Conversion and Management*, 46, 351–359.

[13] Al-Sarkhi, A.; Jaber, J.O.; Abu-Qudais, M.; Probert, S.D., 2006, "Effects of friction and temperature-dependent specific-heat of the working fluid on the performance of a Diesel-engine",*Applied Energy*, 83, 153-165.

[14] Akash, B.A., 2001, "Effect of heat transfer on the performance of an air-standard diesel cycle", *Int. Comm. Heat Maw Transfer*, 28 (1), 87-95.

[15] Chen, L.; Lin, J.; Luo, J.; Sun, F.; Wu, C., 2002, "Friction effect on the characteristic performance of Diesel engines", *Int. J. Energy Res*, 26, 965–971.

[16] Livanos, G. A.; Kyrtatos, N. P., 2007, "Friction model of a marine diesel engine piston assembly", *Tribology International*, 40, 1441–1453.

[17] Bhattacharyya, S., 2000, "Optimizing an irreversible Diesel cycle-fine tuning of compression ratio and cut-off ratio", *Energy Conversion and Management*, 41, 847-854.

[18] Zhao, Y.; Lin, B.; Zhang, Y.; Chen, J., 2006, "Performance analysis and parametric optimum design of an irreversible Diesel heat engine", *Energy Conversion and Management*, 47, 3383 –3392.

[19] Ge, Y.; Chen, L.; Sun, F.; Wu, C., 2005, "Effects of heat transfer and variable specific heats of working fluid on performance of a Miller cycle", *International Journal of Ambient Energy*, 26 (4), 203–214.

[20] Ebrahimi,R.,2011,"Thermodynamicmodeling of performance of a Miller cycle with engine speed and variable specific heat ratio of working fluid", *Computers and Mathematics with Applications*, 62, 2169–2176.

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

[22] Ge, Y.; Chen, L.; Sun, F.; Wu, C., 2006, "Performance of an Atkinson cycle with heat transfer, friction and variable specific heats of working fluid", *Applied Energy*, 83 (11), 1210–1221.

[23] Lin, J.C.; Hou, S.S., 2007, "Influence of heat loss on the performance of an air-standard Atkinson cycle",*Applied Energy*,84,904–920.

[24] Abbassi, M.B.; Gahruei, M.H.; Vahidi, S., 2012, "Comparison of the Performances of Biodiesel, Diesel, and Their Compound in Air Standard Diesel-Atkinson Cycle", *J. American Science*, 8(1), 223-229.

[25] Ebrahimi, R., 2011, "Effects of mean piston speed, equivalence ratio and cylinder wall temperature on performance of an Atkinson engine", *Mathematical and Computer Modelling*, 53, 1289–1297.