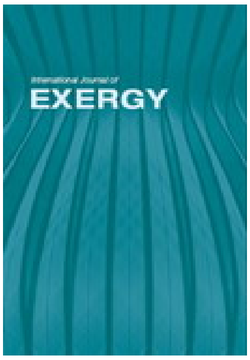


International Journal of Exergy

ISSN online: 1742-8300 - ISSN print: 1742-8297
https://www.inderscience.com/ijex

Introduction of a performance analysis criterion called effective exergetic performance coefficient and application to an engine operated on seven-process cycle

Guven Gonca, Ibrahim Genc, Mehmet Fatih Hocaoğlu

DOI: 10.1504/IJEX.2025.10068583

Article History:

Received:	08 May 2023
Last revised:	01 May 2024
Accepted:	06 May 2024
Published online:	07 January 2025

Introduction of a performance analysis criterion called effective exergetic performance coefficient and application to an engine operated on seven-process cycle

Guven Gonca*

Department of Naval Architecture and Marine Engineering,
Yildiz Technical University,
Besiktas, 34349, Istanbul, Turkey
Fax: +90-2123832989
Email: ggonca@yildiz.edu.tr
Email: guvengca@gmail.com
*Corresponding author

Ibrahim Genc and Mehmet Fatih Hocaoğlu

Department of Electrical-Electronics Engineering,
Istanbul Medeniyet University,
Kadikoy, 34720, Istanbul, Turkey
Email: ibrahim.genc@medeniyet.edu.tr
Email: genc.ibrahim@gmail.com
Email: mfatih.hocaoglu@medeniyet.edu.tr

Abstract: In a few decades, the researchers studying on the optimisation and improvement of performance specifications and emission formations of the internal combustion engines have turned the attention on application of different cycles such Miller cycle and Takemura cycle into internal combustion engines. In this work, a novel analysis criterion named as effective exergetic performance coefficient (EFEXPEC) have been presented and applied to seven-process cycle consisting of Takemura cycle and Miller cycle. Maximum performance specifications such as maximum thermal efficiency, maximum power output, maximum EFEXPEC, power at maximum EFEXPEC and efficiency at maximum EFEXPEC have been examined. The consequences can be assessed by researchers who work on ICEs to actualise the proposed combination practically and to determine maximum EFEXPEC conditions. The maximum value of power (P_{MAX}) is 27.3 kW and it has been obtained at 6,000 rpm and 20 of compression ratio. The maximum value of thermal efficiency is 40.15% and it has been obtained at 0.9 of equivalence ratio and 20 of compression ratio. The maximum value of EFECPEC ($EFECPEC_{MAX}$) is 0.18 and it has been obtained at 0.9 of equivalence ratio and 14 of compression ratio.

Keywords: dual-miller cycle; Takemura cycle; performance analysis; thermo-ecology; engine performance; internal combustion engines.

Reference to this paper should be made as follows: Gonca, G., Genc, I. and Hocaoğlu, M.F. (2025) 'Introduction of a performance analysis criterion called effective exergetic performance coefficient and application to an engine operated on seven-process cycle', *Int. J. Exergy*, Vol. 46, No. 1, pp.1–12.

Biographical notes: Guven Gonca received his BSc, MSE and PhD in Naval Architecture and Marine Engineering in 2007, 2009 and 2013 from the Yildiz Technical University. His MS research project was about detail design and modelling in ship outfitting. He has been to University of California at Berkeley (2012–2013) as a visiting researcher. His PhD thesis subject was about investigation of the effects of steam injection into the turbocharged diesel engine with running Miller cycle on performance and emissions. He is still working as an Associate Professor in Naval Architecture and Marine Engineering at the Yildiz Technical University.

Ibrahim Genc received his BSc, MSE and PhD in Electronics and Communication Engineering from the Istanbul Technical University in 1992, 1996 and 2008, respectively. He worked as a research and teaching assistant at the Ondokuz Mayis University and as a senior researcher at the Marmara Research Center of Turkish Scientific and Research Council. He co-founded two start-ups and he is still with Agena Information and Defense Technologies LLC. Since December 2011, he is an Assistant Professor at the Department of Electrical-Electronics Engineering at Istanbul Medeniyet University. His current research interests include artificial neural networks, systems and modelling for simulation. He was the recipient of the TUBITAK Munir Birsel Foundation PhD Scholarship and he was a visiting researcher at the Institute for Creative Technologies of USC in 2017.

Mehmet Fatih Hocaoğlu received his BSc, MSE, and PhD in Industrial Engineering in 1991, 1994, and 2001 from the Istanbul Technical University, the Yıldız Technical University, and the Sakarya University, respectively. He worked as a research and teaching assistant at the Karadeniz Technical University and as a senior researcher at the Marmara Research Center of the Turkish Scientific and Research Council. He co-founded two start-ups and is currently with Agena Information and Defense Technologies LLC. He is an Associate Professor in the Department of Industrial Engineering at Istanbul Medeniyet University. His current research interests include modelling and simulation, artificial intelligence, and operations research.

1 Introduction

The researchers and engineers work to develop more efficient and eco-friendly combustion engines (ICEs) by virtue of economical and environmental restrictions. The application of Miller cycle (AMC) is more advantageous due to lower cost and ease of application to minimise NOx formation of the ICEs compared to other emission control methods, therefore the scientists have focused on the AMC recently (Gonca, 2017; Gonca et al., 2013, 2015a, 2015b, 2015c, 2015d, 2017; Mikalsen et al., 2009). Gonca (2017) performed an ecological-based performance analysis based on ecological coefficient of performance (ECOP) criterion. Gonca et al. (2013) introduced the dual MC by integrating the classical MC to dual diesel cycle and they parametrically acquired the grid performance curves for the dual MC. Gonca et al. (2015a) theoretically performed a combination of the MC and steam injection method (SIM) and decreased the NO emissions. Gonca et al. (2015b) determined combustion and heat transfer constants for the dual MC engine and examined the impact of the constants on the dual MC engine. Gonca et al. (2015c, 2015d, 2017) actualised empirical and computational studies to acquire performance properties and emission values of a diesel engine with the AMC

(Gonca et al., 2015c), of a diesel engine with the AMC and turbo charging (Gonca et al., 2015d), of a diesel engine with the AMC, steam injection technique and turbo charging together (Gonca et al., 2017). Mikalsen et al. (2009) presented the performance specifications of a combined heat and power generation system including natural gas engine operating on Otto cycle and AMC. Takemura cycle provides combustion at constant temperatures (Kamiuto, 2006). Hence, these cycles can be unified for performance improvement and emission reduction. In the literature, there are a few studies on the seven-process cycle (Gonca and Sahin, 2019; Gonca et al., 2020, 2022) and it has been firstly proposed by Gonca and Sahin (2019). There are so many works based on simulation models in the literature for the engines and their cycles (Andresen, 1983; Bejan, 1996; Chen et al., 1999; Chen and Xia, 2017). Also, the researchers studied on different performance analysis criteria such as ECOP, exergy, power density, efficient power (Fawal and Kodal, 2021; Li et al., 2021; Karakurt et al., 2022; Caglayan and Caliskan, 2019; Patodi and Maheshwari, 2013), etc.

This work reports an ecology-based performance optimisation and analysis for the Dual-Miller cycle and Takemura cycle combination based on a novel performance analysis criterion called effective exergetic performance coefficient (*EFEXPEC*). The impacts of design and operational parameters (*DOP*) on the maximum EFEXPEC conditions (*MEX*), power output and efficiency at the MEX have been investigated by graphs and illustrations based on grid curves.

2 Theoretical model

This work examined the maximum EFEXPEC specifications of the studied cycle which is presented in Figure 1. Computational performance examinations have been conducted to examine the performance specifications depending on the engine DOP by using a finite-time thermodynamics model. Table 1 shows the standard values of the parameters used for design and operation.

The presented performance analysis criterion named as *EFEXPEC* is derived as following:

$$EFEXPEC = \frac{\eta_{ex} P}{T_0 \alpha \varepsilon \dot{S}_{gen}} \tag{1}$$

where η_{ex}, T_0 and $\dot{S}_{gen}$ are exergetic efficiency, ambient temperature (K) and entropy generation per second, α is cycle temperature ratio, ε is cycle temperature ratio, they are given as follows:

$$\alpha = \frac{T_{\max}}{T_{\min}} = \frac{T_4}{T_1} \tag{2}$$

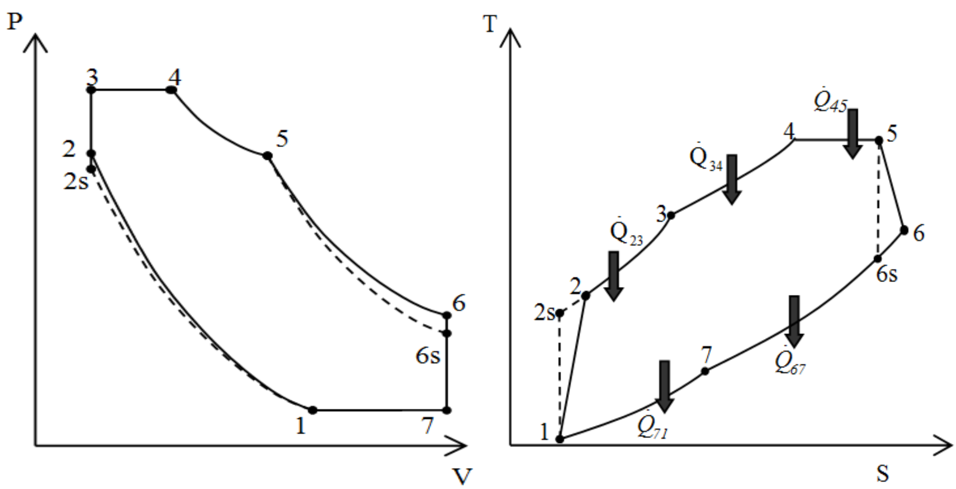
$$\varepsilon = \frac{V_{\max}}{V_{\min}} = \frac{V_T}{V_c} = \frac{V_7}{V_2} \tag{3}$$

Table 1 The engine properties and values of design parameters

Parameter	*Symbol*	*Value*	*Unit*
Intake temperature	T_1	300	K
Intake pressure	P_1	100	kPa
Miller cycle angle	θ_r	10	Ca
Stroke length	L	0.062	m
Bore	d	0.072	m
Cylinder wall temperature	T_0	400	K
Friction coefficient	μ	0.0129	Ns/m

Source: Gonca and Sahin (2019)

Figure 1 *T*-*s* and *P*-*v* diagram of the irreversible seven-process cycle



V_T is total cylinder volume which is given as follows:

$$V_T = V_s + V_c = \frac{(V_s r)}{r-1} \tag{4}$$

$$V_c = \frac{V_T}{r} = \frac{\pi d^2 L}{4} \frac{1}{r-1} \tag{5}$$

where V_c states clearance volume, r is compression ratio, d is the bore diameter (m), L is engine stroke (m). Other equations used in this study have been obtained from previous studies (Gonca and Sahin, 2019; Gonca et al., 2020, 2022). Power output is given below:

$$P_{ef} = \dot{Q}_{in} - \dot{Q}_{out} - P_{fr}, \tag{6}$$

The heat input ($\dot{Q}_{in}$), the heat output ($\dot{Q}_{out}$) and the power dissipated by friction is respectively determined as follows:

$$
\begin{aligned}
\dot{Q}_{in} &= \dot{Q}_{f,c} - \dot{Q}_{ht} \\
&= \dot{m}_T \left[\int_{T_2}^{T_3} C_V dT + \int_{T_3}^{T_4} C_P dT + R_g T_5 \ln\left(r_T\right) \right] \\
&= \dot{m}_T \left(\left[\left(2.506 \cdot 10^{-11} \frac{T^3}{3} + 1.454 \cdot 10^{-7} \frac{T^{2.5}}{2.5} - 4.246 \cdot 10^{-7} \frac{T^2}{2} + 3.162 \cdot 10^{-5} \frac{T^{1.5}}{1.5} \right.\right.\right. \\
&\left.\left. + 1.0433 T - 1.512 \cdot 10^4 \left(-\frac{T^{-0.5}}{0.5} \right) + 3.063 \cdot 10^5 \left(-T^{-1} \right) - 2.212 \cdot 10^7 \left(-\frac{T^{-2}}{2} \right) \right) \right]_{T_2}^{T_3} \\
&+ \left[\left(2.506 \cdot 10^{-11} \frac{T^3}{3} + 1.454 \cdot 10^{-7} \frac{T^{2.5}}{2.5} - 4.246 \cdot 10^{-7} \frac{T^2}{2} + 3.162 \cdot 10^{-5} \frac{T^{1.5}}{1.5} \right.\right. \\
&\left.\left. + 1.3303 T - 1.512 \cdot 10^4 \left(-\frac{T^{-0.5}}{0.5} \right) + 3.063 \cdot 10^5 \left(-T^{-1} \right) - 2.212 \cdot 10^7 \left(-\frac{T^{-2}}{2} \right) \right) \right]_{T_3}^{T_4} \\
&\left. + R_g T_5 \ln\left(r_T\right) \right)
\end{aligned} \tag{7}
$$

$$
\begin{aligned}
\dot{Q}_{out} &= \dot{m}_T \left[\int_{T_7}^{T_6} C_V dT + \int_{T_1}^{T_7} C_P dT \right] \\
&= \dot{m}_T \left(\left[\left(2.506 \cdot 10^{-11} \frac{T^3}{3} + 1.454 \cdot 10^{-7} \frac{T^{2.5}}{2.5} - 4.246 \cdot 10^{-7} \frac{T^2}{2} + 3.162 \cdot 10^{-5} \frac{T^{1.5}}{1.5} \right.\right.\right. \\
&\left.\left. + 1.0433 T - 1.512 \cdot 10^4 \left(-\frac{T^{-0.5}}{0.5} \right) + 3.063 \cdot 10^5 \left(-T^{-1} \right) - 2.212 \cdot 10^7 \left(-\frac{T^{-2}}{2} \right) \right) \right]_{T_7}^{T_6} \\
&+ \left[\left(2.506 \cdot 10^{-11} \frac{T^3}{3} + 1.454 \cdot 10^{-7} \frac{T^{2.5}}{2.5} - 4.246 \cdot 10^{-7} \frac{T^2}{2} + 3.162 \cdot 10^{-5} \frac{T^{1.5}}{1.5} \right.\right. \\
&\left.\left.\left. + 1.3303 T - 1.512 \cdot 10^4 \left(-\frac{T^{-0.5}}{0.5} \right) + 3.063 \cdot 10^5 \left(-T^{-1} \right) - 2.212 \cdot 10^7 \left(-\frac{T^{-2}}{2} \right) \right) \right]_{T_1}^{T_7} \right)
\end{aligned} \tag{8}
$$

$$P_{fr} = \mu \left(V_{ap} \right)^2 \tag{9}$$

The friction coefficient is defined as μ, average velocity of the piston is described as below:

$$V_{ap} = \frac{L \cdot N}{30} \tag{10}$$

where N is engine speed in revolution per minute (rpm) and L is stroke length in metre (m). $\dot{Q}_f$ is heat dissipation depending on the burned fuel:

$$\dot{Q}_f = \eta_c \dot{m}_f LHV \tag{11}$$

where LHV is fuel lower heating value. $\dot{m}_f$ is the fuel mass flow rate (kg/s) and it is attained as below:

$$\dot{m}_f = \frac{m_f N}{120} \tag{12}$$

The mass of the injected fuel per cycle (kg) is given as m_f. The combustion efficiency is described as η_c which is obtained as below (Ebrahimi, 2011, 2012):

$$\eta_c = -1{,}44738 + 4{,}18581/\phi - 1{,}86876/\phi^2 \tag{13}$$

The equivalence ratio (ϕ) is defined as follows:

$$\phi = \frac{\left(m_f/m_a\right)}{F_{st}} \tag{14}$$

where m means mass per cycle in kilogram (kg), subscript a and f denote air and fuel. F_{st} is stoichiometric fuel/air ratio. The air mass per cycle and F_{st} are derived as follows:

$$m_a = \rho_a V_a = \rho_a \left(V_T - V_{rg}\right) \tag{15}$$

$$F_{st} = \frac{\varepsilon \cdot (12.01 \cdot \alpha + 1.008 \cdot \beta + 16 \cdot \gamma + 14.01 \cdot \delta)}{28.85} \tag{16}$$

Atomic number of carbon (C), hydrogen (H), oxygen (O) and nitrogen (N) in fuel are respectively symbolised by α, β, γ, δ. The molar fuel/air ratio is ε which is given as follows (Ferguson, 1986):

$$\varepsilon = \frac{0.21}{\left(\alpha - \frac{\gamma}{2} + \frac{\beta}{4}\right)} \tag{17}$$

ρ means density which is obtained depending on inlet temperature and pressure as follows:

$$\rho_a = f\left(T_1, P_1\right) \tag{18}$$

where f denotes function. The function values are evaluated by Engineering Equation Solver (EES) software (EES Academic Professional Edition, 2022). The chemical formula of the diesel fuel is $C_{14.4}H_{24.9}$ (Ferguson, 1986). $\dot{Q}_{ht}$ in equation (7) signifies energy loss depending on heat transfer and it is defined as follows:

$$\dot{Q}_{ht} = h_{tr} A_{cyl} \left(T_{avg} - T_0\right) = h_{tr} A_{cyl} \left(\frac{T_2 + T_5}{2} - T_0\right) \tag{19}$$

where h_{tr} is coefficient of heat transfer which is described as below (Hohenberg, 1979):

$$h_{tr} = 130 V_T^{-0.06} P_1^{0.8} T_{mix}^{0.4} \left(\overline{S}_p + 1.4\right)^{0.8} \tag{20}$$

The T means temperature, subscripts 0 and avg are cylinder wall temperature and average temperature of the in-cylinder working fluid. The A means surface area in contact with the working fluid in square meter (m^2), subscript cyl signifies cylinder. A_{cyl} is derived as follows:

$$A_{cyl} = \pi dL \frac{r}{r-1} + \frac{\pi d^2}{2} \tag{21}$$

The average temperature of air-residual gas mixture (T_{mix}) is acquired as below:

$$T_{mix} = \frac{\dot{m}_a T_1 R_a + \dot{m}_{rg} T_1 R_{rg}}{\dot{m}_a R_a + \dot{m}_{rg} R_{rg}} \tag{22}$$

R signifies gas-constant, subscript rg indicate residual gas. $\dot{m}$ signifies mass flow rate in kilogram per second (kg/s). Subscript T means total fluid flow rate. They are derived as follows:

$$\dot{m}_T = \dot{m}_a + \dot{m}_f + \dot{m}_{rg}, \tag{23}$$

$$\dot{m}_a = \frac{m_a N}{120} = \frac{\dot{m}_f F_{st}}{\phi}, \tag{24}$$

$$\dot{m}_{rg} = \frac{m_{rg} N}{120} = \dot{m}_a RGF, \tag{25}$$

The residual gas fraction (RGF) is determined as the ratio of residual gas to total introduced working fluid. The cylinder bore diameter (d) is given in meter.

3 Results and discussion

The impact of the DOP on the maximum power (P_{MAX}), on the maximum thermal efficiency (η_{MAX}), maximum EFECPEC ($EFEXPEC_{MAX}$), on the power at the maximum EFEXPEC (P_{MEX}), on the thermal efficiency at the maximum EFEXPEC (η_{MEX}) of a engine operating on the studied cycle is shown by illustrations. The Takemura cycle ratio was changed to obtain maximum values of the performance specifications.

Figures 2(a)–2(c) demonstrate the influences of equivalence ratio (ϕ) and compression ratio (r) and on the P_{MAX}, η_{MAX}, $EFEXPEC_{MAX}$, P_{MEX} and η_{MEX}. The power output values increases between 0.8 and 1 of the equivalence ratio and then it diminishes at the higher values of the equivalence ratio. However, the thermal efficiency values enhances to 0.9 of the ϕ and then it diminishes between 0.9 and 1.5 of the ϕ. The variation trend of the $EFEXPEC_{MAX}$ is similar to that of thermal efficiency at the maximum EFECPEC for a constant r. The compression ratio has the positive effect on the P_{MAX}, P_{MEX}, η_{MAX} and η_{MEX} but it does negatively affect the $EFEXPEC_{MAX}$, because although the power output increases the maximum combustion temperature and exergy destruction also increase. Therefore, the EFEXPEC decreases with increasing compression ratio. The highest and lowest values of the P_{MAX}, η_{MAX}, $EFEXPEC_{MAX}$, P_{MEX} and η_{MEX} are 15.05 kW, 11.88 kW, 40.15%, 21.48%, 0.1796, 0.0405, 14.93 kW, 11.16 kW and 39.83%, 21.15%, respectively.

Figure 2 The influences of r and ϕ on, (a) P_{MAX} and η_{MAX} (b) $EFECPEC_{MAX}$ and P_{MEX} (c) $EFECPEC_{MAX}$ and η_{MEX}

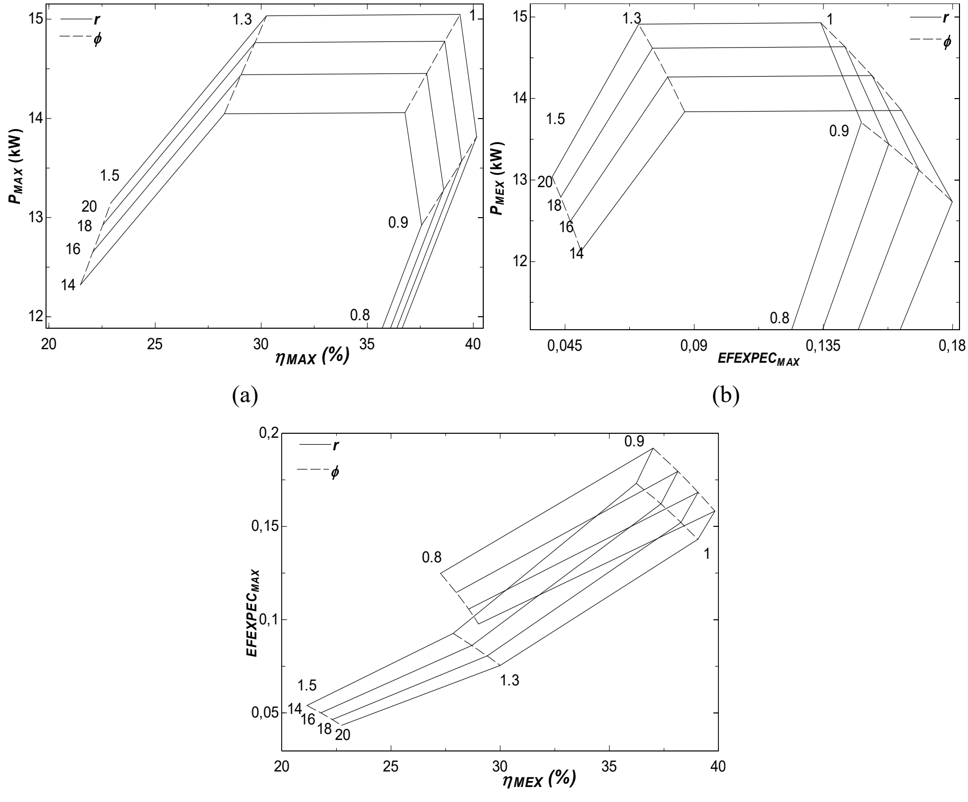


Figure 3 The influences of r and N on, (a) P_{MAX} and η_{MAX} (b) $EFECPEC_{MAX}$ and P_{MEX} (c) $EFECPEC_{MAX}$ and η_{MEX}

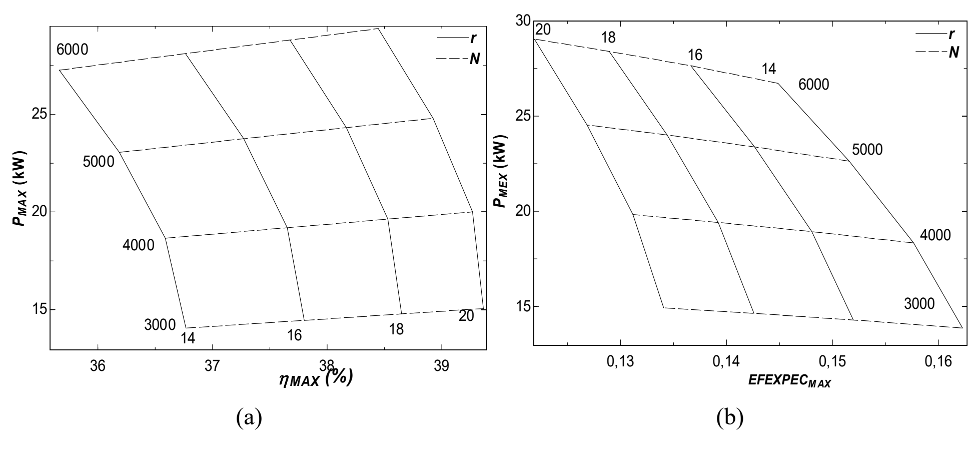
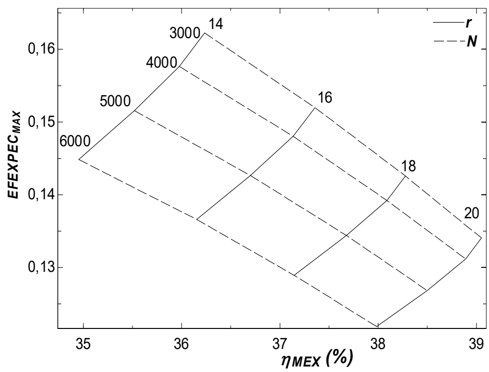
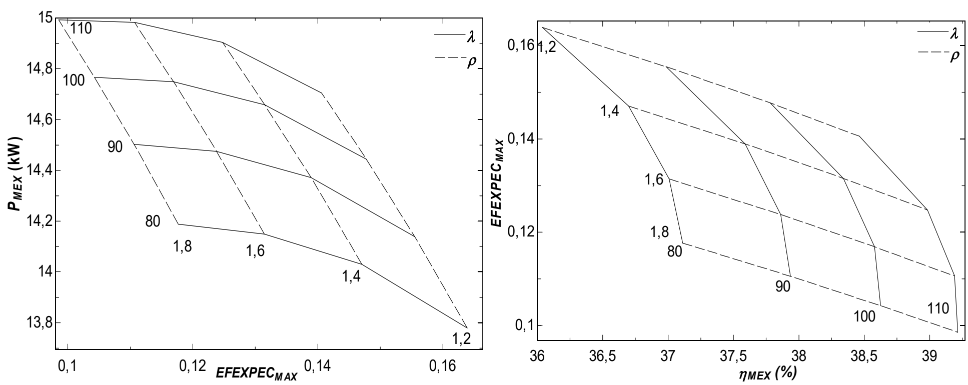


Figure 3 The influences of *r* and *N* on, (a) P_{MAX} and η_{MAX} (b) $EFECPEC_{MAX}$ and P_{MEX} (c) $EFECPEC_{MAX}$ and η_{MEX} (continued)



(c)

Figure 4 The influences λ and ρ on, (a) $EFECPEC_{MAX}$ and P_{MEX} (b) $EFECPEC_{MAX}$ and η_{MEX}



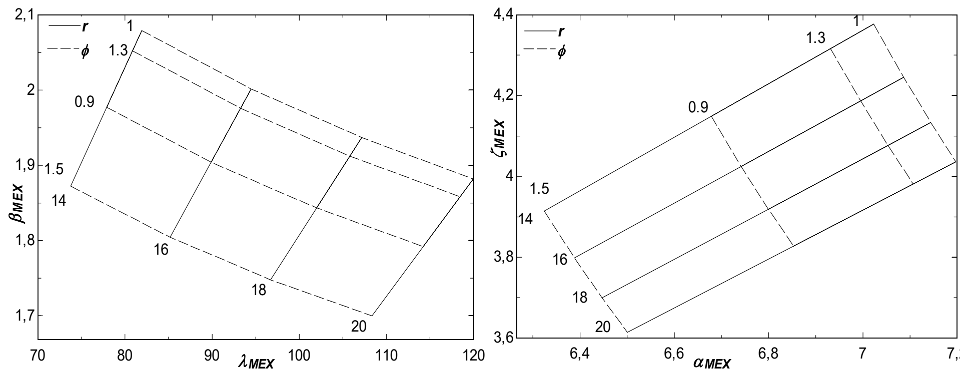
(a) (b)

Figures 3(a)–3(c) demonstrate the influences of *r* and speed (*N*) on the P_{MAX}, η_{MAX}, $EFEXPEC_{MAX}$, P_{MEX} and η_{MEX}. The engine speed has the reverse effect of the *r*. P_{MAX} and P_{MEX} increase and the EFEXPEC decreases with increasing engine speed. However, the thermal efficiency values decreases with decreasing *r* and enhancing *N* since the friction losses, maximum combustion temperature, entropy generation increase with swelling *N* like as compression ratio. The maximum and minimum values of P_{MAX}, η_{MAX}, $EFEXPEC_{MAX}$, P_{MEX} and η_{MEX} are 29.4 kW, 14.06 kW, 39.36%, 35.66%, 0.1622, 0.1219, 29.04 kW, 13.85 kW and 39.05%, 34.95%, respectively.

Figures 4(a)–4(b) display the effects of cycle pressure ratio (λ) and cut-off ratio (ρ) on the $EFEXPEC_{MAX}$, P_{MEX} and η_{MEX}. The P_{MEX} and η_{MEX} increase with increasing λ and ρ. However, the maximum EFEXPEC decreases with increasing λ and ρ due to dominant effects of entropy generation and maximum combustion temperatures. The highest and lowest values of $EFEXPEC_{MAX}$, P_{MEX} and η_{MEX} are 0.1639, 0.09853, 14.99 kW, 13.78 kW and 39.21%, 36.04%, respectively.

Figures 5(a)–5(b) display the impacts of the *r* and ϕ on the pressure ratio (β) at the MEX (β_{MEX}), λ at the MEX (λ_{MEX}), exhaust-temperature ratio (ζ) at the MEX (ζ_{MEX}) and cycle temperature ratio (α) at the MEX (α_{MEX}). Enhancing r causes to swell in λ_{MEX}, α_{MEX} and reduction in the β_{MEX} and ζ_{MEX} at the maximum EFEXPEC. However, there is not a steady variation of β_{MEX}, λ_{MEX}, ζ_{MEX} and α_{MEX} depending on the ϕ variation. They have the maximum and minimum values equivalence ratio at the 1 and 1.5.

Figure 5 The influences of ϕ and r on, (a) β_{MEX} and λ_{MEX} (b) ζ_{MEX} and α_{MEX}



(a) (b)

Figure 6 The influences of N and r on, (a) β_{MEX} and λ_{MEX} (b) ζ_{MEX} and α_{MEX}

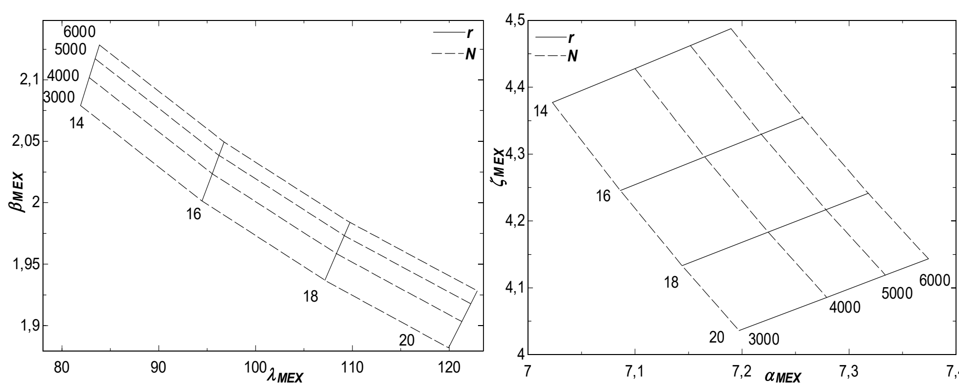


Figure 7 The influences of λ and ρ on the β_{MEX} and ζ_{MEX}

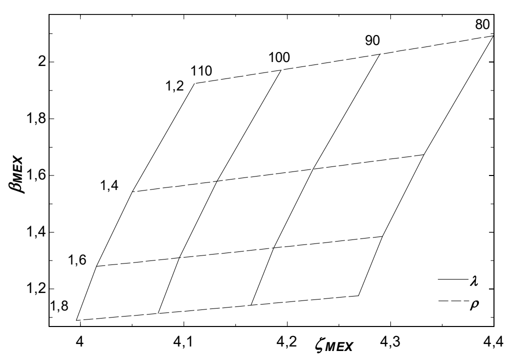


Figure 6 show the influences of the r and N on the β_{MEX}, λ_{MEX}, ζ_{MEX} and α_{MEX}. All of them increase with enhancing N due to frictions and higher combustion temperatures.

Figure 7 demonstrates the impacts of λ and ρ on the variation of the β_{MEX} and ζ_{MEX}. They minimise with increasing λ and ρ since equivalence ratio is constant and so heat input during the combustion process constant. Therefore, the β_{MEX} and ζ_{MEX} reduce, as λ and ρ enhance in order to provide constant heat input for a cycle.

4 Conclusions

The P_{MAX}, η_{MAX}, P_{MEX}, η_{MEX} and $EFEXPEC_{MAX}$ have lower values at the higher and lower of values of the ϕ. They have higher values at the average values of it. The compression ratio has the positive effect on the P_{MAX}, η_{MAX}, P_{MEX}, η_{MEX} but it does negatively affect the $EFEXPEC_{MAX}$. The engine speed has the similar effect as the r. The P_{MAX}, P_{MEX} increase and the $EFEXPEC$ decreases with swelling N. However, the η_{MAX}, η_{MEX} minimise with enhancing N. The η_{MEX} and P_{MEX} increase with increasing λ and ρ. On the other hand, the $EFEXPEC_{MAX}$ decreases with increment of them. Enhancing r provides increment in the λ, α and reduction in the β and ζ at the maximum $EFEXPEC$. The β_{MEX}, λ_{MEX}, ζ_{MEX} and α_{MEX} enhance with enhancing speed. The β_{MEX} and ζ_{MEX} minimise with increasing λ and ρ. The results can be utilised by engine researchers to develop more eco-friendly engines.

References

Andresen, B. (1983) *Finite-Time Thermodynamics*, Physics Laboratory, University of Copenhagen.

Bejan, A. (1996) 'Entropy generation on minimization: the new thermodynamics of finite-size device and finite-time processes', *Journal of Applied Physics*, Vol. 79, No. 3, pp.1191–1218.

Caglayan, H. and Caliskan, H. (2019) 'Thermo-ecological analysis of industrial kilns', *Journal of Environmental Management*, Vol. 241, No. 1, pp.149–155.

Chen, L. and Xia, S. (2017) *Generalized Thermodynamic Dynamic-Optimization for Irreversible Cycles – Thermodynamic and Chemical Theoretical Cycles*, Science Press, Beijing.

Chen, L., Wu, C. and Sun, F. (1999) 'Finite time thermodynamic optimization or entropy generation minimization of energy systems', *Journal of Non-Equilibrium Thermodynamics*, Vol. 24, No. 4, pp.327–359.

Ebrahimi, R. (2011) 'Themodynamic modeling of performance of a miller cycle with engine speed and variable specific heat ratio of working fluid', *Computers and Mathematics with Applications*, Vol. 62, No. 1, pp.2169–2176.

Ebrahimi, R. (2012) 'Performance analysis of an irreversible miller cycle with considerations of relative air-fuel ratio and stroke length', *Applied Math Modeling*, Vol. 36, No. 1, pp.4073–4079.

EES Academic Professional Edition (2022) *V.12.305-3D*, F-Chart Software, USA.

Fawal, S. and Kodal, A. (2021) 'Overall and component basis performance evaluations for turbojet engines under various optimal operating conditions', *Aerospace Science and Technology*, Vol. 117, No. 1, p.106943.

Ferguson, C.R. (1986) *Internal Combustion Engines – Applied Thermosciences*, John Wiley & Sons Inc., New York.

Gonca, G. (2017) 'Thermo-ecological analysis of irreversible dual-miller cycle (DMC) Engine based on the ecological coefficient of performance (ECOP) criterion', *Iran J. Sci. Technol. Trans. Mech. Eng.*, in press, DOI: 10.1007/s40997-016-0060-2.

Gonca, G. and Sahin, B. (2019) 'Performance analysis of a novel eco-friendly internal combustion engine cycle', *International Journal of Energy* Research, Vol. 43, No. 1, pp.5897–5911.

Gonca, G., Sahin, B. and Cakir, M. (2020) 'Performance assessment of a modified power generating cycle based on effective ecological power density and performance coefficient', *International Journal of Exergy (IJEX)*, Vol. 33, No. 2.

Gonca, G., Sahin, B. and Genç, L. (2022) 'Investigation of maximum performance characteristics of seven-process cycle engine', *International Journal of Exergy*, Vol. 37, No. 3, pp.302–312.

Gonca, G., Sahin, B. and Ust, Y. (2013) 'Performance maps for an air-standard irreversible dual-miller cycle (DMC) with late inlet valve closing (LIVC) version', *Energy*, Vol. 5, No. 1, pp.285–290.

Gonca, G., Sahin, B., Ust, Y., Parlak, A. and Safa, A. (2015a) 'Comparison of steam injected diesel engine and miller cycled diesel engine by using two zone combustion model', *J. Energy Inst.*, Vol. 88, No. 1, pp.43–52.

Gonca, G., Sahin, B. and Ust, Y. (2015b) 'Investigation of heat transfer influences on performance of air-standard irreversible dual-miller cycle', *J. Thermophys. Heat Trans.*, Vol. 29, No. 4, pp.678–683.

Gonca, G., Sahin, B., Parlak, A., Ust, Y., Ayhan, V., Cesur, I. and Boru, B. (2015c) 'Theoretical and experimental investigation of the miller cycle diesel engine in terms of performance and emission parameters', *Appl. Energy*, Vol. 138, No. 1, pp.11–20.

Gonca, G., Sahin, B., Parlak, A., Ayhan, V., Cesur, I. and Koksal, S. (2015d) 'Application of the miller cycle and turbo charging into a diesel engine to improve performance and decrease NO emissions', *Energy*, Vol. 93, No. 1, pp.795–800.

Gonca, G., Sahin, B., Parlak, A., Ayhan, V., Cesur, I. and Koksal, S. (2017) 'Investigation of the effects of the steam injection method (SIM) on the performance and emission formation of a turbocharged and miller cycle diesel engine (MCDE)', *Energy*, Vol. 119, No. 1, pp.926–937.

Hohenberg, G. (1979) *Advanced Approaches for Heat Transfer Calculations*, No. SAE 790825.

Kamiuto, K. (2006) 'Comparison of basic gas cycles under the restriction of constant heat addition', *Applied Energy*, Vol. 83, No. 1, pp.583–593.

Karakurt, A.S., Ozsani, I. and Bashan, V. (2022) 'Exergetic performance analysis of high pressure air systems on ships', *International Journal of Exergy*, Vol. 37, No. 1, pp.1–10.

Li, D., Li, S., Ma, Z., Xu, B., Lu, Z., Li, Y. and Zheng, M. (2021) 'Ecological performance optimization of a high temperature proton exchange membrane fuel cell', *Mathematics*, Vol. 9, No. 1, p.1332.

Mikalsen, R., Wang, Y.D. and Roskilly, A.P. (2009) 'A comparison of miller and Otto cycle natural gas engines for small scale CHP applications', *Applied Energy*, Vol. 86, No. 1, pp.922–927.

Patodi, K. and Maheshwari, G. (2013) 'Performance analysis of an Atkinson cycle with variable specific heats of the working fluid under maximum efficient power conditions', *International Journal of Low-Carbon Technologies*, Vol. 8, No. 4, pp.289–294.