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# EFFECT OF MEAN PISTON SPEED AND RESIDUAL GAS FRACTION ON PERFORMANCE OF A FOUR-STROKE IRREVERSIBLE OTTO CYCLE ENGINE

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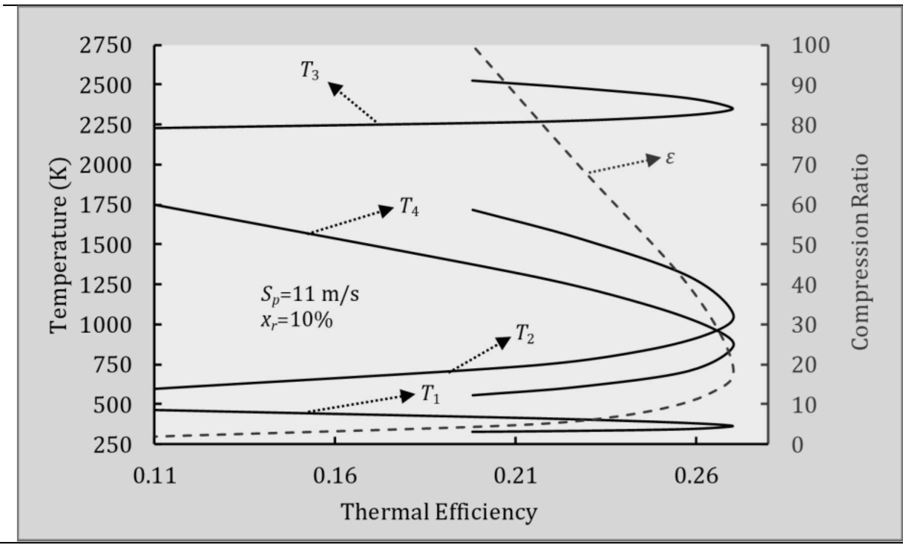
## HIGHLIGHTS

- The irreversible Otto cycle was modeled with the finite-time thermodynamics.
- The performance of the irreversible Otto cycle was investigated in terms of mean piston speed and residual gas fraction.
- The results obtained are reference of the engine designers.

## GRAPHICAL ABSTRACT

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## ABSTRACT

*In this study, a four-stroke irreversible Otto cycle model was constructed using the finite time thermodynamics to investigate the effect of the mean piston speed and the residual gas fraction on the cycle (or engine) performance. Fuel consumption was taken as a function of mean piston speed, and initial cycle temperature was considered as a function of residual gas fraction. It has been assumed that the specific heat does not change depending on the temperature. A detailed numerical example study has been made to see the effect of the mean piston speed and the residual gas fraction on engine performance. As a result of the numerical example made, the cycle thermal efficiency and the dimensionless power output were observed with the increase of the residual gas amount and the mean piston speed. We think that the results obtained are especially important for engine designers.*

## 1. INTRODUCTION

Otto cycle heat engines are widely used in many areas, from industrial to transportation, and are the main power source for many modern machines such as automobiles, agricultural machines, transport vehicles and generators [1]. By using thermodynamics to analyze the performance of the Otto cycle heat engine, the rules for the development

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and use of novel technologies can be determined or can be developed to optimize the Otto cycle engine. Using classical thermodynamics to conduct the first law analysis for the Otto cycle, one can examine the quantitative relationship between efficiency and different losses [2]. Using the second law of thermodynamics to analyze the performance of the Otto cycle, loss of work capacity due to various irreversible losses in the energy conversion process can be investigated [3]. In order to analyze the performance of the Otto cycle it is possible to obtain more realistic results with the finite-time thermodynamics taking into account the irreversibilities in addition to the first law of thermodynamics. [4].

As a novel branch of modern thermodynamic theory, finite time thermodynamics (FTT) has made great progress in recent years. FTT has also been applied to performance analysis, optimization and optimal configuration studies for internal combustion engine cycles [5].

In modern thermodynamic theories it is important to use FTT to analyze and optimize the performance of real thermodynamic cycles and processes. The fundamental problem of FTT can be divided into two cases: research on optimum performance for a given thermodynamic process and on optimal configuration for a thermodynamic process when optimization objectives are given [6-9].

The researches on optimal performances of Otto cycle include the following three aspects: the optimal performance with con-stant specific heats; the optimal performance with variable specific heats; the optimal performance with variable specific heat ratio [5].

There are many studies in the literature about the analysis of the otto cycle with FTT. Wu et al. investigated the effect of combustion on irreversible Otto cycle performance and derived the relationship between maximum work output and optimum compression ratio at maximum operating temperature [10-12]. Chen et al., irreversible Otto cycle, relate the cycle between work and utility, and the relationship between the initial temperature and heat transfer is derived [13]. Hou evaluated irreversible Otto cycle and Atkinson cycles in terms of maximum work output and thermal efficiency [14]. Ge et al. have investigated the effects of specific heat changes on irreversible Otto cycle performance [1]. Ge et al. have built a model for the irreversible Otto cycle with air standard acceptance of internal irreversibilities, frictional losses and heat transfer losses accounted for, and analyzed cycle performance with the entropy generation rate and ecological function parameters for the cycle losses [15]. Ge et al. have created an exergy-based model for the ecological performance of the air standard irreversible Otto cycle [16].

It is possible to find many studies in the literature besides these. Air-standard or air-fuel assumtion can be done with FTT models [17, 18]. However, it is possible that the specific heats do not change with temperature [11,19], or that the specific heat changes linearly, logarithmically, exponentially or polynomally [20-22]. Numerical examples are also included in the studies. In air-fuel assumtion models, it is also possible to examine concepts related to combustion such as residual gas fraction, equivalence ratio and combustion efficiency. In addition, ecological performance analysis can be done using the second law concepts. For heat transfer and friction losses, various models in the literature can be used [5-27].

In this study, FTT model was constructed to investigate the effect of residual gas fraction and mean piston speed on the cycle performance by using constant specific heat in consideration of heat transfer, friction and irreversibilities in Otto cycle of air-fuel assumption and these effects were investigated with a numerical example. In addition, residual gas temperature is included in the calculation for the initial cycle temperature.

## 2. METHODS

The pressure-volume ($P$-$v$) and the temperature-entropy ($T$-$s$) diagrams of an irreversible four-stroke Otto engine cycle is shown in Fig. 1, where the "$s$" subscripts denote reversible processes for compression ($1 \rightarrow 2s$) and expansion ($3 \rightarrow 4s$). $1 \rightarrow 2$ and $3 \rightarrow 4$ are processes in which internal irreversibilities are accounted for. The heat addition ($2 \rightarrow 3$) and the heat rejection ($4 \rightarrow 1$) processes are isochoric processes [23].

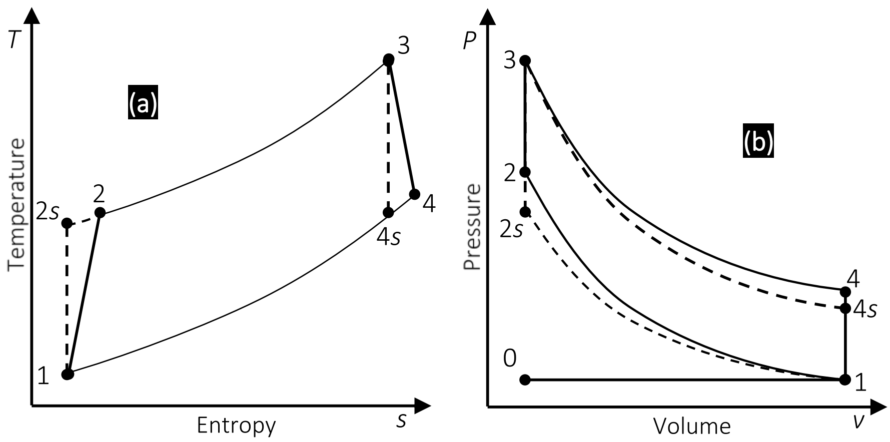


**Fig. 1.** *Otto cycle (a) Ts diagram, (b) Pv diagram*

The relations between the fuel flow ($\dot{m}_f$) and mean piston speed ($S_p$) [8], between $\dot{m}_f$ and the fuel-air-residual gas (mixture) flow ($\dot{m}_t$) [23] are defined as:

$$\dot{m}_f = 0.0003\bar{S}_p \tag{1}$$

$$\dot{m}_t = \dot{m}_f + \dot{m}_a + \dot{m}_r = \frac{\dot{m}_f(\phi + x_{AFs})}{\phi(1 - x_r)} \tag{2}$$

where $x_{AFs}$, $\dot{m}_a$, $\phi$ and $x_r$, are the air-fuel ratio for stoichiometric conditions, air flow, equivalance ratio and the residual gas fraction from previous cycle, respectively. Residual gas fraction can find it as [23]:

$$x_r = \frac{\dot{m}_r}{\dot{m}_t} \tag{3}$$

where $\dot{m}_r$ is residual gases flow. It should be noted here that the residual gases were assumed to consist of $CO_2$, $H_2O$ and $N_2$.

The specific heat at constant volume ($c_{vt}$) and the gas constant ($R_t$) for the working fluid are defined as:

$$c_{vt} = \frac{\begin{Bmatrix} (1 - x_r)(x_{AFs}c_{va} + \phi c_{vf}) \\ +x_r(\phi + x_{AFs})c_{vr} \end{Bmatrix}}{(\phi + x_{AFs})} \tag{4}$$

$$R_t = \frac{\begin{Bmatrix} (1 - x_r)(x_{AFs}R_a + \phi R_f) \\ +x_r(\phi + x_{AFs})R_r \end{Bmatrix}}{(\phi + x_{AFs})} \tag{5}$$

where $c_{va}$, $c_{vf}$ and $c_{vr}$ are the mass specific heat at constant volume for air, fuel and residual gases, and $R_a$ $R_f$,and $R_r$ are the gas constant for air, fuel and residual gases, respectively.

For the two adiabatic processes, the compression and expansion efficiencies [24]:

$$\eta_c = (T_{2s} - T_1)/(T_2 - T_1) \tag{6}$$

$$\eta_e = (T_4 - T_3)/(T_{4s} - T_3) \tag{7}$$

These two efficiencies are used to describe the internal irreversibility of the processes. The equations for reversible adiabatic processes 1→2s and 3→4s are as follows [21]:

$$c_{vt}\ln(T_{2s}/T_1) = R_t\ln(\varepsilon) \tag{8}$$

$$c_{vt}\ln(T_{4s}/T_3) = -R_t\ln(\varepsilon) \tag{9}$$

The added heat flow per second in the isochoric (2→3) heat addition process may be written as [26]:

$$\dot{Q}_{in} = \dot{m}_t c_{vt}(T_3 - T_2) \tag{10}$$

The rejected heat flow per second in the isochoric (4→1) heat rejection process may be written as [22]:

$$\dot{Q}_{out} = \dot{m}_t c_{vt}(T_4 - T_1) \tag{11}$$

The total energy flow of the fuel may be written as [23]:

$$\dot{Q}_f = \eta_{com}\dot{m}_f H_u \tag{12}$$

where $\eta_{com}$ is the combustion efficiency and can be accepted as 0.87 [9]. $H_u$ is the lower heat value of the fuel.

Heat transfer losses for an ideal Otto cycle model can be negligible, but for a real Otto cycle, the heat transfer irreversibility between the working fluid and the cylinder wall is so important that it can not be negligible. The heat loss on the cylinder wall can be expressed by a simple linear equation.

$$\dot{Q}_{ht} = \dot{m}_t\beta(1 - x_r)(T_2 + T_3 - 2T_w) \tag{13}$$

where $\beta$ is the heat loss coefficient of the cylinder wall.

The relatioship between $\dot{Q}_f$, $\dot{Q}_{in}$ and$\dot{Q}_{ht}$ can be defined as follows [24]:

$$\dot{Q}_{in} = \dot{Q}_f - \dot{Q}_{ht} \tag{14}$$

In thermodynamic analyzes $T_1$ is considered a constant value as the initial cycle temperature. The $T_1$ is defined as a specific function because the residual gas fraction is now included in the calculation. It is assumed that $T_1$ changes according to reference temperature ($T_0$), $T_4$ and $x_r$. $T_1$ may be written as:

$$T_1 = T_0 + \frac{x_r R_r(T_4 - T_0)}{R_t} \tag{15}$$

The friction loss can be expressed as a function of $S_p$ [24]:

$$P_\mu = -\mu(S_p)^2 \tag{16}$$

where $\mu$ is the coefficient of friction, which takes into account the global losses.

The effective power output ($P_e$) of the Otto cycle engine is expressed by:

$$P_e = \dot{Q}_{in} - \dot{Q}_{out} - |P_\mu| \tag{17}$$

In this study, fuel consumption is expressed as a function of meanpiston speed. For this reason, the amount of energy input depends on the mean piston speed. To make a fair comparison, the power output is dimensionless according to the initial conditions. The dimensionless power output ($\psi$) is expressed as follows:

$$\psi = \frac{P_e}{\dot{m}_t R_t T_1} \tag{18}$$

The efficiency of the cycle is expressed by [24]:

$$\eta_{th} = P_e/\dot{Q}_f \tag{19}$$

When $S_p$, $c_{va}$, $c_{vf}$, $c_{vr}$, $R_a$, $R_f$, $R_r$, $\beta$, $T_w$, $T_0$, $\mu$, $x_r$, $x_{AFs}$ and $\phi$ are given, temperatures of the critical points ($T_1$, $T_2$, $T_{2s}$, $T_3$, $T_4$, and $T_{4s}$) can be obtained from Eqs. (7-16). Using Eqs. (17-19), power output, dimensionless power output and thermal efficiency can be determined, respectively.

## 3. RESULTS AND DISCUSSION

The following constants and parameters are used for the numerical example (Table 1):

**Table 1.** *Parameters used for numerical example* [5-27]

| Parameter | Value | Parameter | Value |
|---|---|---|---|
| $\eta_c$ | 0.97 | $\eta_e$ | 0.97 |
| $x_{AFs}$ | 15.05 | $T_0$ | 300 K |
| $\beta$ | 0.5 kJ/kgK | $T_w$ | 400 K |
| $\mu$ | 12.9 Ns/m | $H_u$ | 44790 kJ/kg |
| $c_{va}$ | 0.718 kJ/kgK | $R_a$ | 0.287 kJ/kgK |
| $c_{vf}$ | 1.638 kJ/kgK | $R_f$ | 0.073 kJ/kgK |
| $c_{vr}$ | 0.866 kJ/kgK | $R_r$ | 0.307 kJ/kg |
| $\phi$ | 1.00 | $x_r$ | 5%→15% |
| $\varepsilon$ | 2→100 | $S_p$ | 7→15 m/s |

This study focuses on the investigation of the effects of mean piston speed and residual gas fraction on performance of a four-stroke irreversible Otto cycle engine. The performance parameters obtained for both engines are compared.

The variation of critical point temperatures with respect to thermal efficiency is given in Fig 2. In addition, the dashed line shows the relationship between thermal efficiency and compression ratio.

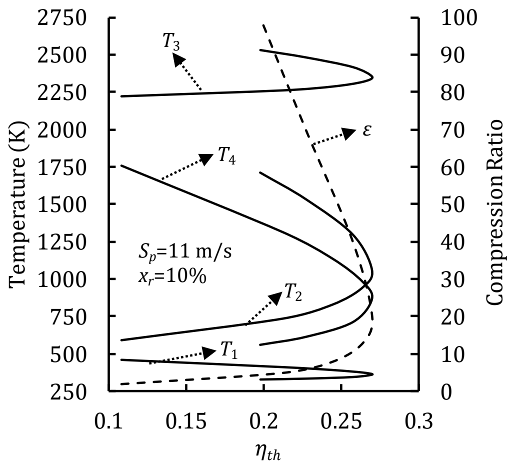


**Fig. 2.** *The temperature versus thermal efficiency*

In Fig. 2, the thermal efficiency increases to a certain compression ratio and the thermal efficiency begins to decrease at higher compression ratios. It is seen that $T_1$ decreased with the increase of $\eta_{th}$. In the case where the $\eta_{th}$ is maximum, $T_1$ is almost minimum. $T_2$ is the optimum value at the point where the $\eta_{th}$ is maximum. Increasing $T_2$ after this value increases the compression ratio. When $T_3$ is examined, it has a similar characteristic depending on $T_2$. The continuation of increase of $T_3$ after the maximum $\eta_{th}$ is due to the increase of the compression ratio as in $T_2$. The same is true for the $T_4$.

Figs. (3-6) show the effect of the mean piston speed on the cycle performance. Figs. (7-10) show the effect of the residual gas fraction on the cycle performance.

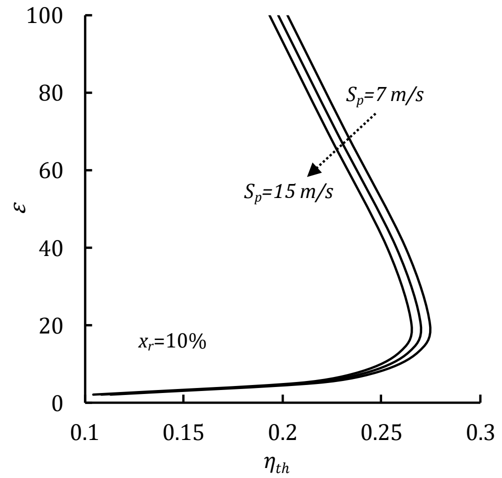


**Fig. 3.** *Effect of mean piston speed on the variation of the compression ratio with thermal efficiency*

Fig. 3 shows the change in compression ratio due to thermal efficiency. As the mean piston speed increases, the maximum thermal efficiency decreases. While the thermal efficiency increases to a certain compression ratio, lower thermal efficiency is obtained at higher compression ratios. However, as the mean piston speed increases, the same thermal efficiency can be achieved at lower compression ratios. Although higher maximum thermal efficiency can be achieved at low mean piston speed, it is more advantageous that the mean piston speed is high at the compression ratios after the maximum thermal efficiency.

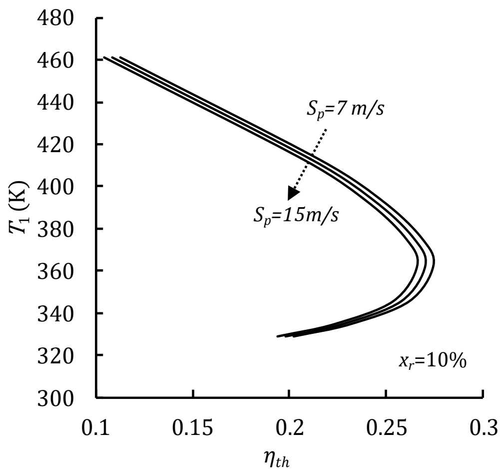


**Fig. 4.** *Effect of mean piston speed on the variation of the initial temperature with thermal efficiency*

Fig. 4 shows the change in the initial temperature $T_1$ with respect to the thermal efficiency. As the mean piston speed increases $T_1$ decreases. In addition, the higher mean piston speed, the higher thermal efficiency is achieved. The increase in $T_1$ after reaching maximum thermal efficiency is due to the increased compression ratio.

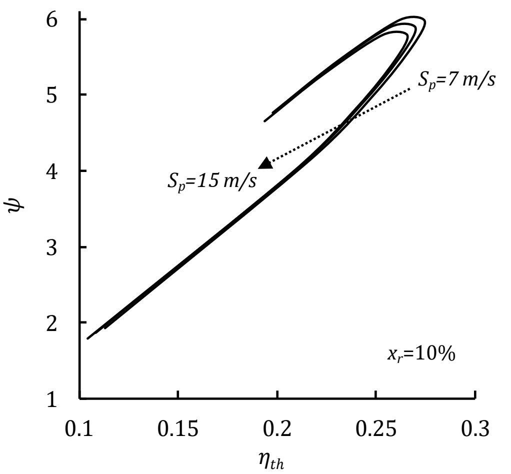


**Fig. 5.** *Effect of mean piston speed on the variation of the dimensionless power output with thermal efficiency*

Fig. 5 shows the variation of the dimensionless power output with respect to the thermal efficiency. As the mean piston speed increases, less thermal efficiency is obtained and less dimensionless power output is obtained. Approximate maximum dimensionless power output is obtained at the point where maximum thermal efficiency is obtained. In fact, the maximum dimensionless power output is obtained at a compression ratio slightly higher than the compression ratio at which the maximum thermal efficiency is obtained.

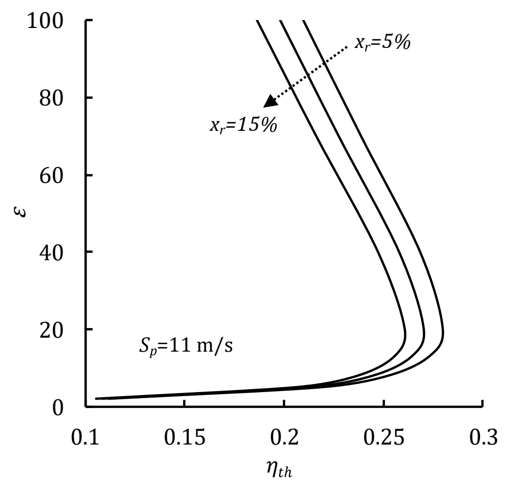


**Fig. 6.** *Effect of residual gas fraction on the variation of the compression ratio with thermal efficiency*

Figure 6 shows the change in the compression ratio with respect to the thermal efficiency. As the residual gas fraction increases, the maximum thermal efficiency decreases.

Fig. 7 shows the change in $T_1$ depending on the thermal efficiency. As the residual gas fraction increases, the maximum thermal efficiency decreases and $T_1$ increases. The reason for the increase in $T_1$ with the increase of the residual gas fraction is that the temperature of the residual gases is higher than the fresh mixture temperature. In many models in the literature, $T_1$ is considered as a constant value. In this study, $T_1$ is modeled as a function of $T_4$ and residual gas fraction.

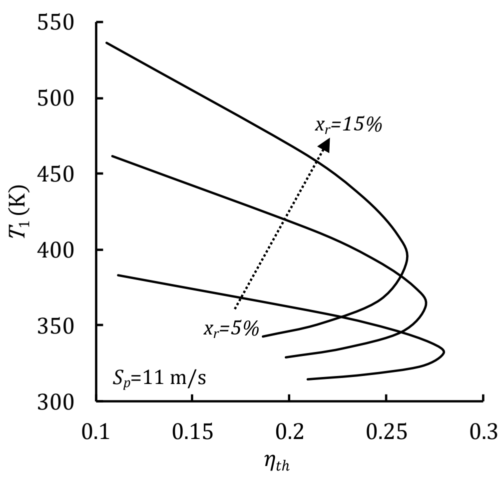


**Fig. 7.** *Effect of residual gas fraction on the variation of the initial temperature with thermal efficiency*

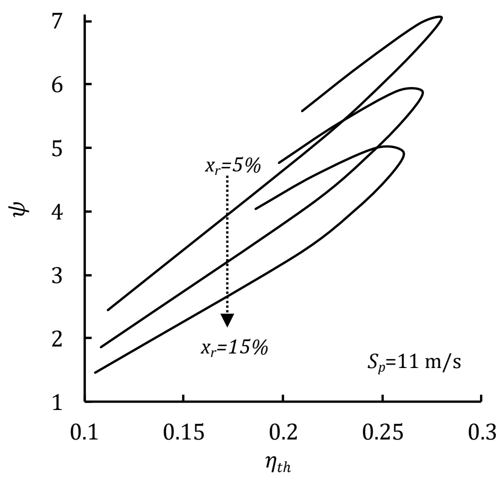


**Fig. 8.** *Effect of residual gas fraction on the variation of the dimensionless power output with thermal efficiency*

Fig. 8 shows the change in dimensionless power output as a function of heat transfer. The residual gas fraction is significantly reduced and the dimensionless power output is reduced. Therefore, the dimensionless power output is higher in the low residual gas quantity.

## 4. CONCLUSION

In this study, four-stroke irreversible Otto cycle are modeled using the finite time thermodynamics (FTT). The effects of the mean piston speed and residual gas fraction on the engine performance of the FTT model are investigated with a numerical example and the results are discussed. In the literature, there has been no study of the effect of residual gas fraction and mean piston speed on the irreversible Otto cycle. However, a function has been derived for the initial temperature using the end-of-cycle temperature and the residual gas fraction.. As a result of this study, the following general results were obtained:

- The thermal efficiency increases up to a certain compression ratio and the thermal efficiency decreases after this compression ratio.
- If the thermal efficiency is maximum, the initial cycle temperature is optimum. Then the initial cycle temperature is also increased by the effect of the compression ratio.
- As the mean piston speed increases, the maximum thermal efficiency that can be obtained from the cycle decreases.
- As the residual gas fraction increases, the initial cycle temperature increases.
- As the residual gas fraction increases, the maximum thermal efficiency that can be obtained from the cycle decreases.
- Dimensionless power output is higher at low piston speed and low residual gas fraction.

With this study, the results that engine designers can benefit from have been obtained. In addition, since the model of the FTT model is used instead of the classical thermodynamic model, the theoretical cycle is further approximated to the actual cycle, and the effect of the engine performance on the independent parameters such as mean piston speed and residual gas fraction can be examined.

## NOMENCLATURES

| | | |
|---|---|---|
| $c_{va}$ | [kJ/kgK] | Specific heat for air |
| $c_{vf}$ | [kJ/kgK] | Specific heat for fuel |
| $c_{vr}$ | [kJ/kgK] | Specific heat for residual gas |
| $c_{vt}$ | [kJ/kgK] | Specific heat for mixture |
| $H_u$ | [kJ/kg] | Lower heat value for fuel |
| $\dot{m}_a$ | [kg/s] | Air flow |
| $\dot{m}_f$ | [kg/s] | Fuel flow |
| $\dot{m}_r$ | [kg/s] | Residual gas flow |
| $\dot{m}_t$ | [kg/s] | Mixture flow |
| $P_e$ | [kW] | Effective Power output |
| $P_\mu$ | [kW] | Global friction loss power |
| $\dot{Q}_{ht}$ | [kJ/s] | Heat leakage per second |
| $\dot{Q}_f$ | [kJ/s] | Fuel energy flow |
| $\dot{Q}_{in}$ | [kJ/s] | Added heat flow |
| $\dot{Q}_{out}$ | [kJ/s] | Rejected heat flow |
| $R_a$ | [kJ/kgK] | Gas constant for air |
| $R_f$ | [kJ/kgK] | Gas constant for fuel |
| $R_r$ | [kJ/kgK] | Gas constant for residual gas |
| $R_t$ | [kJ/kgK] | Gas constant for mixture |
| $S_p$ | [m/s] | Mean piston velocity |
| $T_0$ | [kJ/kgK] | Reference temperature |
| $T_w$ | [kJ/kgK] | Wall temperature |
| $x_{AFs}$ | [-] | Stoichiometric air-fuel ratio |
| $x_r$ | [-] | Residual gas fraction |
| $\eta_c$ | [-] | Compression efficiency |
| $\eta_{com}$ | [-] | Combustion efficiency |
| $\eta_e$ | [-] | Expansion efficiency |
| $\eta_{th}$ | [-] | Thermal efficiency |
| $\Phi$ | [-] | Equivalence ratio |
| $\beta$ | [kJ/kgK] | Heat leakage coefficient |
| $\varepsilon$ | [-] | Compression ratio |
| $\mu$ | [Ns/m] | Coefficient of friction |
| $\psi$ | [-] | Dimensionless power output |

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