THERMODYNAMIC MODELING AND PARAMETRIC STUDY FOR POROUS MEDIUM ENGINE CYCLES

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1. INTRODUCTION

A series of achievements have been made since finite-time thermodynamics was used to analyze and optimize the performance of real heat engines [1-14]. In recent thirty years, the study for the internal combustion engine cycles also has yielded some achievements, and these achievements concentrated on two aspects mainly. The first is to determine the optimal path for the given optimization objective and a series of constraints. The second is to determine the optimal objective for the given path and a series of constraints with different losses.

The superadiabatic engine based on the porous-medium combustion technology can decrease the emissions and has tremendous potential in improving the efficiency. The characteristic such as stable combustion, the compact in structure and a wide range of load regulation have aroused general concern [15-18]. Liu et al [19] modeled the ideal PM cycle and derived the performance relations by classical thermodynamics. Based on Ref. [19], this paper will establish a finite time thermodynamic model of PM cycle with heat transfer loss and friction loss, and analyze and optimize the cycle performance.

2. CYCLE MODEL

A model of porous medium engine cycles is shown in figure 1. The compression process is an isentropic process 1 → 2; the regenerative process is an isochoric process 2 → 3; the heat addition process is an isothermal process 3 → 3'; the expansion process is an isentropic process 3' → 4, and the heat rejection is an isochoric process 4 → 1.

![Fig. 1. T-s diagram for PM cycle.](image)

Considering the heat engine can operate N cycles in a second, thus the heat added to the working fluid in a second is [20]

\[ Q_{in} = N M (C_v (T_3 - T_2) + RT \ln \frac{\nu'_3}{\nu_1}) \]  

(1)
The heat rejected by the working fluid is
\[ Q_{\text{out}} = NM C_v (T_4 - T_1) \] (2)
where \( M \) is the mass per cycle of the working fluid.

For an ideal PM cycle model, there are no irreversible losses. However, for a real PM cycle, heat transfer loss between working fluid and the cylinder wall and friction loss are not negligible. One can assume that the heat loss through the cylinder wall is proportional to average temperature of both the working fluid and the cylinder wall and that the wall temperature is constant \( T_w \). If the released heat by combustion for one kilogram working fluid is \( A_k \), the heat leakage coefficient of the cylinder wall is \( B_k \), one has the heat added to the working fluid by combustion in the following linear relation \[ Q_{\text{in}} = NM\{A_1 - B_1[(T_2 + T_3)/2 - T_0]\} = N M (\alpha - \beta(T_2 + T_3)) \] (3)
where \( \alpha = A_1 + B_1 T_0 \) and \( \beta = B_1/2 \) are two constants related to combustion and heat transfer.

The compression ratio and the volume ratio are defined as
\[ \gamma = v_1/v_2 \quad \rho = v_3'/v_3 \] (4)
Therefore
\[ T_2 = T_1 \gamma \rho^\kappa \quad T_3 = T_3 \rho/\rho^\kappa \] (5)
where \( \kappa = C_p/C_v \) is the ratio of specific heats. When \( \rho = 1 \), PM cycle can become Otto cycle [22-24].

In order to make the cycle operate normally, state 3 must be between states 2 and 3'. When \( T_2 = T_3 \), the upper limit of the volume ratio can be obtained
\[ \rho_{\text{max}} = \frac{\alpha - 2\beta T_1 \gamma \rho^\kappa}{T_2 \rho^\kappa} \] (6)
Therefore, the range of the volume ratio is \( 1 < \rho < \rho_{\text{max}} \).

Combining equations (1) with (3) gives
\[ T_3 = \frac{\alpha - (\beta - C_p)T_1 \gamma \rho^\kappa}{C_v + R \ln \rho + \beta} \] (7)
Taking into account the friction loss of the piston as recommended by Angulo-Brown et al [24] for Otto cycle and assuming a dissipation term represented by a friction force which in a liner function of the velocity gives
\[ f_\mu = \mu \nu = \mu \frac{dx}{dt} \] (8)
where \( \mu \) is a coefficient of friction which takes into account the global losses and \( x \) is the piston displacement. Then, the lost power is
\[ P_\mu = \frac{dW_\mu}{dt} = \mu \frac{dx \cdot dx}{dt} = \mu \nu^2 \] (9)
If one specifies the engine is a four stroke cycle engine, the total distance the piston travels per cycle is
\[ 4L = 4(x_1 - x_2) \] (10)
where \( x_1 \) and \( x_2 \) are the piston position at maximum and minimum volume, respectively.

For a four stroke cycle engine, running at \( N \) cycles per second, the mean velocity of the piston is
\[ \bar{v} = 4LN \] (11)

Thus, the power output of the cycle is
\[ P_{\text{pm}} = Q_{\text{in}} - Q_{\text{out}} - P_\mu = N M [C_v (T_3 + T_1 - T_2 - T_3) + R \ln \rho - 16 \nu (LN)] \] (12)

The efficiency of the cycle is
\[ \eta_{\text{pm}} = \frac{P_{\text{pm}}}{Q_{\text{in}}} = \frac{N M [C_v (T_3 + T_1 - T_2 - T_3) + R \ln \rho - 16 \nu (LN)]^2}{N M [C_v (T_3 - T_2) + R \ln \rho]} \] (13)

3. NUMERICAL EXAMPLES AND DISCUSSION

The following parameters are used in the calculations:
\[ \alpha = 2400 - 2600kJ/kg^\circ, \beta = 1.0 - 1.4kJ/kg^\circ \cdot K, \gamma = 12.9 - 16.9N/m, \rho = 1.0 - 1.6T_1 = 350K, \gamma_p = 6.2, \eta_\mu = 0.36, \eta_k = 0.42, \eta_k = 1.15 \text{ and } P_0 = 2.25kW, \]
Using the above constants and range of parameters, the characteristic curves of \( P_{\text{pm}} - \eta_{\text{pm}} \) can be plotted as figures 2-13. Form figures 2-3, when \( \alpha = 250kJ/kg^\circ, \beta = 1.0kJ/kg^\circ \cdot K, \gamma = 12.9N/m, \rho = 1.6 \text{ and } T_1 = 350K \), one can see that the maximum power output and corresponding compression ratio and efficiency are
\[ P_{\text{max}} = 2.75kW, \gamma_p = 6.2 \text{ and } \eta_p = 0.36 \text{, respectively,} \]
the maximum efficiency and corresponding compression ratio and power output are
\[ \eta_{\text{max}} = 0.42, \gamma_\mu = 11.5 \text{ and } P_0 = 2.25kW, \text{ respectively.} \]
The compression ratios at the maximum power output and the maximum efficiency are different, so the cycle can operate under the maximum power output and the maximum efficiency conditions. The power output versus efficiency curve is loop-shaped one in figure 4, it reflects the performance characteristics of a real engine cycles.

\( \alpha \) and \( \beta \) are two constants related to combustion and heat transfer. Where \( \alpha \) reflects the magnitude of heating value of fuel, the bigger \( \alpha \) is, the higher heating value is, and the less of the heat transfer loss will be. While \( \beta \) reflects the magnitude of heat transfer loss, the bigger \( \beta \) is, the more heat transfer loss is. Figures 2-7 show the effects of the heating value and the heat transfer loss on the cycle performance. One can see that the power output, the efficiency, the efficiency at the maximum power output, as well as the power output at the maximum efficiency of the cycle will decrease with the increase of the heat transfer loss.

Figures 8-10 show the effects of friction loss on the cycle performance. One can see that when \( \mu \) increases, the maximum power output and corresponding compression ratio and efficiency, the maximum efficiency and corresponding compression ratio and power output will decrease.
Fig. 2. The influences of $\alpha$ on the cycle power output.

Fig. 3. The influences of $\alpha$ on the cycle efficiency.

Fig. 4. The influences of $\alpha$ on the power output versus efficiency characteristic.
Fig. 5. The influences of $\beta$ on the cycle power output.

$\rho = 1.6$
$T_i = 350K$
$\alpha = 2500kJ/kg$
$\mu = 12.9N \cdot s/m$

Fig. 6. The influences of $\beta$ on the cycle efficiency.

Fig. 7. The influences of $\beta$ on the power output versus efficiency characteristic.
Fig. 8. The influences of $\mu$ on the cycle power output.

Fig. 9. The influences of $\mu$ on the cycle efficiency.

Fig. 10. The influences of $\mu$ on the power output versus efficiency characteristic.
Figures 11-13 show the effects of the volume ratio $\rho$ on the cycle performance. When $\rho = 1$ the curve in the figure is the performance characteristics of an irreversible Otto cycle [23]. One can see that the power output, the efficiency and the efficiency at the maximum power output of the cycle will increase with the increase of $\rho$. Furthermore, the performance of PM engine cycle is superior to that of Otto cycle.
4. CONCLUSION

In this paper, the PM engine cycle model with the considerations of the heat transfer loss and friction like-term loss was presented. The performance characteristic of the cycle was analyzed. The characteristic relation of the power output and efficiency was derived and the effects of the parameters on the performance were analyzed.

The results show that the power output, the efficiency, the efficiency at the maximum power output, as well as the power output at the maximum efficiency of the cycle decrease with the increases of the heat transfer loss and friction loss and the decrease of $\rho$. Furthermore, the performance of the PM cycle is superior to that of the Otto cycle.

This paper is a theoretical study about porous medium heat engine, so many of parameters varied are not really tunable in actual engineering practice. And in our following study, we will optimize the porous medium heat engine based on the experimental datum.

Nomenclature

- $C_v$ – specific heat with constant volume ($kJ/kg\cdot K$)
- $\kappa$ – ratio of specific heats
- $L$ – total distance of the piston traveling per cycle ($m$)
- $M$ – mass per cycle of the working fluid ($kg$)
- $N$ – number of the cycle operating in a second
- $P_{out}$ – power output of the cycle ($kW$)
- $P_L$ – lost power due to friction ($kW$)
- $Q_{in}$ – heat added to the working fluid in a second ($kJ$)
- $Q_{out}$ – heat rejected by the working fluid in a second ($kJ$)
- $R$ – air constant of the working fluid ($kJ/kg\cdot K$)
- $T$ – temperature at different states ($K$)
- $V$ – volume at different states ($m^3$)
- $X_{p}$ – the piston position at maximum volume ($m$)
- $X_{g}$ – the piston position at minimum volume ($m$)

Greek symbols

- $\alpha$ – constants related to combustion ($kJ/kg\theta$)
- $\beta$ – constants related to heat transfer ($kJ/kg\cdot K$)
- $\eta$ – efficiency
- $\mu$ – coefficient of friction ($N\cdot s/m$)
- $\gamma$ – compression ratio
- $\rho$ – volume ratio

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