Power output and Efficiency of internal combustion engine based on the FTT theory

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Abstract: Typically, in an internal combustion engine, thousands of cycles are performed in a minute. In this sequence of cycles many physical and chemical quantities change from cycle to cycle. For example, the combustion heat changes due to residual gases, imperfect combustion and other reasons. In this work, we present two finite-time thermodynamics models for both an Otto and a Diesel cycle, in which the cyclic variability is studied as occurring in the heat capacities of the working fluid. The fluctuations considered are of the uncorrelated type (uniform and gaussian) and one correlated case (logistic map distribution). We find that in the correlated fluctuations case, the power output and the efficiency of both cycles reach bigger fluctuations than in the uncorrelated cases. This result can provide insights over the performance of internal combustion engines.

Keywords: internal combustion engine; cyclic variability; fluctuations

1. Introduction

In 1996, Badescu and Andresen proposed that finite-time thermodynamics (FTT) can be complemented with some probabilistic concepts allowing a more accurate description of the performance indicators of a power system. These authors studied a continuous flow tube reactor which supplies heat to an engine from a chemical reaction with linear kinetics. In general, typical FTT-models of thermal cycles are worked in steady state and only one cycle is taken as representative of all the other cycles pertaining to a sequence of them. In a typical internal combustion engine, several thousands of cycles are performed in a minute and there exist theoretical and experimental reasons to expect important variations from one cycle to the next. These variations can be found for example in the combustion heat of an Otto cycle. In a recent paper, Daw et al. proposed a discrete engine model that explains how both stochastic and deterministic features can be observed in spark-ignited internal combustion engines. These authors present a model which reproduces the experimental observations of the cyclic variability of the combustion heat in a four-stroke, spark-ignition Otto cycle. Recently, we have reported an irreversible Otto cycle model including chemical reactions. In that model, we analyzed the performance of an Otto engine (see Fig. 1) taking into account power losses due to a kind of lumped friction and we also consider the combustion reaction at the end of the adiabatic compression. In other work, we took the concept of fluctuant combustion heat proposed by Daw et al. as the input of our irreversible Otto cycle model, and then we analyze the behavior of performance outputs, such as the

power (P) and the efficiency (η) of the Otto cycle model.

In that work, we found that the size of the fluctuations in P and η around their mean values can be driven by the size of the combustion heat fluctuations and also by thermodynamic properties of the states of the working fluid. In the present work, we study two thermal cycles, namely the Otto and the Diesel cycles, under heat fluctuations, but using an alternative approach, where the fluctuations are taken as occurring in the heat capacity of the working fluid. In our approach, we take two previous FTT- models for both the Otto and the Diesel cycles. The paper is organized as follows: in Sec. 2, we present a brief resume of our previous thermal cycle models; in Sec. 3 we discuss our fluctuant models and finally in Sec. 4 we present the conclusions.

2. Preliminaries

In Fig. 1 we depict the Otto cycle pressure-volume diagram of the processes followed by the working fluid consisting of n moles of air. In Ref. 7 it was considered that both the "absorbed" heat Q_{eff} and the rejected heat Q_{out} occur at finite times given by

$$t_{1V} = K_1(T_3/T_2)$$

and   (1)

$$t_{2V} = K_2(T_4/T_1)$$

respectively, where K_1 and K_2 are constants linked to the mean variation rate of the temperatures. In this approach the adiabatic processes were taken as approximately instantaneous, as it is common in

FTT-models [9]. In this way, the cycle's period is given by,

$$\tau = t_{1V} + t_{2V} = K_1 (T_3 - T_2) + K_2 (T_4 - T_1). \qquad (2)$$
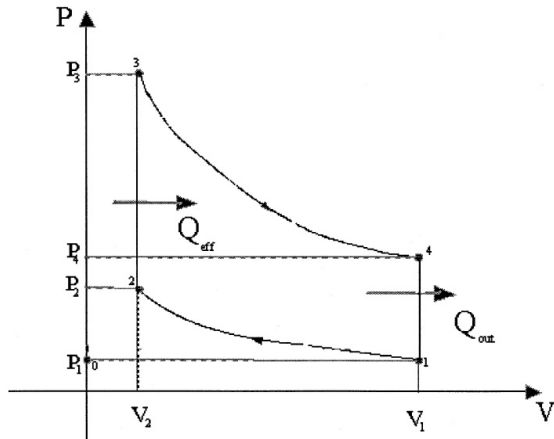

Figure 1. Pressure-volume diagram of an ideal Otto cycle.

In Ref. 7 the cycle's power output without losses was taken as

$$P_R = \frac{W_{TOT}}{\tau} = \frac{C_{v_1} - C_{v_2} r^{1-\gamma}}{K_1 - K_2 r^{1-\gamma}} \qquad (3)$$

where C_{v_1} and C_{v_2} are the constant-volume heat capacities of air in both isochoric processes $2 \rightarrow 3$ and $4 \rightarrow 1$, and $\gamma = C_{p_1}/C_{v_1} = C_{p_2}/C_{v_2}$, being C_{p_1} and C_{p_2} the constant pressure heat capacities of the working fluid during the processes $2 \rightarrow 3$ and $4 \rightarrow 1$ respectively, $r = V_1/V_2$ is the so-called compression ratio. When in the model, losses due to friction, turbulence in the working fluid, heat leaks etc, are added, all of them lumped in only a friction-like term, we have [7]

$$P_\mu = -\mu v^2 = -b(r-1)^2 \qquad (4)$$

where μ is a lumped friction coefficient that embraces all the global losses, v is the piston speed and $b = \mu x_2^2 / \Delta t_{12}^2$, being x_2 the piston position at minimum volume V_2 and $\Delta t_{12} = \tau/2$ the time spent in the power stroke. Thus, the Otto model effective power output is given by

$$P = P_R - P_\mu = \frac{C_{v_1} - C_{v_2} r^{1-\gamma}}{K_1 - K_2 r^{1-\gamma}} - b(r-1)^2, \qquad (5)$$

and the cycle's efficiency by

$$\eta = \frac{P}{Q_{eff}/\tau}$$
$$= 1 - \frac{C_{v_2}}{C_{v_1}} r^{1-\gamma} - \frac{b(r-1)^2}{C_{v_1}}(K_1 - K_2 r^{1-\gamma}) \qquad (6)$$

Equation (6) reduces to $\eta_0 = 1 - r^{1-\gamma}$ (the ideal Otto efficiency) for $C_{v_1} = C_{v_2}$ and $\mu = b = 0$ and Eq. (5) reduces to P = 0 for the ideal reversible (infinite-time) case. Also, for the cycle's period,

$$\tau = K_1(T_3 - T_2) + K_2(T_4 - T_1) \qquad (7)$$

and for the lumped friction losses

$$P_\mu = -b(r_c - 1)^2 \qquad (8)$$

where r_c is the compression ratio $r_c = V_1 = V_2$ and $b = \mu x_2^2 / \Delta t_{12}^2$. The effective power output and the cycle's efficiency are given by [8]

$$P = \frac{C_P(r_c - r_E)(r_E r_c)^{\gamma-1} - C_v(r_c^\gamma - r_E^\gamma)}{K_1(r_c - r_E)(r_E r_c)^{\gamma-1} - K_v(r_c^\gamma - r_E^\gamma)} \qquad (9)$$
$$-b(r_c - 1)^2$$

and

$$\eta = 1 - \frac{r_E^\gamma - r_c^\gamma}{\gamma(r_E - r_c)(r_E r_c)^{\gamma-1}}$$
$$- \frac{b(r_c - 1)^2[K_1(r_E - r_c)(r_E r_c)^{\gamma-1} + K_2(r_E^\gamma - r_c^\gamma)]}{C_p(r_E - r_c)(r_E r_c)^{\gamma-1}} \qquad (10)$$

where C_P is the constant-pressure heat capacity and $r_E = V_1/V_3$ is the expansion ratio. Eq. (10) immediately reduces to the ideal Diesel efficiency [10] when $\mu = b = 0$ and Eq. (9) reduces to P = 0 for the ideal reversible case.

As it was remarked in Refs. 7 and 8, Eqs. (5), (6), (9) and (10) have a reasonable behavior when they are compared with real power and efficiency curves for actual Otto and Diesel engines. In fact, the obtained values for P_{max} and η_{max} in both cases are close to the real P_{max} and η_{max} values. In addition, these equations lead to loop-shaped curves for P versus η plots as it is common in many real engines [11].
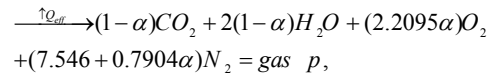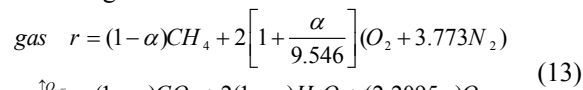
3. The Otto cycle case

In this case, we assume that the intake mixture (gas r) is composed by methane and air according to the following expression:

$$gas\ r = (1-\alpha)CH_4$$
$$+2\left[1 + \frac{\alpha}{9.546}\right](O_2 + 3.773N_2) \qquad (11)$$

where $\alpha \in [0, 0.1]$ determines the proportions of methane and air in the intake mixture, that is, it defines a mixture poor or rich in fuel contents. The molar number n_r of gas r is determined by the initial state of the mixture taken as ideal gas:

$$n_r = \frac{P_1 V_1}{\tilde{R} T_1}; \qquad (12)$$

if we take $T_1 = 350K$, $P_1 = 1.03 \times 10^5 \ Pa$, and $V_1 = 4 \times 10^{-4} \ m^3$, which are typical values of P, V , and T [12] for an initial state in an Otto cycle, with $\tilde{R} = 8.31451 \ J \ mol^{-1} K^{-1}$ the universal gas constant, we have, $n_r = 0.01415 \ mol$. When the combustion reaction occurs, gas r [Eq. (11)] is converted in gas p according to

$$gas \quad r = (1-\alpha)CH_4 + 2\left[1 + \frac{\alpha}{9.546}\right](O_2 + 3.773 N_2)$$
$$\xrightarrow{\uparrow Q_{eff}} (1-\alpha)CO_2 + 2(1-\alpha)H_2O + (2.2095\alpha)O_2$$
$$+ (7.546 + 0.7904\alpha)N_2 = gas \quad p, \qquad (13)$$

where Q_{eff} is the combustion heat.

In Figs. 2a, 2b and 2c we show the power output time series for three cases of fluctuations in the heat capacity C_{v_r} . For the efficiency case, and using the same three noisy inputs (uniform, gaussian and logistic), we obtain similar results, as can be seen in Figs. 3a, 3b, 3c.
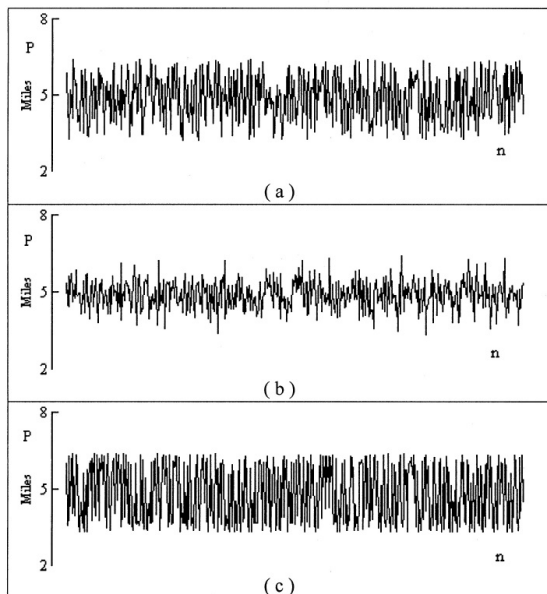

Figure 2. Power output fluctuations of the Otto cycle model: a) Uniform; b) Gaussian; c) Logistic
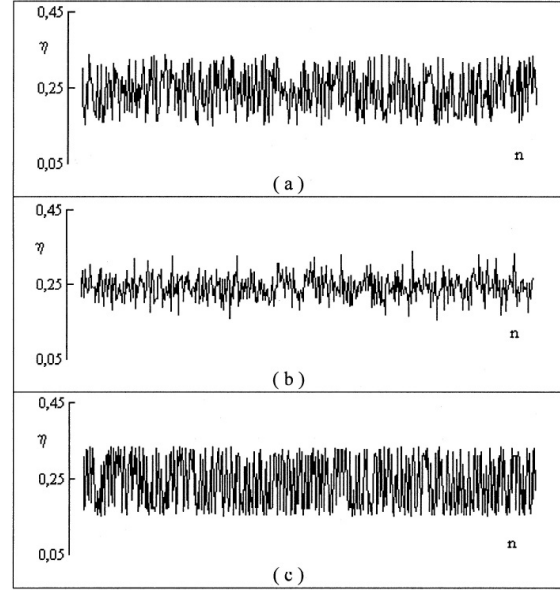

Figure 3. Efficiency fluctuations of the Otto cycle model: a) Uniform; b) Gaussian; c) Logistic

4. Conclusion

There exist many theoretical and experimental reasons for taking a sequence of thermal cycles no as an identical repetition of a representative steady-state cycle, but as a sequence of cycles changing in several thermodynamic quantities. In fact, in internal combustion engines, as the Otto and Diesel engines, the combustion heat changes from cycle to cycle due to imperfect combustion and residual gases inside the cylinder after each combustion event. Evidently, this cyclic variability in the combustion heat must produce changes in the performance of a cyclic sequence, for example, in both the power output and efficiency. Recently, Daw et al. [3] proposed an internal combustion engine model, in which, the combustion heat changes from cycle to cycle. Those authors obtained a reasonable reproduction of the experimental time series of the fluctuant combustion heat for a spark-ignited Otto engine. Starting from this model, we proposed another model [6] in which, the fluctuant combustion heat drives the thermodynamics of an Otto engine including a chemical combustion reaction, and dissipative losses. In that work [6], we find that power and efficiency are fluctuant quantities whose fluctuation sizes (standard deviation and relative fluctuation) can be driven through the managing of the thermodynamic state variables of the working fluid [6]. In the present work, we develop fluctuant models for both an Otto and a Diesel cycle, but assuming that the cyclic variability can be lumped through the fluctuations of the constant-volume heat capacity. In both cases we have obtained that the size

of fluctuations is bigger in a logistic correlated noise than in two uncorrelated noises (uniform and gaussian), that is, in the case when the time series qualitatively resembles the situation in which combustion residuals are maintained from cycle to cycle.

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