DYNAMIC MODEL OF COMBUSTION IN A PISTON ENGINE

E.Y. Bitar*, H.J. Schock# and A.K. Oppenheim*
*University of California
Berkeley, CA 94720-1740, USA
ako@me.berkeley.edu
#Michigan State University
East Lansing, MI 48823

Abstract

The sole purpose of combustion in a piston engine is to generate pressure. Hence, the recorded pressure profile provides a measure of its effective productivity. Provided by it moreover, is the input to a model expressing the performance of combustion as that of a dynamic object. Furnished for this purpose is an algorithm for a micro-electronic control system to optimize the execution of the combustion system. The output of the algorithm is made out of time profiles of the mass averaged thermodynamic parameters tracing the evolution of the combustion field, as well as of the effective product of the consumed fuel, expressed in terms of analytic functions. On this basis, gains in the reduction of fuel consumption are evaluated, bearing directly upon the minimization of pollutant formation, achievable by modulating the exothermic process of combustion by means of a micro-electronic control system,

Introduction

The sole purpose of combustion in piston engines is to create pressure for shifting the process of expansion away from compression and thus form a work cycle. It is, therefore, the measured pressure profile that provides the essential input signal to a closed loop control system for executing its exothermic process expressed in terms of dynamic parameters. For, no matter how sophisticated is a control system, it can modulate only one type of objects: dynamic. A dynamic model of combustion is construed, therefore, that, on the basis of the measured pressure profile, furnishes appropriate feed-back and feed-forward signals for the micro-electronic processor of a control system.

Specifically, the model provides the algorithm for an interface between the sensor and the processor. The algorithm treats, in effect, an inverse problem: the evaluation of the dynamic and thermodynamic properties of the combustion system on the basis of the pressure profile it generates. The results are expressed in terms of the time profiles of mass averaged temperature and density, as well as of the effectiveness with which fuel is utilized for their generation, to provide an assessment of its performance.

For this purpose, the exothermic process of combustion has to be carved out of the processes of compression and expansion – a task involving the identification of its bounds. This task is akin to the well known problem in the theory of laminar flames. To treat it, Penner and Von Karman [1], introduced the concept of an ignition temperature. Hirschfelder et al [2] developed the model of a flame holder that anchors the profile of the reaction rate. Zeldovich and Frank Kamenetsky [3] treated it by a global theory expressing the functional relationship between the reaction rate and the temperature. Williams [4] described it as ”the cold boundary difficulty.” A difficulty of this kind arises in any dynamic model of combustion, where the thermochemistry of ignition is out of scope, irrespectively whether it takes place under laminar or turbulent conditions. For this reason, it is regarded as the essential singularity of combustion.

The crucial problem involved in establishing the location of this singularity is due to its nature of a sharp, saddle-like pole - a corner. Since nature abhors corners, it cannot be identified by a measured data point. For that reason, the data tracing the evolution of pressure in an engine cylinder are expressed in terms of analytic functions, whence the location of this pole is pinpointed by their sharp intersection. The procedure developed for this purpose is called *pressure diagnostics*. Its background and modus operandi are described in [5]. Presented here is its application to a HCCI (Homogeneous Charge Compression Ignition) engine – a system that today is at the forefront of R&D in the technology of piston engines – a pioneering entry into the no-mans land between gasoline and diesel engines.

Engine

To illustrate the operating features of the model, its application to data provided by a dynamometer test of a VW TDI diesel engine with a compression ratio of r = 16.5, modified for operation as a HCCI engine where combustion is initiated by auto-ignition. For the test, the engine run at a speed of N = 1200 rpm on gasoline of 87 octane number, premixed with-air at an air-equivalence ratio of λ = 2.2, that was supplied at an inlet pressure of p_a = 1.4 atm and a temperature T_a = 325K.

Model

The model portrays the evolution of the exothermic process of combustion between the end of inlet, marked by **a** in Fig. 5, and the start of exhaust, marked by **z** in Fig 5. Its input is provided by a digitally recorded pressure profile, $p(\Theta)$, where Θ is the crank angle, displayed in Fig. 1, with data identified by circles. Profile of the cylinder volume, $v(\Theta)$, normalized with respect to v_a, is provided in an analytic form by the kinematics of the crankshaft mechanism. The input data are treated in two stages.

In the first stage, the trajectories of the data are expressed by analytic functions. Since significant sections (between **b** and **c** in Fig 5) of the processes of compression and expansion lend themselves, to accurate expressions by polytropes, the key to accomplish this task is provided by the **polytropic pressure model**, $\pi \equiv pv^n$, whose profile is displayed by circles in Fig. 2, where the compression and expansion polytropes are portrayed by horizontal lines, referred to as ***rails***. The transition from the lower to the upper rail, referred to as the ***dynamic stage***, is expressed by the **life function***, derived by regression of the data. To distinguish the functions from the data, they displayed by continuous lines. The initial point, **i**, of the dynamic stage is pinpointed by the sharp intersection of the life function with the low rail. The final point, **f**, is identified by its maximum reached at the upper rail. As demonstrated on both figures, the fit of the functions to the data is quite accurate, except, notably, for the immediate vicinity of points, **i** and **f**, - the two singularities located at the bounds of the dynamic stage.

In the second stage, the working substance is treated as a chemical system specified by the reactants, R, consisting of the fresh charge made out of fuel, F, and air, A, that is in a mixture with recirculated residual and/or exhaust gas, B. The evolution of the exothermic process, in the course of which the reactants, R, are converted into products, P, referred to as the ***exothermic stage,*** is evaluated on the basis of the balances of mass, volume and energy, with thermodynamic properties of the system established by equilibrium analysis, carried out by the use of STANJAN

* vid.[5] Chapter 4 and Appendix B

[6] with data for fuel obtained from the NIST Chemistry Webbook [7]. The results are presented in Fig. 3 – the state diagram of $e(w)$, where e is the internal energy and $w \equiv pv$ is the dynamic energy, both expressed in kJ/g. Delineated on it are the state lines for the fuel, F, the air, A, the reactants, R, and the products, P, as well as the paths of the processes, in the course of which a state on R is transformed into a state on P.

Parameters of the exothermic stage are expressed by the profiles of mass fractions of the generated products, $y_P(\Theta)$, that, for the system, are equivalent to mass fractions of the consumed fuel, together with their net effective part, $y_\varepsilon(\Theta)$, and the gross effective part, $y_E(\Theta)$. The net effective part, $y_\varepsilon(\Theta)$, is generated for charging the internal energy of the working substance – the essential role of combustion. The gross effective part, $y_E(\Theta)$, takes into account also the work performed concomitantly by the piston. The fraction of the generated products, $y_P(\Theta)$, includes, moreover, the energy lost by heat transfer to the walls. With the latter, the temperature profiles of the system, $T_S(\Theta)$, and of its components, $T_R(\Theta)$, and, $T_P(\Theta)$ are evaluated.

The results are expressed in terms of algebraic equations; their numerical algorithm is carried out by a MATLAB program.

Modulation

On the basis of the dynamic model of combustion, gains in performance of the combustion system, achievable by modulating its execution, are evaluated. The optimum performance is established by shifting the life function of the polytropic pressure model, $\pi(\Theta)$, toward the TDC along the horizontal rails of the compression and expansion polytropes, so that its point of inflection is located at TDC, as depicted in Fig. 4. From its data, the corresponding shifts in the work cycle, $p(v)$, presented by Fig. 5, and in the net effectiveness of fuel utilization, $y_\varepsilon(\Theta)$, displayed in Fig. 6, are determined. The modified profiles are delineated by broken lines, in contrast to continuous lines of the actual performance, while their initial and terminal points are denoted by primes.

According to the data of Fig. 5, the modulated IMEP, is, 8 8.3551 atm, while the actual IMEP, is 7.7712 atm – a gain of 7.51%. According to the data of Fig. 6, the maximum of the modulated effectiveness of fuel utilization is 0.7480, while that of the actual net effectiveness, is 0 6772 – a gain of 10.45%. These gains can be achieved by modulating the exothermic process of combustion by the use of pulsed turbulent jets, a task accomplishable by a PJI&I (Pulse Jet Injection and Ignition) system, operated under the control of a MECC (Micro-Electronically Controlled Combustion) system [5] – a technology that is particularly well suited to hybrid power plants run on any hydrocarbon fuel, including prominently hydrogen whose combustion with air by flames can attain extremely high temperature peaks at which significant amounts of NO are produced.

References

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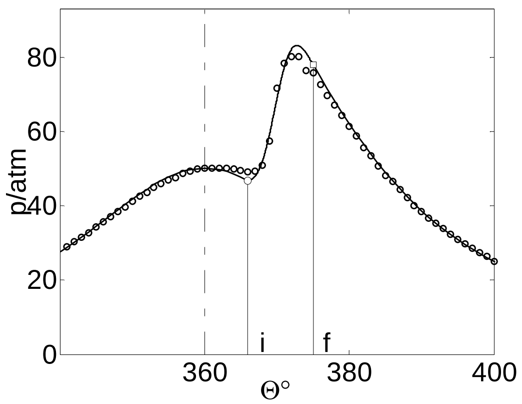

Fig. 1 Pressure profile

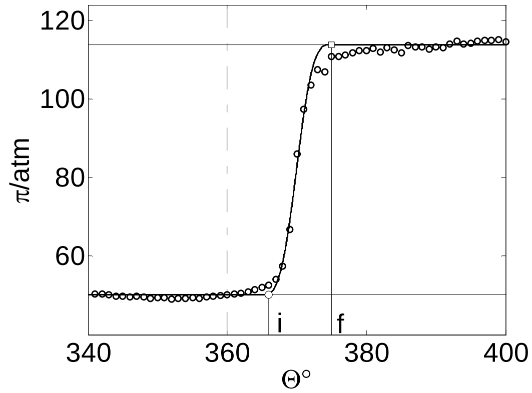

Fig. 2. Profile of polytropic pressure model

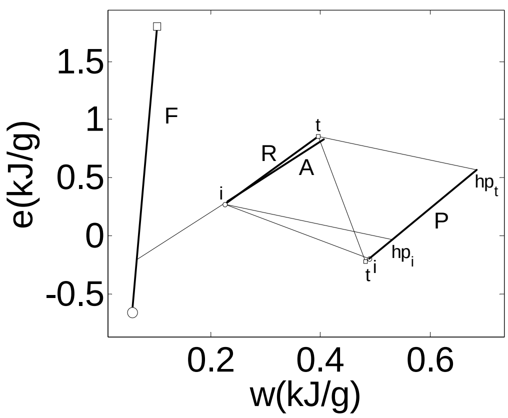

Fig. 3. State diagram, e(w)

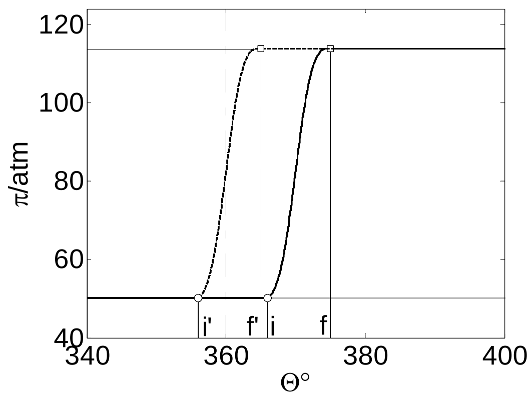

Fig. 4. Profiles of polytropic pressure model

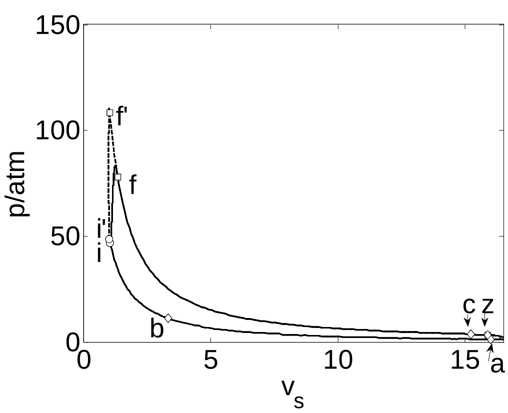

Fig.5. Work cycle

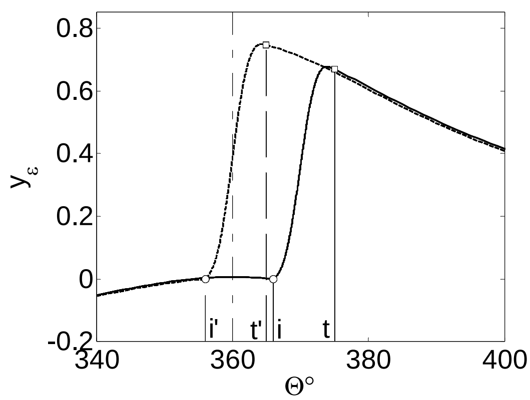

Fig.6. Net effectiveness of fuel utilization