

One Dimensional Modeling and Experimental Validation of Single Cylinder Pneumatic Combustion Hybrid Engine

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ABSTRACT

The objective of this paper is to present and to validate a numerical model of a single-cylinder pneumatic-combustion hybrid engine. The model presented in this paper contains 0-D sub-models for non-spatially distributed components: Engine cylinder, Air tank, wall heat losses. 1-D sub-models for spatially distributed components are applied on the compressive gas flows in pipes (intake, exhaust and charging). Each pipe is discretized, using the Two-Steps Lax-Wendroff scheme (LW2) including Davis T.V.D. The boundaries conditions used at pipe ends are Method Of Characteristics (MOC) based. In the specific case of a valve, an original intermediate volume MOC based boundary condition is used. The numerical results provided by the engine model are compared with the experimental data obtained from a single cylinder prototype hybrid engine on a test bench operating in 4-stroke pneumatic pump and 4 stroke pneumatic motor modes. In each mode, the prediction of the mass flow rates, amplitude and timing of the charging pipe waves are satisfactory, without using any discharge coefficient. Indicated work and p-V diagrams are similar between simulation and measurements in the case of pneumatic pump mode. For the pneumatic motor mode the model underestimates cylinder pressure during the charging process.

INTRODUCTION

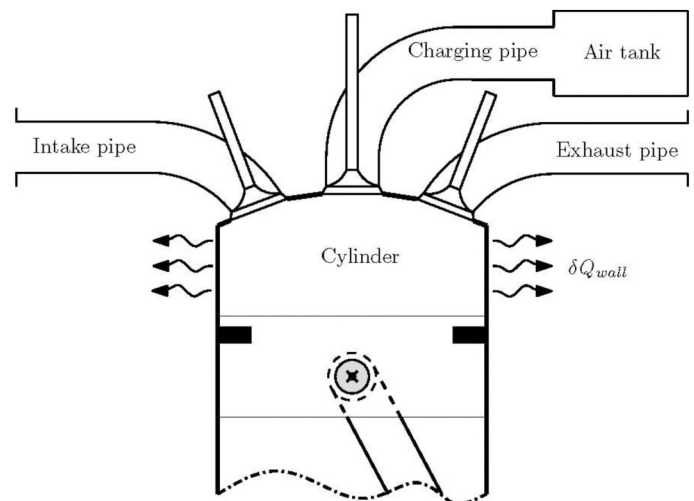


Figure 1. Schematic representation of a Single Cylinder

The concept of the pneumatic hybrid engine (1) (2) is based on a conventional internal combustion engine in which one additional valve, called the charging valve, connects the combustion chamber to an air tank. See Figure 1. In order to reduce costs, the intake and exhaust valve remain camshaft-driven and only the charging valve is moved by a fully variable actuator (3). As a result, the new major modes allowed by this additional valve (pneumatic pump, pneumatic motor and supercharged) remains 4-Stroke cycles:

- **Pneumatic Pump:** This mode transforms vehicle kinetic energy into a re-usable pneumatic energy. The cycle begins with a conventional intake stroke. The charging valve opens during compression strokes as soon as the in-cylinder pressure reaches that of the charging port. The valve closes

shortly after T.D.C, and the cycle finishes with conventional expansion and exhaust strokes.

- **Pneumatic Motor:** This mode is used to transform pneumatic energy into mechanical work without any combustion. The cycle begins with classical intake and compression strokes. The charging valve opens when approaching TDC, and remains open during part of the expansion stroke in order to maintain pressure at a high level and generate the desired indicated work. The cycle finishes with a conventional exhaust stroke.

- **Pneumatic Supercharged mode:** This mode is used to compensate for the “turbo lag” effect, while temporarily boost the engine during a strong acceleration transient. The charging valve opens shortly after intake valve closure, and closes when the desired additional air mass has flowed into the in-cylinder. The end of the cycle remains the same as a conventional four stroke one.

Several previous publications (4) (5) (6) have shown that the concept of a pneumatic-combustion hybrid engine could provide benefits in terms of fuel consumption and greenhouse gas emissions that are comparable with those of an electric hybrid vehicle (7) but with lower weight and presumably lower cost (6). To improve the performance and efficiency of the hybrid pneumatic concept, the development mainly consists in determining the timing of opening and closing of the charging valve, for a given operating mode (pneumatic motor mode, pneumatic pump mode, supercharged mode) and for a given objective (maximization or nominal control of mechanical work, maximization or nominal control of mass or enthalpy transferred, maximization of global efficiency etc.). It has been experimentally shown (8) that large amplitude pressure waves (near 2 bars of amplitude) are created inside the charging pipe as a result of the charging valve opening and closure. In order to take this pressure wave phenomenon into account a 1D simulation platform is needed.

However, the way in which gas transfer between the cylinder and the Air tank is performed depends on many parameters, such as: charging pipe length and cross section, air tank mean pressure, air tank temperature, charging valve hydraulic diameter and number, charging pipe roughness, charging valve lift. The large number of parameters to be explored for each operating mode and each objective makes optimization almost impossible to perform experimentally. A reliable, flexible, and fast numerical 1D model is therefore needed in order to develop control laws. Two previous studies (9) (10) have used a 1-D commercial simulation software for engine simulations, but do not present any comparison between experimental and simulated results for wave timings and amplitude, mass flow rates etc. From our point of view, the use of such commercial code for a pneumatic hybrid engine poses one problem and leaves one question unanswered. The problem is that the software requires a precise knowledge of

the discharge coefficient versus lift of each simulated valve (11). These coefficient arrays have to be determined by experiments thus a prototype cylinder head is required. An estimated discharge coefficient will therefore be more or less erroneous, leading to serious discrepancies between simulations and future measurements. The unanswered question is that the valve boundary condition implemented in this kind of software is seldom validated on pressure ratios as high as those encountered in a pneumatic hybrid engine (up to 22). To circumvent the previous points, authors have developed their own MOC valve boundary condition that obviates in many cases the use of a Coefficient of discharge and has been experimentally validated on a valve gas dynamics test bench (12). The objective of the paper is now to present the complete engine model that embeds this valve boundary condition, and to compare experiments versus simulations.

The paper is organized as follows: Firstly, 0D components models are described: the cylinder and the air tank model. Secondly, the 1D sub models for spatially distributed components are described: the LW2 numerical scheme and the simplified TVD scheme proposed by Davis are presented, followed by the Intermediate Volume Valve boundary condition. Thirdly the results produced by the complete single cylinder hybrid engine are presented and compared to measurements obtained with a prototype engine.

QUASI-DIMENSIONAL COMPONENTS OF THE ENGINE MODEL

AIR TANK

The air tank thermodynamic state shown on [Figure 1](#) is determined using the first law of thermodynamics:

$$dU = \delta W + \delta Q + h_i \cdot dm \quad (1)$$

Because the tank has a constant volume, $\delta W = -pdV = 0$. According to Joule's law $dU = mc_v dT + c_v T dm$, with T the temperature of the gases contained in the Tank. The air tank is assumed to be adiabatic, then $\delta Q = 0$. For a perfect gas, the specific enthalpy is given by $h_i = c_p T_i$, with T_i the temperature of the flowing gases. Finally, a re-arrangement of [Eq. \(1\)](#) gives:

$$dT = \frac{(c_p T_i - c_v T) \cdot dm}{mc_v} \quad (2)$$

The air flow mass dm is determined at the open end boundary (see corresponding section of this paper) using the method of characteristics (M.O.C). A classical numerical integration

then provides the total air mass inside the tank. Using Eq. (2), the elementary temperature rise in the tank is determined. Applying a numerical integration, the new tank temperature is computed. Finally, using the perfect gas law ($pV=mrT$), the mean tank pressure is determined.

ENGINE CYLINDER

The cylinder is the central part of the system. It exchanges heat and mass with the outside. Its thermodynamic state is determined using the first law of thermodynamics (Eq (1)).

W is the mechanical work generated by the moving walls, Q is the total heat, m the mass and H is the enthalpy. Subscript i indicates an intake flow, c indicates a charging pipe flow and e indicates an exhaust flow. Then:

$$dm = dm_i + dm_c + dm_e \quad (3)$$

Each mass flow rate is determined using a valve boundary condition (See corresponding section of this paper). A classical numerical integration technique then yields the new total mass inside the cylinder.

The total heat Q is the sum of a combustion heat Q_c and convective heat transfer through cylinders walls Q_w .

$$\delta Q = \delta Q_c + \delta Q_w \quad (4)$$

δQ_w and δQ_c are respectively determined using the Woshni (13) and Wiebe (14) models. These classical correlations are not detailed in this paper. Re-arranging Eq (1) finally gives:

$$dT = \frac{-pdV + \delta Q + c_p \sum (T_k dm_k) - c_v T dm}{mc_v} \quad (5)$$

A numerical integration gives the new temperature of the cylinder. The volume is determined from the engine geometry and crankshaft position (15). The pressure is then calculated using ideal gas law.

ONE-DIMENSIONAL COMPONENTS OF THE ENGINE MODEL

FUNDAMENTALS

Flows, in each pipe, are assumed to be mono dimensional. The ideal compressive gas is assumed to be inviscid. Three conservation laws are written for a control volume, respectively the mass, momentum and energy. Figure 2 shows the flow of an ideal compressible gas through an infinitesimal section of pipe, p , ρ , u , t , x and F are

respectively the pressure, the density, the velocity, the time, the space and the cross section area.

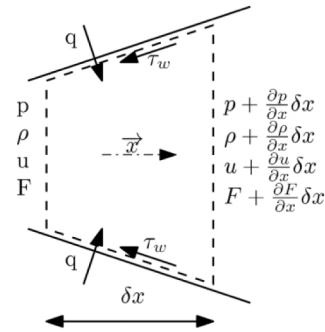


Figure 2. Control Volume

Continuity equation: The rate of change of mass within the control volume is equal to net mass flow rate through the element. This principle leads to (16) (17):

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} + \frac{\rho u}{F} \frac{dF}{dx} = 0 \quad (6)$$

Momentum equation: The rate of change of momentum within the control volume is equal to the sum of the forces applied on the control volume. In order to model the friction between the flow and the wall, a shear stress τ_w , opposed to the flow, is introduced. Expressing the shear stress in terms of

the pipe wall friction f gives $\tau_w = \frac{1}{2} \rho u^2 f$. D is the diameter of the pipe.

$$\frac{\partial \rho u}{\partial t} + \frac{\partial \rho u^2}{\partial x} + \frac{\rho u^2}{F} \frac{dF}{dx} + \rho G = 0 \quad (7)$$

$$G = \frac{1}{2} u |u| f \frac{4}{D} \quad (8)$$

Energy equation: The energy equation is derived from the first law of thermodynamics applied to the control volume. Heat transfer per unit mass of gas q has been introduced to take into account the heat transfer between the pipe and the fluid. q is determined from an approximate treatment (16), where T_w and T_g are the temperature of the pipe wall and gas respectively. e_0 and h_0 are respectively the stagnation specific energy and enthalpy.

$$\frac{\partial \rho e_0}{\partial t} + \frac{\partial \rho u h_0}{\partial x} + \frac{\rho u h_0}{F} \frac{dF}{dx} - \rho q = 0 \quad (9)$$

$$q = \frac{2 f u}{D} \frac{\gamma R}{\gamma - 1} (T_w - T_g) \quad (10)$$

TWO-STEP LAX-WENDROFF SCHEME + T.V.D

The three previous governing equations can be favorably rewritten in vectorial form:

$$\frac{\partial W}{\partial t} + \frac{\partial F(W)}{\partial x} + C(W) = 0 \quad (11)$$

$W = [\rho, \rho u, \rho e_0]^T$ is the state vector of conservative variables: mass, momentum and total energy

$F(W) = [\rho u, \rho u^2 + p, \rho u h_0]^T$ is the flux vector. Vector $C(W)$ is the correction vector. It takes into account section variation, pipe wall friction and wall heat transfer:

$$C(W) = \begin{bmatrix} \rho u \\ \rho u^2 + p \\ \rho u h_0 \end{bmatrix} \frac{d(\ln F)}{dx} + \begin{bmatrix} 0 \\ \rho G \\ -\rho q \end{bmatrix} \quad (12)$$

Eq (11) can be solved by a two-step Lax-Wendroff method (18), as shown in Figure 3.

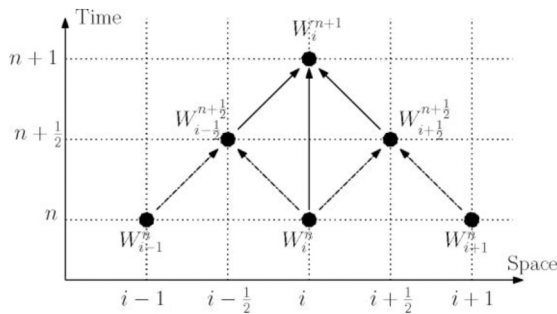


Figure 3. 2 steps Lax Wendroff (Richtmeyer) stencil

First step :

$$W_{i+\frac{1}{2}}^{n+\frac{1}{2}} = \frac{1}{2} [W_{i+1}^n + W_i^n] - \frac{\Delta t}{2\Delta x} [F(W_{i+1}^n) - F(W_i^n)] - \frac{\Delta t}{4} [C_{i+1}^n + C_i^n] \quad (13)$$

$$W_{i-\frac{1}{2}}^{n+\frac{1}{2}} = \frac{1}{2} [W_i^n + W_{i-1}^n] - \frac{\Delta t}{2\Delta x} [F(W_i^n) - F(W_{i-1}^n)] - \frac{\Delta t}{4} [C_i^n + C_{i-1}^n] \quad (14)$$

Second step :

$$W_i^{n+1} = W_i^n - \frac{\Delta t}{\Delta x} \left[F(W_{i+\frac{1}{2}}^{n+\frac{1}{2}}) - F(W_{i-\frac{1}{2}}^{n+\frac{1}{2}}) \right] - \Delta t C_i^n \quad (15)$$

Because the LW2 scheme is not Total Variation Diminishing (T.V.D) (19) spurious oscillations appear at shocks and discontinuities. A flux limiter has to be added to the original LW2 scheme. The simplified T.V.D scheme of Davis (20) has been used. The advantages of this scheme are that it is symmetrical (i.e. does not depend on the direction of travel of the characteristics), and that it is easily implemented in the calculation process:

$$W_i^{n+1} = W_i^n - \frac{\Delta t}{\Delta x} \left[F(W_{i+\frac{1}{2}}^{n+\frac{1}{2}}) - F(W_{i-\frac{1}{2}}^{n+\frac{1}{2}}) \right] - \Delta t C_i^n + \left[\bar{K}_{i+\frac{1}{2}}^+(r_i^+) + \bar{K}_{i+\frac{1}{2}}^-(r_{i+1}^-) \right] (W_{i+1}^n - W_i^n) + \left[\bar{K}_{i-\frac{1}{2}}^+(r_{i-1}^+) + \bar{K}_{i-\frac{1}{2}}^-(r_i^-) \right] (W_i^n - W_{i-1}^n) \quad (16)$$

The functions $\bar{K}_i^{+-}$ and r_i^{+-} are not detailed in this paper, and the reader is referred to the original paper (20).