Development and Validation of a 0-D/1-D Model to Evaluate Pulsating Conditions from a Constant Volume Combustor

P. Gallis[1], D. Misul[1], S. Salvadori[1], M. Bellenoue[2], B. Boust[2]

[1]Department of Energy, Politecnico di Torino, Corso Duca degli Abruzzi, 24 – 10129, Torino, Italy

[2]PPRIME Institute, CNRS-ENSMA-University of Poitiers, Avenue Clément Ader, 1 – 86961, Chasseneuil, France

Abstract

The growing demand in electrical power, the increased number of flight passengers, and the constantly increasing mean global temperature level push towards a remarkable performance improvement of Gas Turbines (GTs), which are yet probably close to their technological limit. Hence, research community in turbomachinery is trying to adopt a different GT cycle based on the Pressure Gain Combustion (PGC) concept. This innovative technology exploits a pressure rise within the combustion process (e.g., Constant Volume Combustor, Rotating Detonation Combustor, etc.), thus theoretically leading to higher thermal efficiency (+15% for small/medium GT size) and conversely to lower specific fuel consumption. Institute Pprime operates a Constant Volume Combustor (CVC) with rotary valves fed by a mixture of air and liquid iso-octane. A rectangular duct coupled with a converging-diverging nozzle is placed downstream from the combustion chamber. An experimental campaign by Pprime offers time-resolved pressure measurements of the test rig that serve as boundary condition and validation data for numerical simulations. The present paper aims at performing 0-D / 1-D modelling of CVC using the GT-Power software, which is an Internal Combustion Engine (ICE) modelling tool that involves piston's motion. The latter is here modified in accordance to the peculiar motionless case of CVC. By considering a parametrization of discharge coefficients, a modified non-dimensional burning rate, and an appropriate heat transfer model the simulation is able to match the experimental pressure fluctuation with acceptable accuracy. The main outcome of the present activity relies on the definition of credible CVC exit conditions, which otherwise are unavailable. The obtained total pressure and total temperature pulsating conditions are necessary to study the existing nozzle, and will be used to design a new transition duct able to weaken flow pulsation and to properly feed the turbine, thus counteracting the cycle's efficiency reduction associated to the unsteady CVC outflow.

keywords: Pressure Gain Combustion, Constant Volume Combustor, Direct Injection, Rotating Valves, 0-D / 1-D Modelling

1 Introduction

According to the data published by Eurostat [1], an exponential increase of almost 43% of the air passengers of European countries from 2008 led to the remarkable number of one billion air passengers in 2019. At the same time, the rapid increment of electric power demand and the increase in mean global temperature are reported. Conventional Gas Turbines' (GTs) thermodynamic efficiency has probably reached its limit due to the steady, subsonic constant pressure combustion process. Consequently, current research by the turbomachinery community is characterized by relevant changes in the definition of the gas turbine components to increase the thermodynamic efficiency by several points.

An attractive solution derives from Pressure Gain Combustion (PGC) [2]. PGC exploits a pressure rise along the combustion process, thus theoretically allowing for higher thermal efficiencies and lower specific fuel consumption [3]. In particular, PGC could function efficiently with a reduced pressure ratio delivered by the compressor, thus operating with less compressor's stages and reducing the overall cost and weight of a gas turbine [4]. On the other hand, a reduced performance associated to the unsteady exhaust flow of a PGC combustor, the non-uniform temperature field at the turbine inlet and the demanding secondary air system may cancel the theoretical boost of the cycle's efficiency [5].

One of the processes of PGC is the isochoric combustion. Institute Pprime developed a Constant Volume Combustor (CVC) with rotary valves, using mixture of air and liquid iso-octane [6]. A coupled experimental and numerical activity [7] denoted the importance of both the local spark timing velocity and the residual burned gases on the function and cycle variations of the CVC. Later, a converging-diverging nozzle was placed after the chamber. An experimental campaign of fast response pressure measurements was conducted in order to characterise the system in respect of pressure gain [8]. The experimental activity uncovered interesting evidences of the CVC performance in respect to operating frequency and the size of the subsequent nozzle. Nevertheless, the need of a detailed analysis of the exhaust flow field was stressed.

Under that prism, the present paper aims at 0-D / 1-D modelling of the CVC using the commercial software GT-Power, which serves as a robust simulation tool for countless cases of Internal Combustion Engines (ICE). The experimental data by Pprime [8] were deployed as boundary conditions and validation data for the numerical model. The substitution of the rotary valves properties for the poppet valves is performed and the discharge coefficient profiles are parameterized using the n^{th}-Root Function.

Furthermore, the burning rate and the heat transfer model of the chamber are carefully elected in order to produce a representative cycle in respect of the experiments. As a result, the numerical model offers the spatially-average time-resolved profiles of stagnation pressure and temperature after the exhaust valves, which will act as inlet boundary conditions for the unsteady CFD simulation of the exhaust nozzle and for the design of an effective transition duct in an upcoming activity.

2 0-D/1-D Model Setup

2.1 CVC Configuration

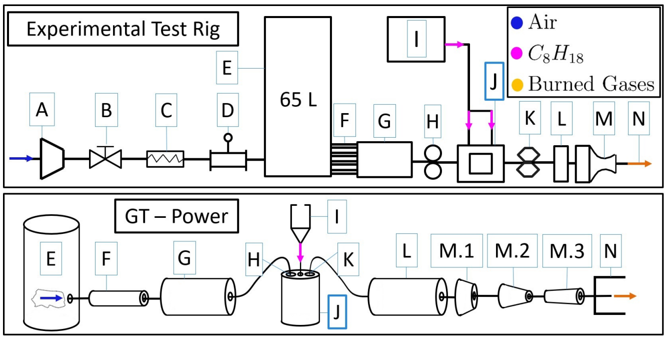

Figure 1: CVC Configuration and Equivalent GT-Power Model

At the top of Fig. 1, once can notice the experimental configuration of the CVC by Pprime [8]. Additionally, at the bottom of Fig. 1 it could be identified the equivalent 0-D / 1-D model that was developed. The whole intake air supply system of test rig consists of a Compressor (A), a Dome Regulator (B), a Heater (C), a Mass Flow Meter (D), and a Reservoir of 65 L (E). In the model, the analysis starts from the Reservoir component (E), in which the steady stagnation pressure (P_{in}) and temperature (T_{in}) are obtained by experiments and are inserted as inlet boundary condition to the model system. The four intake pipes (F) that connect the reservoir with the rectangular intake plenum (G) are substituted by one adiabatic tube component coupled by a circular duct. The inlet (H) and exhaust (K) rotary valves are modelled as regular poppet valves. The operating frequency of the valves is 25 Hz. For each valve component, the lift over the time is tuned appropriately and the reference valve's diameter is carefully elected in order to result in the same cross section area of each rotary valve. The combustion chamber (J) is a cylindrical chamber with the same surface area and volume (0.65 L) as the rectangular one of the test rig. The experimental wall temperature (T_{wall}) is imposed in order to evaluate the heat losses. In particular, it must be underlined that the piston of the cylindrical chamber remained motionless offering a constant volume component during the cycle's period. Moreover, the injection system (I) is simulated using an injector with exactly the same properties of the test rig ($\dot{m}_f = 20$ $[g/s]$, $\phi_{overall} = 1$) supplying the chamber with liquid iso-octane. The whole exhaust system's volume is 0.62 L and it is consisted of a rectangular spacer coupled by a converging-diverging nozzle ISO9300 with a throat diameter 20 mm. The rectangular spacer (L) is represented as a circular adiabatic pipe using the hydraulic diameter. The converging-diverging nozzle (M) is here replaced by three adiabatic conical tubes (M.1, M.2 & M.3) to take into account the transition of rectangular to circular cross section area, the converging, and the diverging parts respectively. At its end, the outlet of the system is modelled with ambient conditions (N).

GT-Power solves the time-resolved equations of mass, momentum, and energy of each component, considering them as sub-volumes. A convergence of the numerical analysis was achieved when the absolute difference of pressure for two subsequent cycles is less than 10^{-3} bar and the absolute difference of the pressure peak's time moments for two consecutive cycles is less than 10^{-4} deg.

2.2 Valve's Parametrization

It is important to underline that the gaps of the rotary valves are taken into account as a lash value to the poppet valves of the model. As a result, even when the valves are closed there is an exchange of mass between the components due to the gaps. A crucial step of the analysis is the selection of discharge coefficient profiles. A selection of a constant coefficient profile is inadequate to model reality. In fact, the coefficients are a function of the valves' lift. The use of the n^{th}-Root Function (Eq. 1) offers the proper modification of the coefficients over the time imposing their values at minimum (Lash) and maximum lift.

$$f(x) = \frac{A}{B} \cdot x^{1/n} + C \tag{1}$$

The discharge coefficient profiles (Cd) over the lift (L) can be seen at Eq. 2 and their values for each valve are reported in Table 1.

$$Cd(L) = [\frac{Cd_{max} - Cd_{Lash}}{(L_{max} - Lash)^{1/n}}] \cdot (L - Lash)^{1/n} + Cd_{Lash} \tag{2}$$

Valve	**n**	$\mathbf{Cd_{max}}$	$\mathbf{Cd_{lash}}$
Intake	2.5	0.3	13 % of Cd^{in}_{max}
Exhaust	1	0.7	6 % of Cd^{ex}_{max}

Table 1: Properties of Discharge Coefficients Profiles

2.3 Non-Dimensional Burning Rate Profile

Afterwards, a suitable non-dimensional profile of the burning rate (X_b) is imposed to the model. The use of Hyperbolic Tangent Function (Eq. 3) is here preferred. This formula is selected instead of traditional curves (e.g., Wiebe Function) because of the parameters t_0 and RF, which allows to appropriately adjust the duration and the steepness of the burning rate, respectively. Hence, the initial and final point of the combustion are controlled modifying simultaneously the gradient of the curve over the time.

$$X_b = \frac{e^{RF(t-t_0)} - e^{-RF(t-t_0)}}{e^{RF(t-t_0)} + e^{-RF(t-t_0)}} \tag{3}$$

2.4 Heat Transfer Model

Furthermore, the heat transfer model of the chamber must be defined. First of all, there is the need to evaluate the average chamber's velocity (Eq. 4). The mixture velocity in the combustor when the valves are closed (U_A) is retrieved by using the Eddy-Break Up model [9]. The moments when only the exhaust valve is open (U_B) or only the intake valve is open (U_C), the exhaust or inlet valve's throat velocity (U_{out} or U_{in}) is elected, using a scaling factor. Eventually, when the two valves are simultaneously open an average of the two scaled velocities is chosen (U_D).

$$U_{cc} = \begin{cases} U_A = U_0\ [1 + (C_{\epsilon 2} - 1)\frac{t-t_0}{\tau_{0,EBU}}]^{\frac{-1}{2(C_{\epsilon 2}-1)}} \\ U_B = s_{out}(t)\ \cdot\ U_{out} \\ U_C = s_{in}(t)\ \cdot\ U_{in} \\ U_D = \frac{s_{in}(t)\ \cdot\ U_{in}\ +\ s_{out}(t)\ \cdot\ U_{out}}{2} \end{cases} \tag{4}$$

The scaling factors are obtained by evaluating the continuity across the valves' throat and the chamber's middle cross section area K (Eq. 5).

$$\begin{aligned} \dot{m}_{in/out} &= \dot{m}_{cc|K} \rightarrow \\ U_{cc} = U_{in/out} &\cdot \frac{\rho_{in/out}\ \cdot\ A_{in/out}}{\rho_{cc}\ \cdot\ A_{cc}} \rightarrow \\ s_{in/out}(t) &= \frac{\rho_{in/out}\ \cdot A_{in/out}}{\rho_{cc}\ \cdot\ A_{cc}} \end{aligned} \tag{5}$$

Once the average chamber's velocity is attained for each time moment, the Reynolds Number (Re) is also known. By defining a value for Prandtl Number ($Pr \approx 0.7$), the Nusselt Number (Nu) can be calculated. Consequently, the heat transfer coefficient inside the chamber is derived by the Dittus-Boelter Correlation (Eq. 6), and the heat losses though the walls are obtained.

$$h_{cc} = \frac{\lambda\ Nu}{D_K} = \frac{\frac{\mu cp}{Pr}\ \cdot\ 0.023\ Re^{0.8}\ Pr^{0.33}}{D_K} \tag{6}$$

3 Results

At the top of Fig. 2, the comparison between the experimental and the numerical data for the pressure inside of the chamber is presented. It is important to stress that the cycle starts with the ignition. Then, the combustion occurs and is followed by the exhaust phase, the scavenging process, and the intake phase. The cycle ends with the direct injection inside of the chamber. The pressure traces of the active cycles over one cycle's period inside the chamber are reported using a black line. After performing the ensemble average over each time step, the green curve is attained. A straight comparison of the experimental data could be performed after displaying the red curve obtained by the simulation. On the bottom of Fig. 2, the percentage error of the simulation with respect to the ensemble average of the experiments can be seen. In addition, for each time step the range of error between the maximum and minimum recorded pressure values with respect to the ensemble average are reported.

Concerning the combustion simulation in Fig. 2, a deep analysis is necessary to determine its real accuracy. As it is mentioned by Labarrere et.al [7], inside of the CVC chamber strong cycle variations are present from period

Pressure Traces of Active Cycles ($[P_{cc}]_a$), Ensemble Average ($\overline{P}_{cc}$), Simulation ($[P_{cc}]_{GT}$) & their error evaluation

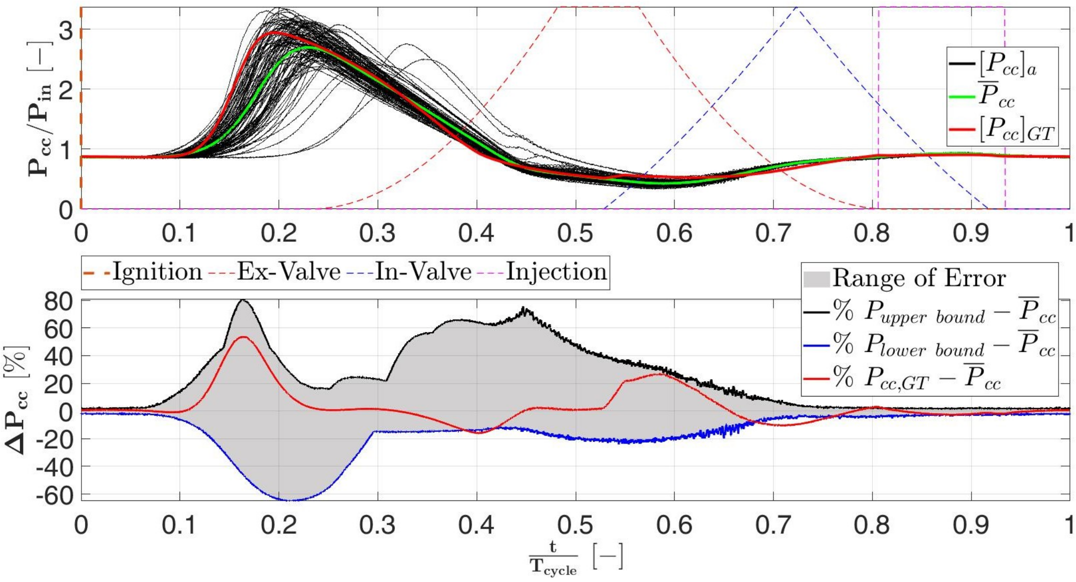

Figure 2: Validation of Simulation and Evaluation of Errors

to period, due to the presence of burned gas residuals. A cycle with a strong combustion event is accompanied by a subsequent milder combustion characterised by a slight delay. As a result, the ensemble average tries to balance the chronic and dynamic difference between the two types of cycle. Thus, the difference between the ensemble average and the simulation evaluated during the combustion phase cannot fully define the success of the simulation.

In this case, a comparison between the simulation results and the range of active cycles values might provide more interesting information. At the bottom of Fig. 2, once could observe that the simulation is inside the acceptable range in the majority of the period. Some minor exceptions ($\frac{t}{T_{cycle}} = 0.4,\ [0.69 - 0.75]\ \&\ 0.8$) occur due to the difficulties of matching the pressure level of a very peculiar combustor using a 0-D / 1-D model. The approximation of the discharge coefficient profiles offered the best possible solution of the chamber's pressure, concerning the very tight range of the experiments during the scavenging and intake phase ($\frac{t}{T_{cycle}} =\ [0.55 - 1]$).

4 Conclusions

The topic of this activity is the production of 0-D / 1-D Model of the CVC developed by Institute Pprime to retrieve the time-resolved spatially-average properties of the flow after the exhaust rotary valves using the commercial software GT-Power. A comparison of the experimental test rig and the model setup is presented and the boundary conditions of the system are underlined. Later, the parametrization of the discharge coefficient profiles over the valves' lift is explained. The reasons of the election of the Hyperbolic Tangent Function as non-dimensional burning rate of the system are thoroughly described. Afterwards, the evaluation of the average chamber's velocity in order to account the heat losses through the chamber's wall is commented.

A straight comparison of the experimental signals, their ensemble average and the simulation results for the pressure inside the chamber is presented. The results of GT-Power may differ from the ensemble average, but they are in accordance with the majority of the strong combustion cycles. The error of simulation over the average signal is inside the range of error of the experiments during the major part of the cycle's period. Consequently, the numerical analysis of the activity achieved to reproduce a representative cycle of CVC test rig.

The stagnation properties downstream of the exhaust valves will be used as the unsteady spatial uniform inlet boundary conditions for the 3-D numerical analysis of the converging-diverging nozzle. In the end, an optimization process will take place in order to define an appropriate geometry to couple the CVC with a High Pressure Turbine Stage (HPT) able to alleviate the pulsating behaviour of the exhaust flow field and provide qualitative flow to turbine.

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