

CFD Gas-Dynamic Noise Prediction of a VVA Engine Intake System

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ABSTRACT

Modern VVA systems offer new potentialities in improving fuel consumption for spark-ignition engines at low and medium load, meanwhile they grant a higher volumetric efficiency and performance at high load. Recently introduced systems enhance this concept through the possibility of modifying the intake valve opening, closing and lift, leading to the development of almost 'throttle-less' engines.

However, at low loads, the absence of throttling, while improving the fuel consumption, also produces an increased gas-dynamic noise at the intake mouth. Wave propagation inside the intake system is in fact no longer absorbed by the throttle valve and directly impact the radiated noise.

In the paper, 1D and 3D simulations of the gas-dynamic noise radiated by a production VVA engine are performed at full load and in two part-load conditions. Both models are firstly validated at full load, through comparisons with experimental data.

Different settings of the VVA device at part-load are moreover considered to quantify the trade-off existing between fuel consumption and radiated noise reduction. The analyses put into evidence that the optimal control strategy for fuel consumption improvement does not allow a satisfactory abatement of the radiated noise, as well.

Presented results show that, at low engine speed, the VVA setting realizing the lowest fuel consumption also produces an acceptable overall noise level. On the contrary, a small fuel consumption penalty must be paid at medium speed to realize a sufficiently low radiated noise.

The proposed 1D-3D procedure hence constitutes a very useful tool to realize a fully-numerical pre-calibration of the engine under development.

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INTRODUCTION

Modern internal combustion engines show more and more complex architectural solutions aiming to increase performance, and reduce fuel consumption, pollutants emissions and radiated noise.

According to the well-known "Downsizing" design philosophy, a tendency is consolidated in reducing the spark-ignition engine displacement, while employing a turbocharger to restore the prescribed power/torque output [1]. In this way, fuel consumption improvements are obtained at low load, due to lower mechanical and pumping losses. Additional advantages are related to a higher low-speed torque and thus to a better vehicle drivability [2,3].

A further reduction of pumping losses can be moreover obtained through the employment of completely flexible valve actuation systems. Initially, these systems only allowed to control intake and/or exhaust valve phasing (VVT - Valve Variable Timing devices or cam phasers) [4,5,6,7]. Most recent solutions give indeed the opportunity to independently define valve opening and closing events, and maximum lift (VVA - Variable Valve Actuation systems) [8,9]. This flexibility allows to control the engine load with a very limited butterfly valve throttling [10,11], so drastically reducing the pumping work at a low/medium load. Moreover, a reduction in NO_x emission and an improved combustion stability at idle can be realized [12,13,14,15,16]. The combustion stability can be further enhanced if a double lift

mode during each intake stroke is available, thanks to the higher in-cylinder turbulence.

A typical drawback of a VVA system is, however, the worsening of gas-dynamic noise emissions at low load. The reduced throttling of the intake ducts, while allowing a reduction of the pumping work, also produces a less effective attenuation of pressure wave in the intake system [17]. For the above reason, a precise setting of parameters for the engine control must be specified in order to get the best compromise between improved fuel consumption and reduced radiated noise [18,19].

In this paper a deep analysis of a production turbocharged VVA engine is performed. Table 1 reports the characteristics of the investigated engine.

Table 1. Engine Main Features

Model	2 cylinders, 8 valves, VVA
Displacement	875 cm ³
Stroke/Bore	86 mm / 80.5 mm
Connecting rod length	136.85 mm
Compression Ratio	9.9
Max Brake Power	64.6 kW @ 5500 rpm
Max Brake Torque	146.1 Nm @ 2500 rpm

Intake valve actuation is provided by an electro-hydraulic system which modules the valve lift according to the operating conditions. The system consists of a small capacity between the cam and the valve filled by lubricant oil, whose emptying is regulated by an electronically controlled electro-valve [20].

Numerical analyses are performed on the above engine through the use of 1D [21,22] and 3D tools [23,24,25,26]. Both performance and the intake-radiated sound pressure level (SPL) are predicted in different engine operating conditions.

1D analyses are carried out through the employment of the commercial GT-Power™ software [27], including the schematization of the whole engine from the intake mouth up to the exhaust tailpipe. The time-varying pressure profile in a section located just upstream the turbocharger compressor is extracted from 1D analyses and is used as a boundary condition in a more refined 3D CFD unsteady analysis. CFD simulations, performed with FLUENT™ package [28], are limited to the air filter box, and include the evaluation of both gas-dynamic and aerodynamic noise sources. Overall SPL is evaluated at 1cm from the intake mouth.

The whole engine speed range at full load is firstly investigated. 1D and 3D results are validated through comparisons with experimental data collected at the test-bench. The 1D model shows the capability to predict the main overall performance parameters, and the radiated noise with good accuracy. 3D CFD analyses allow to further improve the prediction of the gas-dynamic noise, in terms of overall SPL and frequency spectrum, as well.

In addition, two typical part-load conditions (2000 rpm, 2 bar BMEP and 3000 rpm, 3 bar BMEP) are analyzed. In the

selected constant-speed/load points, different combinations of the control parameters below are explored:

- β throttle opening angle
- θ spark advance timing
- φ_2 intake valve closing angle

Different control strategies in fact realize different fuel consumption and radiated noise levels, and a trade-off between them is discussed.

Numerical results in particular put into evidence that the control strategy realizing the lowest fuel consumption does not always allow a satisfactory abatement of the radiated noise, as well. Different settings of the VVA system are, hence, required to the aim of contemporary minimizing fuel consumption and gas-dynamic noise.

FULL LOAD OPERATION

Performance Prediction - 1D Model

The engine model is developed in GT-Power environment, well known commercial software based on a 1D description of flow inside the intake and exhaust pipes. The combustion process and turbulence phenomena are described using sub-models introduced in GT-Power through “user routines”. In particular, a “fractal combustion model” [29,30] is employed; it is a phenomenological model sensing both combustion system geometry (head and piston shape, spark plug position, etc.) and operating parameters such as the spark advance and the air to fuel ratio. The turbulence phenomenon, in particular, is described by a sub-model accounting for turbulence production during to the intake stroke [26,30] and its subsequent destruction; this allows to take into account variations in burning speed related to different VVA control settings.

VVA device is handled in the model by an additional “user routine”, which selects the lift profile depending on the engine operating conditions and control parameters. A “database” of predefined lift profiles, is preliminary computed for various engine speeds (starting from 1000 rpm up to 6000 rpm, step 500 rpm) and φ_2 values. Valve profiles are derived by a model of the valve actuation system developed in Amesim™ environment, as a function of the cam profile, the inertial data (valve mass, etc.), the spring preload, the engine speed, the electro-valve opening and closure angles and the thermo-physical properties of lubricant oil (especially, the oil viscosity).

The 1D schematization of the tested engine includes the presence of a “virtual microphone”, located at 1 cm from the intake mouth. This object computes the pressure field at the microphone location by treating the intake orifice as a simple pulsating monopole, and assuming free field conditions [27,31].

1D model results and validation

The experimental settings of the intake valve closure, the A/F ratio and the spark advance are specified in the 1D model at each engine speed and full load. Waste-Gate turbine opening is identified by a PID controller with the aim of reproducing the experimental derived pressure level in the cylinder #2 at the compression TDC. The “fractal combustion model” includes few tuning constants, mainly acting on tumble level and burning speed. Once tuned, model constants are kept fixed in each analyzed operating condition.

The results shown in Figures 1-2 show the numerical-experimental comparison in terms of power, torque and brake specific fuel consumption (BSFC). Due to confidentiality reasons, in the Figures the difference between computed parameters and the measured value relative to 5500 rpm is plotted. The agreement for all the performance parameters is satisfactory, except for the higher engine speeds; computed relative BSFC (Figure 2) is in fact considerably different than experimental one for rpm>3500. This is probably due to uncertainties in the Amesim-computed lift profiles, affecting the volumetric efficiency and the pumping work.

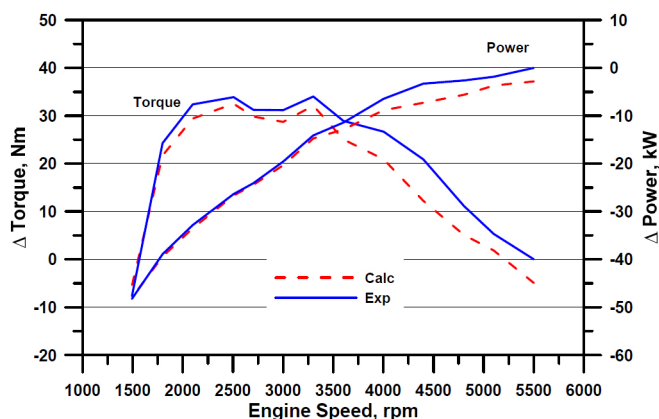


Figure 1. Relative torque and shaft power

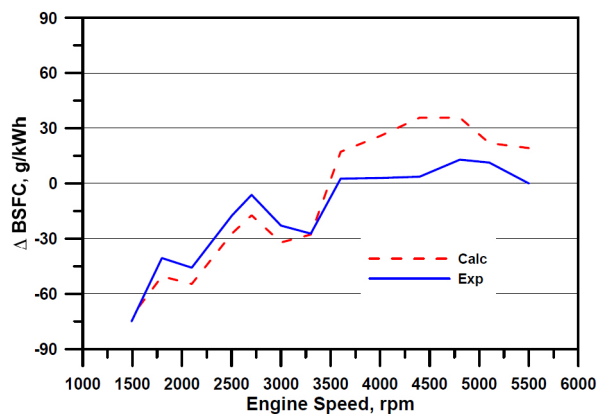


Figure 2. Relative BSFC

Figures 3a-3c show the experimental/numerical comparison of pressure cycles for 5100, 3000, and 2100 rpm.

In this case, pressure cycles are non-dimensionalized with reference to the maximum level of the experimental pressure cycle at 5500 rpm. Especially at low engine speeds, spark timing occurs late after TDC, creating the displayed double-peak pressure cycles; this choice, although hardly penalizing the indicated mean effective pressure (IMEP) and BSFC, is required to avoid knocking occurrence. Despite the above non-conventional operating conditions, a great accuracy is reached in the prediction of the in-cylinder pressure cycle, mainly due to the refined user-combustion model employed.

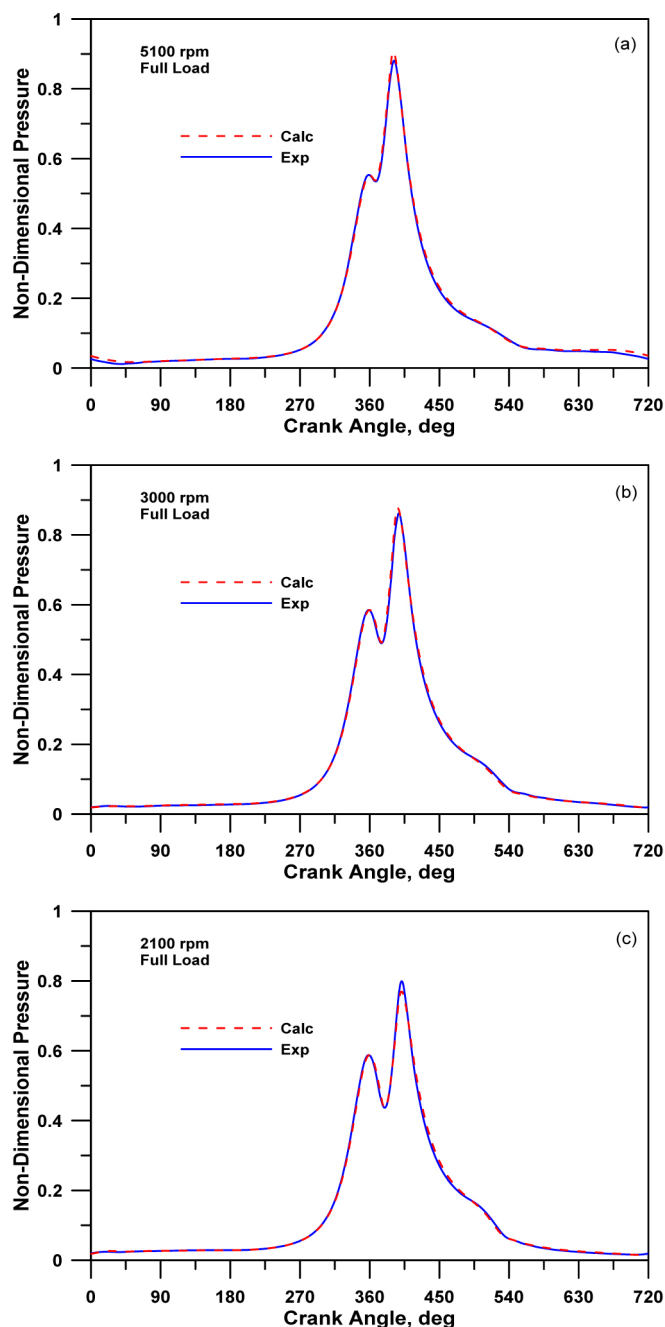


Figure 3. Dimensionless Cylinder Pressures at 5100 (a), 3000 (b) and 2100 (c) rpm, full load