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Technical Paper

Influence of the manufacturing process tolerance on the swirl number of a low-capacity engine

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ABSTRACT

We analyze the fluid-dynamic behavior of the cylinderhead of a low-capacity engine in the steady regime. Attention is paid to the influence of the manufacturing process tolerance on the variability of the swirl number generated by the intake port. Both the discharge coefficient and swirl number are measured for a number of cylinderheads manufactured in series. While the discharge coefficient values lie within the range specified by the manufacturer, most of the swirl number values fell outside that range. More importantly, the variability of the swirl number exceeds the prescribed limit. The influence of the manufacturing process tolerance on that variability is examined from numerical simulations. Specifically, we consider the position of the intake port with respect to the machined elements of the cylinderhead. The variation of that position in the range given by the manufacturing tolerance explains in itself the scattering of the swirl number observed in the experiments. To reduce the swirl number variability, we propose a procedure which combines advanced manufacturing techniques and fluid-dynamic simulation.

1. Introduction

The internal combustion engine has undergone a continuous evolution since its invention in 1876. This evolution has affected all types of engines and applications. In the last two decades, special attention has been paid to the reduction of both the emitted pollutants and fuel consumption. These two aspects of the problem are greatly influenced by the admission phase of the charge renovation process. The air flow entering the chamber leads to correct combustion depending, to a great extent, on the cylinderhead shape

The experimental and numerical studies on the flow across cylinderheads can be categorized into two major classes: (i) those considering the air motion across the cylinderhead in an isolated way [1–5], and (ii) those which analyze the whole fluid-dynamic process in the engine. In the first group, one measures the discharge coefficient and the swirl/tumble number characterizing the steady flow driven by a constant pressure drop applied to the intake port. The second group focuses on both the influence of the combustion chamber geometry and the timing of the piston/valves motion on the unsteady flow caused by that motion [6–12,4,13,14].

Payri et al. [6] examined how the combustion chamber shape affects the entering flow both numerically and with laser Doppler anemometry. Pastor et al. [15] made use of a similar methodology to study the steady regime. Schogl et al. [3] and Jawad and Arslan [5] analyzed also numerically and experimentally the steady and unsteady flows across the intake port of a gasoline engine. Dembinski and Angstrom [7] studied the flow in a combustion chamber from both Particle Image Velocimetry (PIV) and numerical simulations, while Sushma and Jagadeesha [10] examined numerically the effects of the combustion chamber geometry on the unsteady flow pattern. The dependence of the cylinderhead shape on the mixing of high-viscosity fuels was analyzed by Saad and Bari [13]. The maximization of the swirl and tumble numbers in low-capacity engines has been considered in several studies [8,16,17,14]. For instance, Costa et al. [16] measured those quantities with the PIV technique in the steady regime. You et al. [17] applied the Design of Experiment (DoE) procedure to optimize the intake port shape of a low-capacity engine.

Despite the overwhelming volume of publications on internal combustion engines, the influence on the flow variability of the manufacturing process tolerance has been hardly studied. The effect of the deviation of the cylinderhead shape on the swirl

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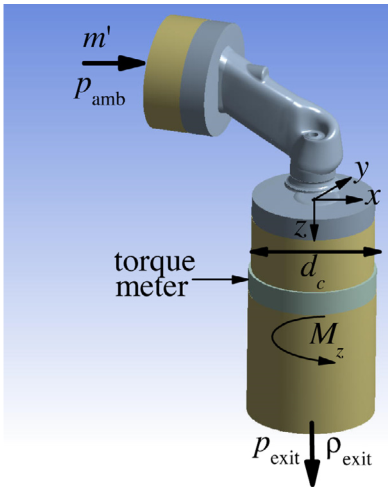

Fig. 1. Sketch of the problem.

number has been considered only in a few works. Shuliang et al. [18] defined three types of deviations owing to the foundry and machining processes, and quantified experimentally the corresponding swirl number variations. Zhang et al. [19] analyzed the most common deviations produced by the foundry process, and determined numerically their fluid-dynamic effects. Mi [20] examined how the flow is affected by the deviation of the intake port position with respect to its nominal value. Finally, Lu et al. [21] found variations of the swirl number of up to 20% associated with the tolerance of both the foundry and machining processes. They also determined the tolerance range for those variations to be less than 10%.

In this paper, we will analyze the influence of the manufacturing process tolerance on the steady flow across the cylinderhead of a low-capacity engine fabricated by Deutz. First, we will measure the variations of both the discharge coefficient and the swirl number for the 4 cylinders of 19 cylinderheads manufactured in series. Second, we will identify four geometrical parameters whose variations may be responsible for the scattering of the swirl number to a great extent. Third, we will simulate the flow across the cylinderhead for different values of those parameters. Following the DoE methodology, we will determine their influence on the swirl number. In this way, we will demonstrate that the variability of the identified parameters associated with the tolerance of the manufacturing process explains the scattering of the swirl number values.

The paper is organized as follows. The problem is formulated in Section 2. We describe the experimental and numerical methods in Sections 3 and 4, respectively. The results are presented and discussed in Section 5. The paper closes with some conclusions in Section 6.

2. Formulation of the problem

We will consider a diesel, direct-injection, and inline 4-cylinder engine, whose nominal speed, torque, and power are 2600 rpm, 350 Nm, and 74.9 kW, respectively. The cylinderhead has two valves per cylinder. The cylinder diameter and stroke are 96 and 125 mm, respectively.

The problem formulation is sketched in Fig. 1. The air enters the cylinderhead through the intake port directly from the ambient. The cylinderhead incorporates the inlet and exhaust valves. In our study, the exhaust valve is completely closed, while the inlet valve lift can take the values 2, 4, 6, 7, 8, 10, and 12 mm. The lower part of the cylinderhead is a chamber which consists of two elements. The upper one is a cylinder d_c = 96 mm in diameter and 70 mm in length, and represents the upper part of the engine cylinder. The lower one is another cylinder 100 mm in diameter and 126 mm in length whose upper component is a torque meter 20 mm in thickness. The equivalent roughness k_s = 0.2 mm of the inner solid surface is assumed to be constant. We will take the upper cylinder diameter d_c as the characteristic length of the problem.

The steady flow across the cylinderhead is driven by a constant pressure drop $\Delta p = p_{amb} - p_{exit}$, where p_{amb} and p_{exit} stand for the ambient pressure and the pressure at the chamber exit, respectively. The rest of the hydrodynamic variables characterizing the flow are the density ρ_{exit} at the chamber exit, as well as the air dynamical viscosity μ and specific heat coefficients c_p and c_v. We will measure the air mass flow rate

$$m' = \int_S \rho\, v_z\, dS \tag{1}$$

crossing the cylinderhead, and the axial flux of angular momentum

$$M_z = \int_S \rho\, r\, v_\theta\, v_z\, dS \tag{2}$$

at the entrance of the torque meter. In these expressions, ρ, v_θ, and v_z are the density, and the angular and axial velocity components, respectively, while S is the cylinder cross-section at the torque meter entrance. In addition, r is the radial coordinate.

Let us define the discharge coefficient (also referred to as the flow number) $\alpha \equiv 100 \times m'/m'_0$, where

$$m'_0 = \frac{\pi d_c^2}{4}\left\{\frac{2\gamma p_{amb}\rho_{exit}}{\gamma-1}\left(\frac{p_{amb}}{p_{exit}}\right)^{1/\gamma}\left[\left(\frac{p_{exit}}{p_{amb}}\right)^{2/\gamma}-\left(\frac{p_{exit}}{p_{amb}}\right)^{(\gamma+1)/\gamma}\right]\right\}^{1/2} \tag{3}$$

is the mass flow rate crossing an infinitely thin circular orifice in the inviscid regime [22], and $\gamma = c_p/c_v$ the adiabatic constant. For a fixed geometry (including the inlet valve lift), the discharge coefficient obeys to the formal relationship $\alpha = \alpha(\mathrm{Re}, \gamma, k_{sr})$, where $\mathrm{Re} = \rho_{exit} v_c d_c/\mu$ is the Reynolds number, $v_c = 4m'/(\pi d_c^2 \rho_{exit})$ is the characteristic velocity, and $k_{sr} = k_s/d_c$ is the relative equivalent roughness. Analogously, the swirl number $\mathcal{S} \equiv M_z \rho_{exit} d_c/(2m'^2)$ can be expressed as $\mathcal{S} = \mathcal{S}(\mathrm{Re}, \gamma, k_{sr})$. The relative equivalent roughness and the adiabatic constant are kept constant both in the experiments and numerical simulations. The influence of the Reynolds number becomes negligible for large enough values of this parameter. Therefore, the discharge coefficient α and the swirl number $\mathcal{S}$ essentially depend on the shape of the cylinderhead and the valve lift. In order to characterize the fluid-dynamic behavior of the cylinderhead, independently of the valve lift, one defines the weighted average value [2]

$$\langle \mathcal{S} \rangle = \frac{\sum_{i=1}^{N} \alpha_i \mathcal{S}_i}{\sum_{i=1}^{N} \alpha_i}\ , \tag{4}$$

where the subindex i stands for each of the N = 7 valve lifts considered in our analysis. The manufacturer has established the nominal values α = 7.8 and $\langle \mathcal{S} \rangle = 0.51$ for the discharge coefficient and swirl number, respectively. In both cases, the allowed variability is ±5% of the nominal value.

3. Experimental method

The test bench FTB2000 CE (Avl Tipelmann) was used to measure both the discharge coefficient α and the swirl number $\mathcal{S}$ as a function of the inlet valve lift in the steady regime. The bench produces a constant pressure drop by suctioning air from the ambient with a fan whose speed can be controlled. The air stream circulates through the intake port and enters the upper cylinder. A torque meter is connected to that cylinder at a distance of 70 mm from the valve seats. This device determines the axial flux of angular momentum M_z by measuring the torque necessary to straighten the