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Frequency based approach for simulating pressure waves at the inlet of internal combustion engines using a parameterized model

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HIGHLIGHTS

- ▶ Transfer function technique for engine intake wave action simulation.
- ▶ Frequency domain characterization of dynamic pressure using shock tube experiments.
- ▶ Simulink and GT-Power coupling using transfer function methodology.
- ▶ Parameterized analytical model depending on tube geometry for dynamic pressure.
- ▶ Intake pressure simulation.

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ABSTRACT

Today's downsized turbocharged engines mainly focus on improving low end torque and increasing mass flow rate, this is done in order to improve the overall thermodynamic efficiency of the engine and to gain a lower BSFC. An integral part of any combustion engine is the air intake line that has a first order effect on engine filling and emptying. The wave action that takes place is usually simulated using one-dimensional codes. This paper presents a novel technique based on a frequency domain characterization of the intake line. A link over a wide frequency spectrum is identified between the instantaneous mass flow at the valve and the dynamic pressure response. This model is implemented into Simulink via a transfer function and coupled to GT-Power to produce an engine simulation. A shock tube experimental campaign was conducted for a number of tubes with varying lengths and diameters. The parameters of this transfer function are measured for each case then combined with gas dynamic theory and a frequency analysis to identify a law of behavior as a function of pipe geometry. The final model is validated on a single cylinder engine in GT-Power for a variety of pipe geometry.

1. Introduction

The air intake line of an internal combustion engine is the perfect application of unsteady fluid dynamics. The reciprocating motion of the intake valves causes pressure pulses to be generated that propagate in the piping system. In the case of a multi-cylinder engine for a constant engine speed, pressure bursts from different cylinders establishes a system of standing waves that affect the emptying, filling and scavenging processes of the engine. The volumetric efficiency and the output torque of the engine are thus influenced by this wave action, which is turn dependant on two factors: the excitation of the engine (at the valves interface) and the geometry itself of the intake line. This subject have been studied and analyzed for many years. Broome [1] distinguished between inertial and wave effects, that when combined together with a proper valve timing induces ram effect. On the other hand, the design of the engine manifold itself requires being able to calculate the unsteady compressible flow of the air flowing through the engine intake and exhaust. Nowadays, this is possible using one-dimensional calculation codes that discretize the continuous ordinary differential equations of gas dynamics in the space and time domains [2].

Winterbone and Pearson [3] give an excellent review of the current techniques and numerical schemes employed in 1D codes, these have become the benchmark for engine simulation in the automotive industry. However, experimental engine tests are a must in order to calibrate different parameters such as pressure loss coefficients. Also the dimensions of the modeled geometries in the case of a complex part often differ from their physical values

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Nomenclature

BSFC	brake specific fuel consumption
c_0	local speed of sound (m/s)
C_1, C_2	propagation constants (mbar)
F	force of air mass (N)
FFT	fast Fourier transform
f_i	frequency of the *i*th mode (Hz)
Δf_i	resonant mode *i* bandwidth (Hz)
G	function found by curve fitting
i	harmonic frequency number
j	complex constant ($j^2 = -1$)
L	physical length of pipe (m)
L^*	corrected length of pipe (m)
ΔL	end correction (m)
loss_{coef}	Pressure loss coefficient (mbar/kg h)
p	relative pressure (mbar)
$P(s)$	Laplace of pressure (mbar)
p_0	initial pressure at rest (mbar)
p_{absolute}	absolute pressure (mbar)
p_{loss}	steady pressure losses (mbar)
Pr	Prandtl number
Δp	pressure rise hammer effect (Pa)
qm	mass flow (kg/h)
$Qm(s)$	Laplace of mass flow (kg/h)
s	Laplace variable ($s = j\omega$)
S	cross section of the tube (m^2)
t	time (s)
TF	transfer function (mbar/kg h)
u	velocity (m/s)
Δu	velocity rise hammer effect (m/s)
X_i	inertial parameter for the *i*th mode (1/m)
X_{in}	total inertial parameter (1/m)

Greek letters

ω	angular velocity (rad/s)
ε	damping parameter
ρ	air density (kg m^{-3})
γ	ratio of specific heats (1.4)
φ	momentum (kg m/s)
α	attenuation coefficient (Np/m)

Subscripts

0	Initial state
absolute	absolute pressure
maximum	maximum pressure
loss	constant pressure drop

in order to reach an acceptable level of accuracy. Also, the time spent on a complete 1D engine model is not negligible, not only in terms of simulation times for a good convergence, but also the time needed to set up experiments, measure data, do the post-treatment and model again.

Another way to investigate wave action at the intake is to employ frequency based analysis techniques that make use of reciprocating nature of the engine to access the frequency content and characterize a given system. Ohata and Ishida [4] and Matsumoto and Ohata [5] used analytical acoustic equations to characterize the behavior of the propagating waves at the intake written in terms of the Laplace variable. Using simple models for ducts, volumetric efficiency curves could be deduced as well as self-induction and interference effects of the different cylinders. This meant that such techniques can be used successfully to evaluate the unsteady behavior of the intake system. Harrison and Stanev [6] presented a technique based on wave decomposition methods using a pair of distant pressure sensors to evaluate forward and backward pressure components. This lead to the calculation of the reflection coefficient at the intake of a single cylinder engine and highlighted the fact that wave action at the intake, despite the large amplitudes of pressures, can be considered linear in nature. Concretely, this means that transfer function models which are linear parameterized models are quite capable of reproducing pressure traces.

Desmet [7] derived an analytical relationship that links the air speed entering a combustion chamber to the pressure fluctuations knowing the geometry and speed of the piston. This equation was used to highlight the interest of delayed intake-valve closing, this is due to the inertia of the air mass in the intake, known as the inertial ram effect. Acoustic methodology was developed by Harrison et al. [8,9] where a linear acoustic model was presented for studying wave action at the intake. This differs from acoustics dedicated to radiated noise because the frequency range of interest is much lower and the wave propagation itself is coupled to inertial effects especially at the primary runners. The acoustic model considers the intake valve and throat area as the acoustic source and uses impedance characterization and mass flow through the intake valve to calculate acoustic pressure fluctuations. It was found that an inertial effect parameter [10] must be added to the modeling in order to obtain correct amplitudes. The following work is based on identifying a transfer function that similarly uses mass flow through the intake valve to calculate pressure response; however it incorporates a ram parameter that takes into consideration inertial effects. This transfer function can be derived from a second order differential equation that characterizes pressure response as a function of the excitation mass flow.

Fontana and Huurdeman [11] deduced such an equation of pressure and mass flow. They based their equation on an analogy between the column of air existing in the intake of an engine and electric RLC circuits. Chalet et al. [12] employed a similar equation by making an analogy with a mechanical mass-spring-damper oscillatory system. This equation can be written in the form given by Eq. (1). This equation contains different parameters such as the relative pressure p, the transient mass flow excitation qm, the angular frequency ω, the damping parameter ε and the inertial parameter X_{in}.

$$\frac{1}{\omega^2}\frac{d^2p}{dt^2} + 2\frac{\varepsilon}{\omega}\frac{dp}{dt} + p = X_{in}\frac{dq_m}{dt} \tag{1}$$

The relative pressure response p is given by the following equation

$$p_{\text{absolute}} = p_0 + p_{\text{loss}} + p \tag{2}$$

where p_0 and p_{loss} are respectively the initial pressure at equilibrium and pressure drop corresponding to a mean flow. Fontana and Huurdeman [11] proposed a new technique for engine simulation, it relies on a link between pressure and mass flow rate in the frequency domain. This was achieved by applying a Laplace transformation on Eq. (1) and arranging terms, Eq. (3) is thus obtained

$$TF(s) = \frac{P(s)}{Qm(s)} = \frac{X_{in}s}{\left(\frac{s}{\omega}\right)^2 + \frac{2\varepsilon}{\omega}s + 1} \tag{3}$$

$P(s)$ and $Qm(s)$ are respectively the Laplace transform of the pressure and mass flow. The transfer function $TF(s)$ in Eq. (3) is identified for a given geometry following an impulse excitation of mass flow. This is possible thanks to a unique test rig called the dynamic flow bench: the considered engine intake line is mounted on the