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Control-oriented aspirated masses model for variable-valve-actuation engines

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ABSTRACT

Modelling the cylinder filling process on variable-valve-actuation equipped spark-ignition engines is addressed in this paper. It is developed using first principles equations for the aspirated gas masses. These analytic equations include cylinder-peripheral thermodynamic conditions along with the intake and exhaust valve lift histories. Following a general formulation of the breathing phenomenon, a versatile model is proposed for any type of variable valve gear device. Two particular cases of variable-valve-timing and camless equipped engines are detailed. The model is calibrated using experimental data obtained at test bench. For both cases, good representativeness of the model shows the relevance of the approach.

1. Introduction and motivations

Internal combustion engines found in cars most commonly use a four-stroke cycle. The four strokes refer to intake, compression, combustion and exhaust strokes occurring during two crankshaft rotations (see Fig. 1). First, fresh air coming from the intake manifold is aspired into the cylinder through the intake valve. Fuel is then injected (for direct injection engines) and the piston compresses the fuel–air mixture during the compression stroke. Combustion appears and pushes the piston through the power stroke. Finally, burned gases are expelled towards the exhaust manifold through the exhaust valve.

Conventional engines are designed with fixed mechanically actuated valves (such as illustrated in Fig. 1). The position of the crankshaft and the profile of the camshaft determine the valve events (i.e. the timing of the opening and closing of the intake and exhaust valves). As valve motion is the same for all operating conditions, this represents a significantly costly trade-off in engine design (Leone, Christenson, & Stein, 1996; Sandquist, Wallesten, Enwald, & Stromberg, 1997; Stein, Galietti, & Leone, 1995). However, the ideal scheduling (in terms of consumption, emissions, and combustion stability) of the valve events greatly differs between different operating conditions (e.g. at idle, partial load or high load). Variable-Valve-Actuation (VVA) devices have then been introduced since the early 90's to optimise the engine behaviour over its operating range. These modify the valves' lift profiles. Engines equipped with VVA can be categorised by their means of actuation: cam-based and electromagnetic-based actuators.

Cam-based actuators can be classified into discretely staged cam-profile switching systems (Hatano, Iida, Higashi, & Murata, 1993), Variable-Valve-Timing (VVT) systems and continuously variable cam-profile systems (Gray, 1989). One of the applications given in this paper concerns the second technology. VVT allows the change to the valve timings, but not the valve lift profiles and durations themselves. The camshafts can only be advanced or delayed with respect to its neutral position on the crankshaft. VVT can be controlled by a hydraulic actuator called a cam phaser (Asmus, 1991). Engines can have a single cam phaser (intake cam only) or two cam phasers (both intake and exhaust cams) as pictured in Fig. 2(a).

Electromagnetic-based actuators permit to replace the camshaft by electromagnetic valve actuators (Frederic, Picron, Hobraiche, Gelez, & Gouira, 2010; Picron, Postel, Nicot, & Durieu, 2008). A valve control unit drives the actuators in order to open and close the valves at a given fixed lift and transition time. The main function of the system is to independently open and close the valves at the prescribed crankshaft angle timing. Fig. 2(b) presents the valves' lifts of an intake camless equipped engine.

In this paper, turbocharged equipped spark-ignition (SI) engines are considered. For this class of engines, VVA devices permit to improve efficiency and performances to the detriment of cylinder filling process simplicity. This impact is twofold.

Firstly, modifying the intake valve closing angle permits to modulate the effective cylinder capacity, and, so, the admissible air volume. This leads to a consumption decrease by minimising the pumping losses[1] at low load. At high load, it permits to take advantage of the ram effect to maximise the fresh air charge.

Secondly, modifying the valve overlap (time when both valves are open together) permits an internal gas recirculation. This

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[1] Work transferred between the piston and the cylinder gases during the intake and exhaust strokes (Heywood, 1988).

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Nomenclature

Area	common opening area of both valves, m^2
IVC	intake valve closing angle, deg
IVO	intake valve opening angle, deg
m^e_{asp}	mass aspirated coming from the exhaust, kg
$\dot{m}^e_{asp}$	mass flow through the exhaust valve, kg/s
m^i_{asp}	mass aspirated coming from the intake, kg
$\dot{m}^i_{asp}$	mass flow through the intake valve, kg/s
m_c	cylinder mass at *ivc*, kg
m^e_{evc}	exhaust aspirated mass, kg
m_{MAF}	aspirated mass from the MAF sensor, kg
$\dot{m}_{MAF}$	aspirated mass flow from the MAF sensor, kg/s
$m^e_{overlap}$	mass of gas flowing through both valves, kg
n_{cyl}	number of cylinders, dimensionless
N_e	engine speed, rpm
OF	overlap factor, m^2 deg
P_c	cylinder pressure at *ivc*, Pa
P_e	exhaust manifold pressure, Pa
P_{evc}	cylinder pressure at *evc*, Pa
P_i	intake manifold pressure, Pa
r	ideal gas constant, J/kg/m^3
T_c	cylinder temperature at *ivc*, K
T_e	exhaust manifold temperature, K
T_{evc}	cylinder temperature at *evc*, K
T_i	intake manifold temperature, K
V_d	displacement volume, m^3
V_{evc}	cylinder volume at *evc*, m^3
V_{ivc}	cylinder volume at *ivc*, m^3
γ	specific heat ratio, dimensionless
δ_{ivc}	correcting term due to ram effect, deg
η_{vol}	volumetric efficiency, dimensionless
Φ_e	exhaust valve timing position, deg
Φ_i	intake valve timing position, deg

Abbreviations

BDC	bottom dead center
degATDC	degree after top dead center
degBTDC	degree before top dead center
ECU	electronic control unit
evc	exhaust valve closing
ivc	intake valve closing
ivo	intake valve opening
MAF	mass air flow
NO_x	nitrogen oxides
SI	spark-ignition
TDC	top dead center
VVA	variable-valve-actuation
VVT	variable-valve-timing

phenomenon depends on the engine operating condition since it is a function of the pressure difference between the intake and exhaust manifolds. Under partial load, it permits an internal exhaust gas recirculation that leads to a consumption and Nitrogen Oxides (NO_x) emissions reduction (Heywood, 1988; Leroy et al., 2009). At high load, increasing the valve overlap permits some fresh gases to flow through the cylinder, driving out the residual burned gases. This is the scavenging effect that allows an earlier turbine initiation. This contributes to increase the engine load at low engine speeds (Kleeberg, Tomazic, Lang, & Habermann, 2006; Petitjean, Bernardini, Middlemass, & Shahed, 2004).

In a control standpoint, one basic task consists of managing engine torque output according to the driver's requests, while limiting pollutant emissions. On SI engines, torque control is achieved by

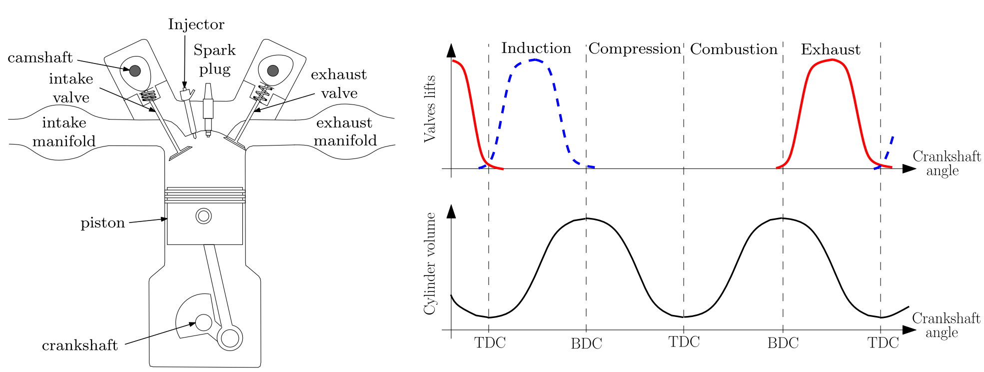

Fig. 1. Engine components around the combustion chamber and phases of the four-stroke cycle. Intake (dashed) and exhaust (solid) valve lifts are given as a function of the crankshaft angle. BDC and TDC refer to bottom dead center and top dead center, respectively.

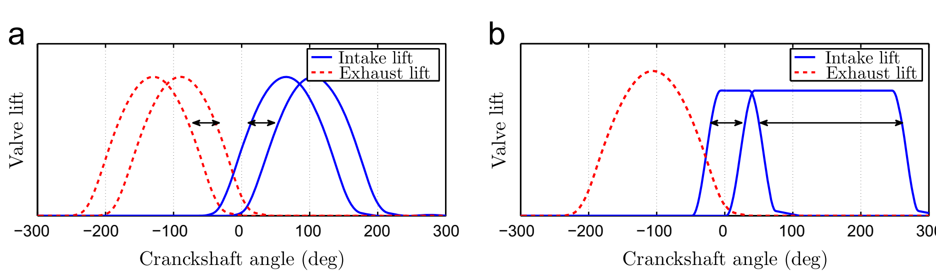

Fig. 2. Two examples of VVA technologies: intake and exhaust variable valve timing (VVT) and half-camless. (a) Dual independent VVT. Valve timing actuators can move separately. (b) Camless actuator controls independently the intake valve opening and closing.