

The Performance Characteristics of an Production Oriented Air Hybrid Powertrain

2010-01-0821

Published
04/12/2010Cho-Yu Lee, Huz Zhao and Tom Ma
Brunel University

Copyright © 2010 SAE International

ABSTRACT

In a previous paper [1], the authors have proposed a cost effective air hybrid concept based on a proprietary intake system and cam profile switching (CPS) system [2]. It was shown through engine simulations that the pneumatic hybrid operation could be achieved with about 15% regenerative efficiency. The proposed air hybrid operation can be achieved with proven technologies and engine components and hence it represents a cost-effective, reliable and quick deployable solution for low carbon vehicles.

In this work, a four-cylinder 2 litre diesel engine has been modelled to operate on refined air hybrid engine configurations and the braking and motoring performance of each configuration have been studied.

Both air hybrid systems can be constructed with production technologies and incur minimum changes to the existing engine design. The regenerative engine braking and starting capability is realised through the employment of innovative proprietary intake system and production cam profile switching (CPS) devices. The 1st hybrid engine configuration is shown to be capable of producing compressed air to 15 bar during its compression mode operation. Both hybrid systems will allow the engine to be cranked by the compressed air at moderate pressure without using an additional starter or dedicated valves in the cylinder head. Therefore, the proposed air hybrid engine systems can be considered as a very cost-effective regenerative hybrid powertrain and can be implemented in vehicles using existing production technologies.

1. INTRODUCTION

An air hybrid powertrain can be achieved with standard vehicle transmission and would require low cost modification

to the engine. The system is able to convert braking energy into pneumatic energy of compressed air stored in an air tank during deceleration. The stored compressed air can then be reused either to start the engine in every stop/start operation for the biggest fuel saving or to be used as boost air for eliminating the turbo lag.

In order to reuse the braking energy, the vehicle operation consists of 3 basic modes, i.e. compression mode (CM), cranking mode and normal firing mode. During deceleration, the engine stops fuel injection and switches from firing mode to CM when the induced air will be compressed through a gas transfer valve into the air tank.

In many other proposed air hybrid concepts [3,4,5,6,7,8,9,10], the air hybrid operation requires the use of fully variable valve actuation systems, whether it is electro-mechanical, hydraulic, or pneumatic camless system. Tai et al. [3] modelled the use of a camless electro-hydraulic valvetrain for air hybrid operation and their results predicted the 36% round-trip efficiency of air hybrid operation and 38% reduction in fuel consumption. In order to implement the air hybrid operation in an engine, a camless system consisting of hydraulic pumps and motors and expensive hydraulic accumulators had to be used [5]. Trajkovic et al. [6] performed air hybrid engine experiments by using pneumatic valve actuators and showed that a regenerative efficiency of 40-48% could be achieved. In a recent work [7], the air hybrid operation was demonstrated in a single cylinder engine by using an additional electro-hydraulic actuated valve in the cylinder head, in a similar way to engine starting by compressed air as used in some marine applications.

In comparison, the authors have proposed more practical and cost effective air hybrid engine configurations that can be implemented with minimum change to the engine design and production ready components. The concept involves the

employment of a proprietary one way intake system and cam profile switching (CPS) devices. It is capable of regenerative compressed air production and its subsequent usage for stop-start, boosting and other applications that would require the use of pressurised air supply. In the previous paper [1], It has been shown that the proposed air hybrid engine is capable of regenerative braking and starting the engine with a regenerative efficiency of 15%. In this paper, results will be presented and discussed on the improved hybrid engine design and configurations.

2. BRUNEL 1ST AIR HYBRID ENGINE CONCEPT

2.1. CONFIGURATION

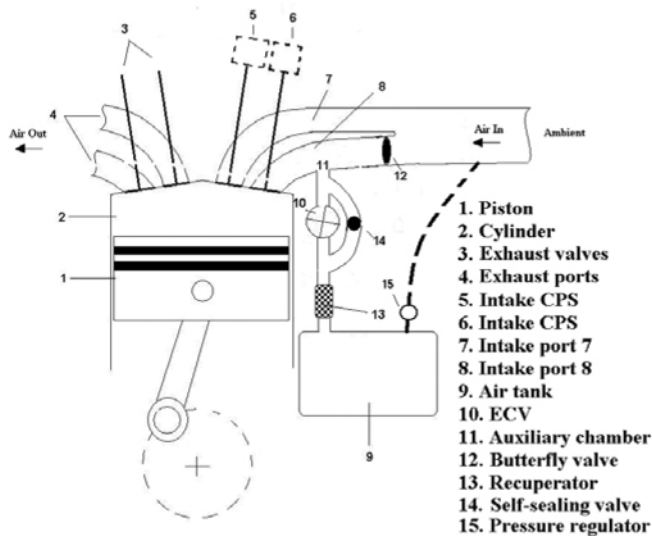


Figure 1. Brunel 1st Air hybrid engine schematic diagram

Figure 1 shows an air hybrid engine having a piston (1) reciprocating in a cylinder (2). Intake valves (5 and 6) connect the cylinder with intake ports (7 and 8 respectively) and exhaust valves (3) connect the cylinder with exhaust ports (4). Both intake valves are timed to open and close by a Cam Profile Switch (CPS) system. A port throttle valve (12) is additionally provided in the intake port 8 connected to the intake valve. The use of a throttle valve will pose minimum flow restriction and hence little impact on the WOT performance. However, the throttle valve has to be designed to withstand higher pressure without leakage.

When the engine is running in the normal firing mode, the port throttle valve remains fully open and air can be induced into cylinder through the upper intake port (7) and the lower intake port (8). For the CM and cranking mode, the port throttle valve is shut and air can be kept without escaping out of the lower intake port to atmosphere.

An air tank (9), stores high-pressure air, and is connected to the lower intake port through a recuperator (13) and an Energy Control Valve (ECV) into an auxiliary chamber. The recuperator absorbs and stores heat energy when compressed air is supplied from cylinder to air tank in CM. It also releases heat energy while releasing compressed air from air tank to cylinder in EM. Therefore, high temperature and high pressure compressed air can be reused. In the simulation, the heat transfer coefficient of the air tank is set to 0.0 and hence there is no heat loss in the air tank. Therefore, the recuperator is not modelled separately in the analysis but the compressed air will be assumed to retain their temperature in the air tank which is equivalent to 100% recuperator efficiency. A solenoid valve has been assumed to be the ECV (10) that controls air flow out of the air tank for EM. For CM operation, a check valve has been adopted to be the SSV (self-sealing valve) that controls air flow into the air tank.

In the current study of a 4-cylinder engine, the engine also includes three more cylinders, a fuel system and an ignition system which are not shown in Figure 1 for highlighting the main subject of the present study.

2.2. PRINCIPLE OF OPERATION

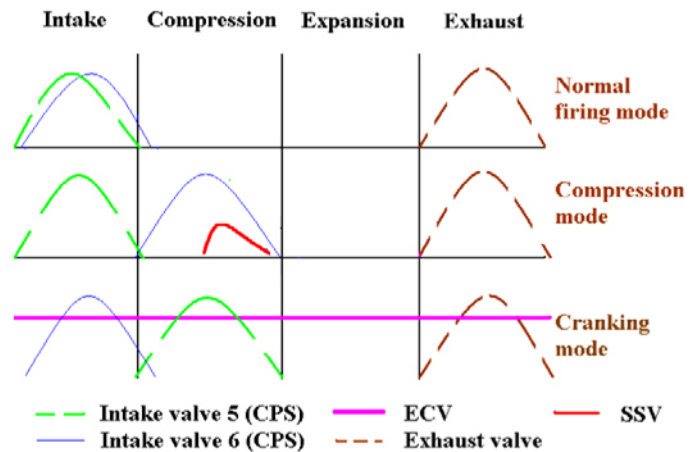


Figure 2. Engine valves, SSV and ECV timing for normal firing mode, CM and cranking mode.

Figure 2 shows the valve timing diagrams for the normal firing mode, CM and EM/cranking operation. During the normal firing mode, both intake and exhaust valves operate with their default profiles. During vehicle deceleration and braking, intake valve 5 opens normally but the opening of intake valve 6 is shifted to the compression stroke of the four-stroke cycle so that air in the cylinder is compressed and accumulated in Auxiliary chamber 11 formed by the port throttle valve, ECV and SSV so that engine braking is realised. Due to the pressure difference between the Auxiliary chamber and Air tank, SSV opens and lets compressed air into the Air tank towards the end of the compression stroke. During the cranking operation, the engine is turned into an air

expander by keeping ECV open and opening Intake valve 6 during the normal intake stroke so that high pressure air forces the piston down and generates the motoring work. Furthermore, the opening of intake 5 is moved to the compression stroke with duo purposes: (i) preventing the compressed air escaping to the intake manifold as the pressurised air expands in the cylinder during the intake stroke, (ii) minimising the parasitic compression work by venting the cylinder air to the intake manifold.

The size of the auxiliary chamber volume is an important parameter in this concept. It is determined by the positions of the port throttle valve, SSV and ECV. For the 2.0litre engine used in the current study, the engine displacement volume and clearance volume are 500cm³ and 28.7cm³ per cylinder respectively, resulting in a geometric compression ratio of 18.4:1. However, when the engine is switched to the compression mode, the actual compression ratio (R_c) decreases as the volume of the auxiliary chamber is included in the cylinders' total clearance volume. For the auxiliary chamber volume of 96.4cm³ examined below, the actual compression ratio is calculated as:

$$R_c = \frac{500 + 28.7 + 96.4}{28.7 + 96.4} = 5.0 : 1 \quad (1)$$

2.3. ENGINE DETAILS

The modelled engine was a 2.0 litre diesel engine with four cylinders and the air hybrid operation was simulated in Ricardo's WAVE software. The simulation analyzes the dynamics of pressure waves, mass flows, and energy losses in ducts, plenums, and the manifolds of the engine. Engine and valves data are given in [Table 1](#) and [Table 2](#) respectively. A 40 Litre air tank has been assumed in this study.

Table 1. Engine dimensions and characteristics

Number of cylinders	4
Cylinder bore	86mm
Piston stroke	86mm
Connecting rod length	160mm
Displacement volume	500cm ³
Clearance volume	28.7cm ³
Total volume of one cylinder	528.7cm ³
Compression ratio	18.4:1
R_c for air hybrid mode	(a) 5.0:1

Table 2. Valves dimensions and characteristics

Intake valve	2
Diameter	23.5 mm
Opening point (normal)	20° BTDC
Closing point (normal)	60° ABDC
Maximum lift	8.75 mm
Exhaust valve	1
Diameter	23.4 mm
Opening point	60° BBDC
Closing point	35° ATDC
Maximum lift	8.75 mm
port throttle valve	1
Diameter	24.9 mm
SSV	1
ECV	1

2.4. EFFECTS OF TIMING OF INTAKE VALVE 6 FOR CM

Preliminary studies have shown that for a fixed valve opening duration the closing time of the intake valve 6 has significant effect on the engine braking and compressor mode operation. In addition to producing engine braking, the priority of the compression mode operation is to maximise the recovered braking energy by capture as much air as possible. Therefore, it is necessary to evaluate the effect of Intake valve 6 closing timing on engine braking torque and its optimisation to accumulate the most compressed air into the air tank in every deceleration.

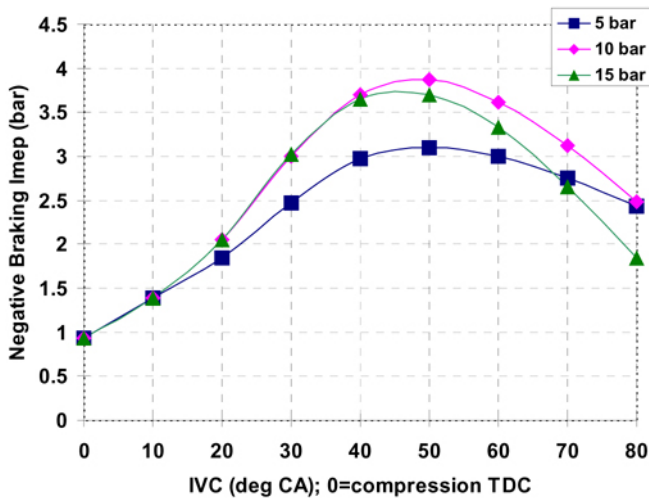


Figure 3. Predicted $imep_b$ for 1500 rpm engine speed

Figure 3 shows the predicted variation of negative indicated braking $imep_b$ values as a function of inlet valve (IV) 6 closing points at tank pressures of 5 bar, 10 bar and 15 bar, respectively. It can be seen from the results that high braking performance is achieved for a range of IV 6 closing timings between 40° - 60° CA ATDC. $imep_b$ drops rapidly if the valve timing is shifted out of the above range. The braking $imep_b$ value peaks at 50° CA, regardless of the tank pressure. It should note that the actual braking torque exerted to the vehicle will include the frictional torque and hence the actual braking $bmep_b$ will be typically 1 bar above the values shown in Figure 3. In addition, it is worth mentioning that the change in braking torque with valve timing can be harnessed as a means to controlling the engine braking so that appropriate mechanical and engine braking forces can be adjusted according to the vehicle's demand.

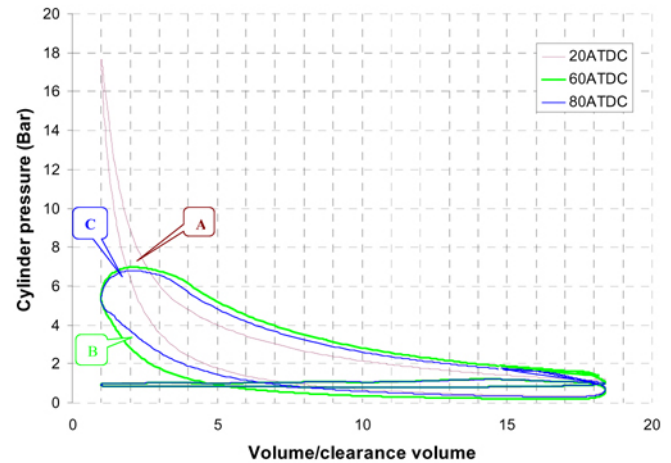


Figure 4. Indicator diagram for various IV6 closing timings

The predicted cylinder indicator diagrams for three valve timings are shown in Figure 4 at 1500 rpm engine speed and 5 bar tank pressure. In Figure 4, the highest peak cylinder pressure but the lowest area enclosed by the pressure diagram is obtained when IV6 closing point is at 20° ATDC because of its early closure during the compression stroke. The advanced closing of IV 6 reduces the amount of compressed air to be transferred to the air tank and leads to increased expansion work, hence lower engine braking work as well as less compressed air stored. In addition, for a fixed cam profile, the advancement of valve closing is accompanied by the earlier opening of the valve, which results in a positive overlap between intake valves 5 and 6. During the overlap period, some of the compressed air in the auxiliary chamber left from the previous cycle experiences short circuiting and flows back to the intake system. This phenomenon is responsible for the observed lower compression pressure associated with the advanced valve closing timing shown. For the most retarded valve close timing of 80° ATDC, the compression process starts later and some of the compressed air in the cylinder flows back into the cylinder after TDC, resulting in higher expansion pressure and hence smaller braking work (area C) as well as less air being stored in the air tank. For IVC closing at 60° ATDC, the negative engine braking work shown by area B is the biggest as it has higher compression pressure and lower expansion pressure.

Figure 5 shows the rise in the air tank pressure for 60 engine revolutions at 1500rpm as a function of IV 6 close timings at 5 bar, 10 bar and 15 bar respectively. For example, the lower curve shows that the air tank pressure increases from 5bar to 6 bar after 60 engine revolutions at 1500rpm if the IV6 closes at 30° ATDC. Irrespective of the initial air tank pressure, the most prominent rise in the tank pressure is achieved at IV6 close timing 60° ATDC and the rise of pressure is about 2bar.