

Blowdown Interference on a V8 Twin-Turbocharged Engine

2011-01-0337
Published
04/12/2011

Apoorv Agarwal, Hosuk Jung and Kevin Byrd
Ford Motor Co.

Robert A. Stein, Ahmad Kassem and Paul Whitaker
AVL Powertrain Engineering Inc.

Christian Spanner
AVL List GmbH

Copyright © 2011 SAE International

doi:10.4271/2011-01-0337

ABSTRACT

The exhaust blowdown pulse from each cylinder of a multi-cylinder engine propagates through the exhaust manifold and can affect the in-cylinder pressure of other cylinders which have open exhaust valves. Depending on the firing interval between cylinders connected to the same exhaust manifold, this blowdown interference can affect the exhaust stroke pumping work and the exhaust pressure during overlap, which in turn affects the residual fraction in those cylinders. These blowdown interference effects are much greater for a turbocharged engine than for one which is naturally aspirated because the volume of the exhaust manifolds is minimized to improve turbocharger transient response and because the turbines restrict the flow out of the manifolds.

The uneven firing order (intervals of 90°-180°-270°-180°) on each bank of a 90° V8 engine causes the blowdown interference effects to vary dramatically between cylinders. These effects are illustrated in this paper for a twin-turbocharged engine with single scroll turbochargers and log style exhaust manifolds. An AVL tool called Gas exchange and Combustion Analysis (GCA) is used to estimate residual fraction for each cylinder based on measured intake port, cylinder pressure, and exhaust port pressure profiles. The uneven firing interval causes imbalance between cylinders in residual fraction (increasing knock), fresh air (increasing CO due to uneven air-fuel ratio), and pumping work. These effects also preclude running high overlap for scavenging at low rpm, and diminish the potential benefits of dual cam phasing at part load.

The advantages and disadvantages of various methods to diminish the magnitude of these effects are investigated through 1D performance simulation and engine dynamometer testing, including the use of twin scroll turbochargers, exhaust camshafts with different exhaust valve opening timings on pairs of cylinders, and a balance tube between exhaust manifolds.

INTRODUCTION

In the light duty vehicle segment, gasoline turbocharged direct injection (GTDI) engines such as Ford Motor Company's EcoBoost™ are becoming established as a high volume technology solution for improving fuel economy. GTDI significantly increases the torque range of an engine and thereby allows broader application of a given architecture, enhancing utility and investment efficiency. Fuel economy is improved relative to larger displacement naturally aspirated engines through the benefits of downsizing and downspeeding. Additionally, stoichiometric operation allows the most stringent worldwide emissions standards to be met with an inexpensive three way catalyst (TWC), in contrast to various other technologies for improving fuel economy which require lean NOx aftertreatment.

The application of GTDI to a 90° V8 architecture imposes a number of gas exchange and combustion challenges due to the uneven firing order on each bank required for engine balance. At full load the uneven firing order results in high levels of trapped residual gas in at least two cylinders

which increases the propensity to knock and can limit performance and increase full load fuel consumption. At part load, the uneven firing order results in uneven cylinder filling leading to poor cylinder-to-cylinder air-fuel ratio distribution and higher residual levels in some cylinders which can limit variable valve timing strategies for fuel economy improvement.

While blowdown interference is present on a naturally aspirated V8 engine, it is much more problematic with a turbocharged V8 engine due to the higher BMEP levels which increase the magnitude of the blowdown pulses, the smaller volume of the exhaust manifolds which is required for fast turbocharger transient response, and the restriction to exhaust flow from the exhaust manifolds imposed by the turbines.

This paper describes the challenges associated with a V8 GTDI engine due to uneven firing order. The resulting gas exchange issues are illustrated and solutions are explored through 1D performance simulation and engine dynamometer testing.

PROBLEM DEFINITION

GTDI offers significant fuel economy benefit relative to naturally aspirated gasoline engines due to downsizing and downspeeding. Downsizing causes a smaller displacement turbocharged DI engine to operate at higher BMEP in order to match the performance of the larger naturally aspirated engine. Additional gains are achieved by downspeeding utilizing a longer final drive ratio to operate the engine at lower speeds and higher BMEP to provide the required road load power demand.

However, as these strategies force the engine to run at higher BMEP levels, design and development effort is required to reduce the engine's octane requirement to maximize compression ratio and control knock at high loads, thereby achieving maximum fuel efficiency. One area requiring careful optimization is reduction of trapped residual gas.

Trapped exhaust residual gas resulting from incomplete scavenging of exhaust gas during the exhaust stroke will raise the temperature of the inducted air charge, thereby increasing the propensity to knock due to correspondingly higher end gas temperatures. Figure 1 shows single cylinder data for a test at 2000 rpm, 18.5 bar NMEP where trapped residual levels were increased by varying the valve overlap [1]. This test was run at stoichiometric A/F ratio ($\lambda=1$) with ignition timing set at borderline detonation (BLD).

Residual levels were estimated using AVL's Gas Exchange and Combustion Analysis (GCA) system [2, 3, 4, 5], which calculates gas exchange flows into and out of the cylinder and in-cylinder residual mass fraction (RMF) using crank angle resolved intake, exhaust, and cylinder pressure measurements as input to a 1D code that runs on the indicating equipment at the engine dynamometer test cell.

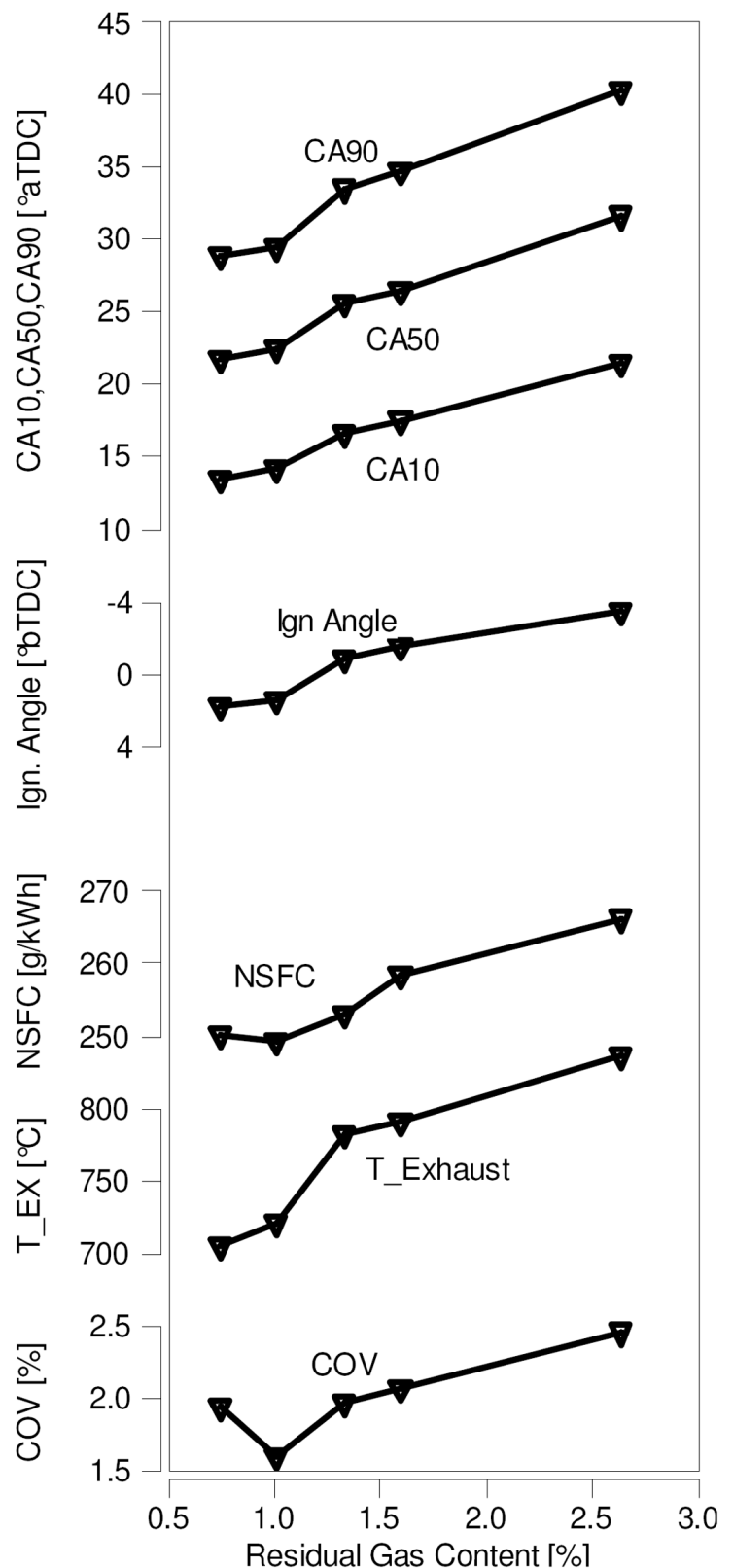


Figure 1. Impact of residual gas fraction on combustion parameters and exhaust temperature.

At this speed/load, the engine was significantly knock-limited even with minimal residual gas content. As residual levels increased, ignition timing had to be further retarded due to knock. As residual gas content increased from 1% to 2.6%, combustion phasing was retarded almost 10 additional degrees, resulting in poor combustion stability as indicated by the coefficient of variation of IMEP (COV of IMEP). Net specific fuel consumption (NSFC) increased by ~ 7% and exhaust temperature increased from 700°C to 830°C due to the later combustion phasing. At higher speed conditions, this exhaust temperature increase would translate into a further increase in fuel consumption as fuel enrichment would be required to maintain acceptable turbine inlet temperature. At these residual levels, it would be necessary to reduce compression ratio to allow more advanced combustion phasing, and this would degrade efficiency across all operating conditions.

One of the benefits of combining direct injection, variable cam timing, and turbocharging is that at low engine speeds where BMEP is typically limited by low boost pressure, increased valve overlap can be used to scavenge the cylinder of exhaust gas and increase mass-flow to the turbine increasing boost and BMEP. However, blowdown interference on the V8 engine limits the ability to scavenge, as the exhaust pressure during overlap for some cylinders is above intake pressure due to the influence of the blowdown pulse from the following cylinder in the firing order.

The overall impact of blowdown interference is summarized in the diagram of [Figure 2](#).

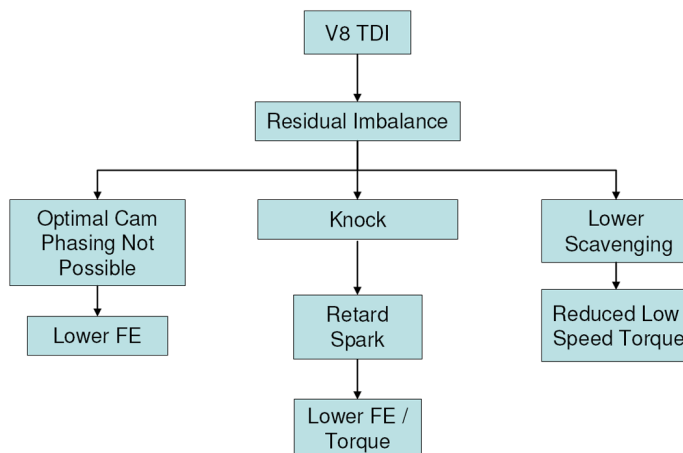


Figure 2. Impact of V8 blowdown interference.

I4 BLOWDOWN INTERFERENCE AND RESOLUTION

The issue of blowdown interference with a 180° firing interval on a turbocharged in-line four cylinder (I4) engine is well known, and is illustrated in [Figure 3](#) [6]. In this figure, red lines denote exhaust, and blue lines denote intake. The top lines of the graph shows exhaust flow out of the

cylinder and intake flow into the cylinder calculated using GCA. Shown next are the exhaust and intake camshaft profiles, and then the pressure profiles in the exhaust and intake ports.

During the first part of overlap, the intake pressure is above the exhaust pressure, resulting in positive scavenging flow from intake to exhaust. Then the pressure in the exhaust port increases above the pressure in the intake port due to the influence of the blowdown pulse from the next cylinder in the firing order. This results in backflow from the exhaust port into the cylinder, negatively impacting scavenging and increasing residual fraction, which increases the tendency to knock.

In practice, 180° blowdown interference on the I4 engine can be overcome at low engine speeds by using a short exhaust camshaft duration with a relatively late exhaust valve opening. The later opening causes the blowdown pulse from the next cylinder in the firing order to arrive later, with reduced or minimal impact on the overlap period. The short exhaust duration does cause a penalty in high speed performance. However, the benefits of allowing scavenging at low speed are very beneficial for transient response and low speed torque in the vehicle.

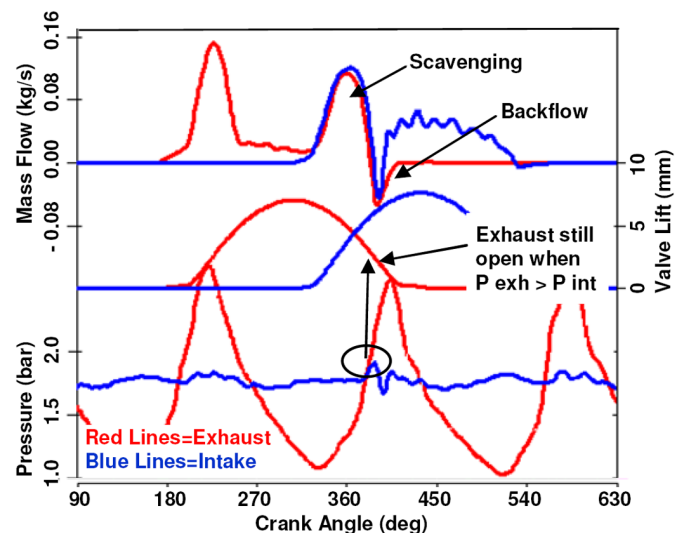


Figure 3. Blowdown interference on the I4 engine with standard exhaust duration.

V8 BLOWDOWN INTERFERENCE AT LOW ENGINE SPEED

The issue of blowdown interference on the V8 engine is more problematic than on the I4 engine because of the V8's uneven firing interval. The cylinder numbering convention of the Ford V8 is shown by the schematic in [Figure 4](#). In the schematic of [Figure 5](#) [7], the cylinders are arranged from left to right 90° apart according to the firing order of 1 5 4 8 6 3 7 2 which is indicated by the solid arrows.

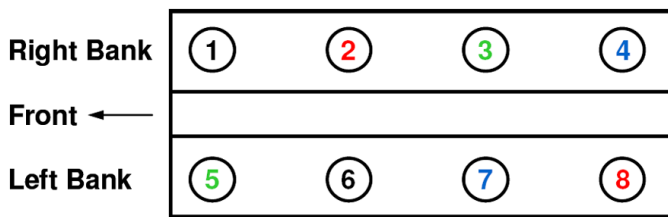


Figure 4. Cylinder numbering convention of the Ford V8.

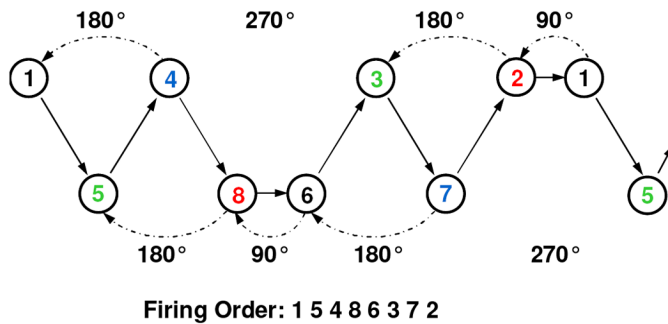


Figure 5. Illustration of the uneven firing order on each bank of the 90 degree V8 engine.

Each bank has two 180° firing intervals, one 90° firing interval, and one 270° firing interval. Blowdown interference due to the 180° and 90° firing intervals is denoted by the dotted line arrows (the blowdown pulse of cylinder 4 interferes with cylinder 1, etc.). In Figures 4 and 5, cylinder pairs (one cylinder on each bank) with identical preceding and following firing intervals are denoted by the same color: 1 & 6 (black), 2 & 8 (red), 3 & 5 (green), and 4 & 7 (blue).

Figure 6 shows the pressure measured in each cylinder's exhaust runner on the right bank of a turbocharged V8 engine at 2000 rpm, 17 bar BMEP. The pressures in the left bank manifold for paired cylinders are nearly identical. The exhaust manifolds are log style with very short runners (Figure 7) to provide for rapid turbocharger transient response. Because of the high speed of propagation of the pressure waves at exhaust gas temperature and the short travel distance in the exhaust manifold, the pressures in each exhaust runner are nearly identical at low engine speed.

In Figure 6, the location of TDC overlap is denoted for each cylinder. As shown, the blowdown pulse of cylinder 4 occurs during overlap of cylinder 1 (at 360° crank angle). Similarly, the blowdown pulse of cylinder 2 occurs during overlap of cylinder 3 (at 90° crank angle). This is termed 180° interference since the following cylinder in the firing order is 180° CA later. Also as shown, the blowdown pulse of cylinder 1 occurs during the exhaust stroke of cylinder 2. This is termed 90° interference since the following cylinder in the firing order is 90° CA later. Cylinder 4 does not suffer from blowdown interference and has low pressure during overlap since the blowdown pulse from cylinder 3 occurs 270° CA later in the firing order, and arrives after the exhaust valves of cylinder 4 have closed.

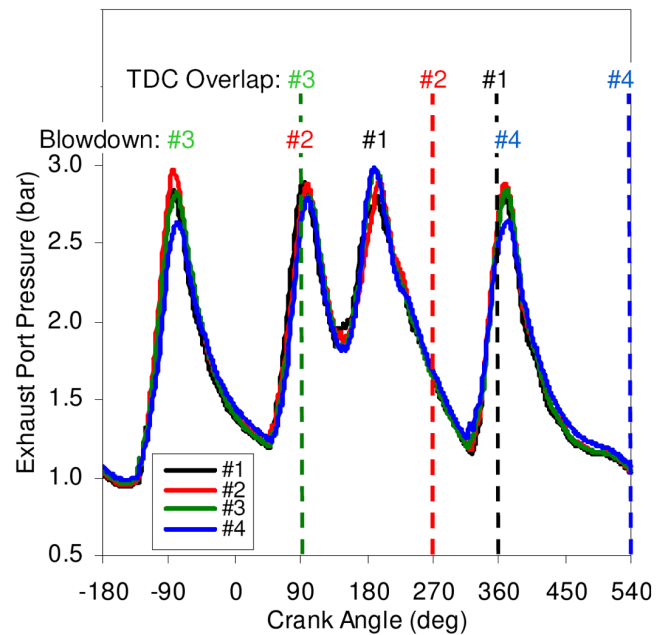


Figure 6. Right bank exhaust pressure of V8 engine at 2000 rpm, 17 bar BMEP.

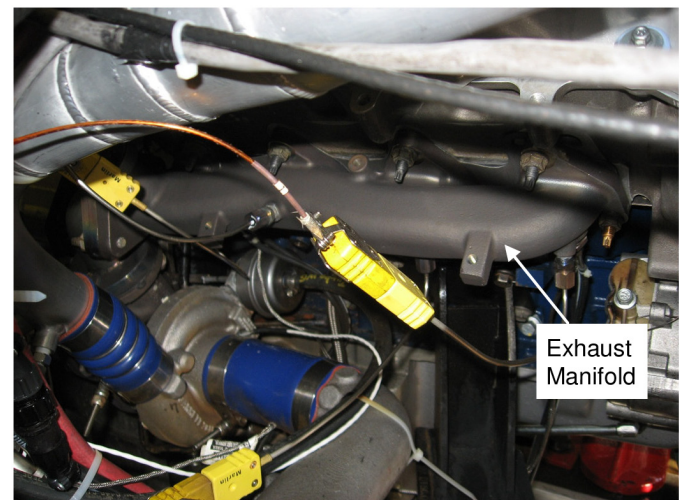


Figure 7. Single scroll turbocharger log style exhaust manifold.

Corresponding cylinder pressure pumping loops for the right bank of the turbocharged V8 engine at 2000 rpm, 17 bar BMEP are shown in Figure 8. The pumping loops for the left bank paired cylinders are nearly identical.

Cylinders 1 & 6 and 3 & 5 suffer from 180° interference from the following cylinder in the firing order, which causes a high pressure blowdown pulse during overlap at low-medium engine rpm, similar to the I4 engine. Cylinders 1 & 6 additionally suffer from a 90° preceding interval in the firing order. The latter part of the exhaust flow from the 90° preceding cylinder (2 & 8) occurs during the early part of the exhaust stroke of the cylinder of interest (1 & 6) and causes increased pressure during the exhaust stroke, increasing pumping work.

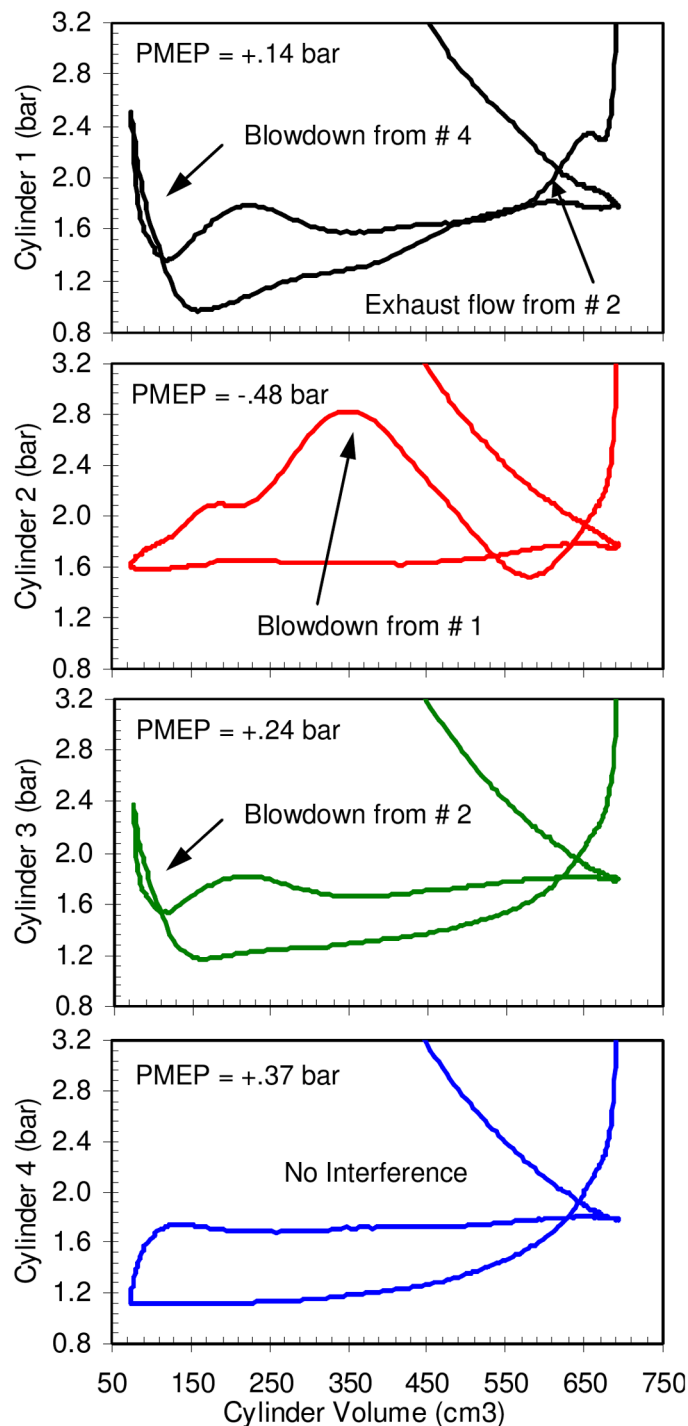


Figure 8. Cylinder pressure pumping loops for the right bank of the V8 engine at 2000 rpm, 17 bar BMEP.

Cylinders 2 & 8 suffer from 90° interference from the following cylinder in the firing order. The blowdown pulse from this cylinder (1 & 6) increases the cylinder pressure during the exhaust stroke, increasing pumping work.

Cylinders 4 & 7 do not have any issues with blowdown interference since the preceding interval is 180° and the following interval is 270°.

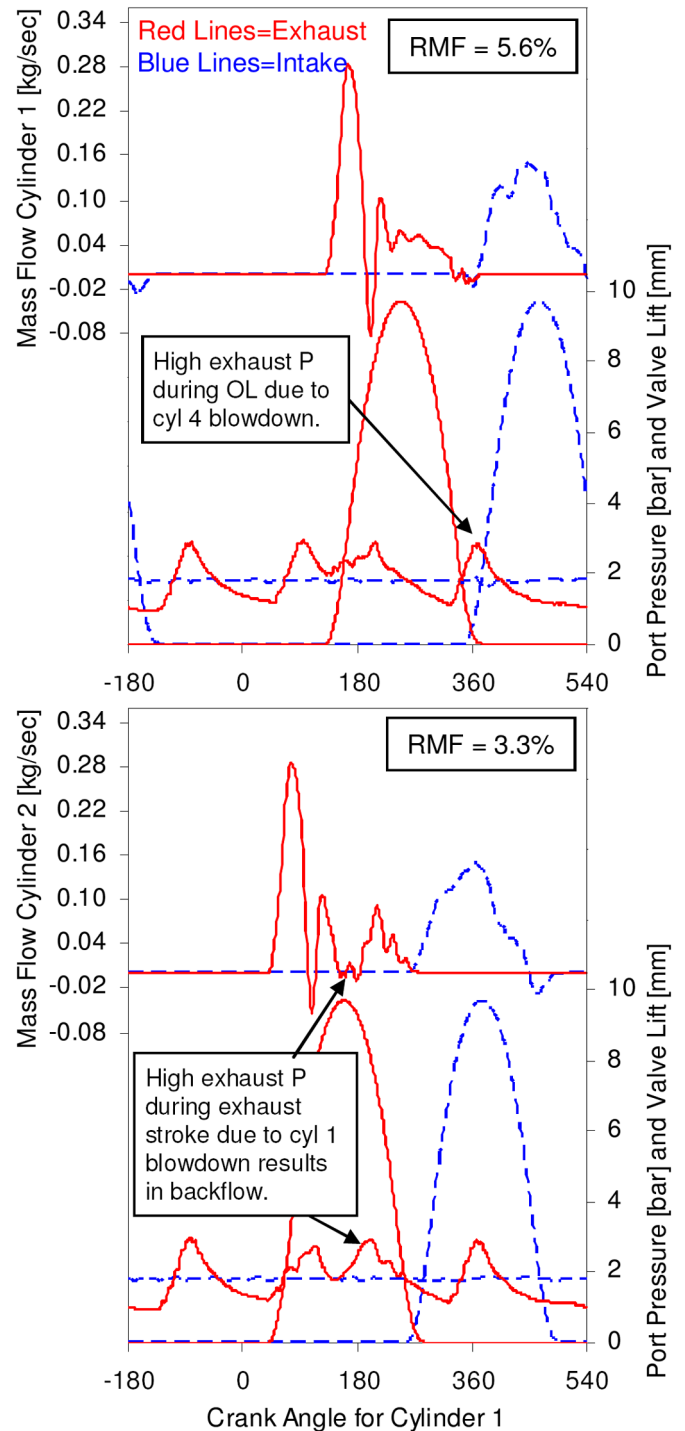


Figure 9. GCA output for 2000 rpm, 17 bar BMEP illustrating blowdown interference on cylinders 1 and 2.

Cylinders 4 & 7 have positive pumping work because intake pressure is above exhaust pressure with no blowdown interference. Cylinders 2 & 8 have negative pumping work due to the 90° interference pulse of cylinders 1 & 6. The pumping work of cylinders 3 & 5 is less positive than that of cylinders 4 & 7 due to the 180° interference pulse. The positive pumping work of cylinders 1 & 6 is further reduced due to the effect of the 90° preceding cylinder exhaust flow.

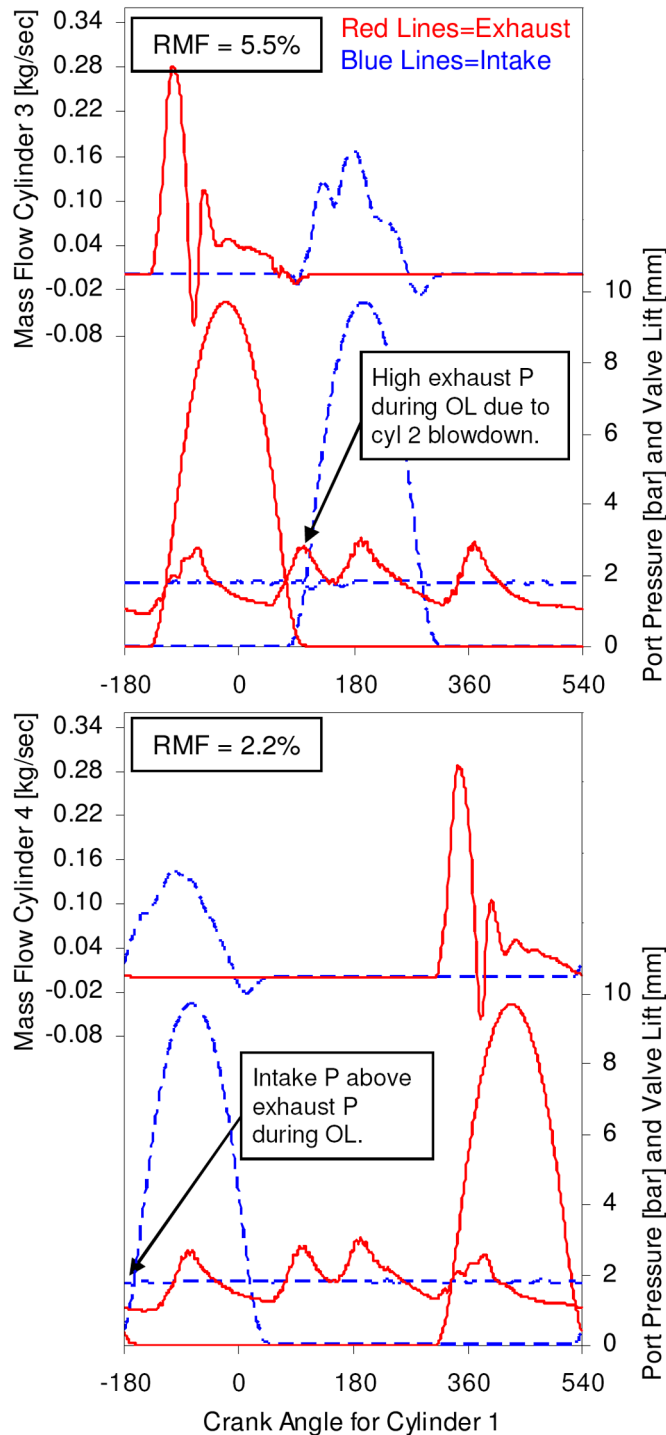


Figure 10. GCA output for 2000 rpm, 17 bar BMEP illustrating blowdown interference on cylinders 3 and 4.

As shown in the pumping loops of Figure 8, cylinders 1 & 6 and 3 & 5 suffer from high exhaust pressure during overlap due to the 180° interference. As a consequence, these cylinders will have higher residual fraction than cylinders 4 & 7. This is illustrated in the corresponding GCA output for the right bank shown in Figures 9 and 10. In these Figures, red lines denote exhaust and dotted blue lines denote intake. The top plots for each cylinder show exhaust flow out of the cylinder and intake flow into the cylinder. The bottom plots for each cylinder show intake and exhaust pressure profiles and the valve lift curves. These results were obtained at low overlap, which was used to minimize the impact of the interference. Nevertheless, the residual mass fraction (RMF) for cylinders 1 and 3 with 180° interference is about 5.5% as compared to 2.2% for cylinder 4 which does not suffer from interference.

V8 BLOWDOWN INTERFERENCE AT HIGHER ENGINE SPEED

As engine speed is increased, the pressure wave travel time in the exhaust manifold becomes significant in terms of crank angle degrees. This causes the blowdown interference pulses to arrive later at the cylinder of interest, and paired cylinders for each bank no longer behave identically. Cylinder pressure pumping loops for the right bank of the V8 engine at 4750 rpm are shown in Figure 11.

As previously explained, cylinders 4 & 7 do not suffer from blowdown interference. For cylinders 3 & 5, the 180° interference pulse from cylinders 2 & 8 arrives after exhaust valve closing at 4750 rpm and consequently the pumping loop is very similar to that of cylinders 4 & 7. For cylinders 1 & 6, the exhaust flow from the 90° preceding cylinder (2 & 8) causes increased pressure during the exhaust stroke, increasing pumping work.

Of primary interest at high speed is the effect of the 90° interference on cylinders 2 & 8. Because the 90° pulse arrives later in the cycle of cylinders 2 & 8, it not only increases exhaust stroke pumping work, but also increases the exhaust pressure during overlap, increasing residual fraction. As a consequence, cylinders 2 & 8 are more knock-limited at high engine speed than the other 6 cylinders. An overlay of the cylinder pressure pumping loops as a function of engine speed is shown in Figure 12 along with the corresponding PMEP for each cylinder at the top of the figure. The pumping loops for 2000 rpm and 4750 rpm were previously shown individually in Figures 8 and 11.