

## Combustion System Development and Analysis of a Downsized Highly Turbocharged PFI Small Engine

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### ABSTRACT

This paper provides some insight into the future direction for developing smaller capacity downsized engines, which will be needed to meet tight CO<sub>2</sub> targets and the world's future powertrain requirements. This paper focuses on the combustion system development and combustion analysis results for a downsized 0.43 liter highly turbocharged engine. The inline two cylinder engine used in experiments was specifically designed and constructed to enable 25 bar BMEP. Producing this specific output is one way forward for future passenger vehicle powertrains, enabling in excess of 50% swept capacity reduction whilst maintaining comparable vehicle performance.

Previous experiments and analysis have found that the extent to which larger engines can be downsized while still maintaining equal performance is combustion limited. Hence, small engine combustion is explored over a number of parametric studies, including a range of manifold absolute pressures up to 270 kPa, engine speeds exceeding 10,000 rev/min and compression ratios ranging from 9 to 13. Experimental results indicate that small engine combustion hurdles can be overcome to reliably extend the specific output to 25 bar BMEP. This is believed to be the highest recorded specific output for a non-intercooled small spark ignition PFI engine operating on pump gasoline. However, the boosted combustion effects illustrate that the thermal efficiency is highly dependent on the combustion efficiency, which deteriorates rapidly if uncontrolled combustion, specifically knock in the end-gas region is encountered. However, with this combustion system design strategy, potential drive cycle fuel consumption improvements in excess of 20% are still achievable.

### INTRODUCTION

Growing concerns about interruption to oil supply and oil shortages have led to escalating and unstable global oil prices.

In addition, increased public acceptance of the global warming problem has prompted car manufacturers to agree to carbon emission targets in many regions including most recently, the United States. Other legislating bodies are sure to follow this lead with increasingly stringent targets. As a result of these issues, spark ignition engines in their current form will need significant improvements to meet future requirements. One technically feasible option is smaller capacity downsized engines (~70 mm bore diameter) which operate at a higher manifold pressure for a given road load (part load pumping loss benefits) when compared to larger capacity counterparts. These small downsized engines with enhanced power could be used in the near term to reduce both carbon emissions and fuel consumption in passenger vehicles.

For downsized engines to be comparable to their larger counterparts, the specific output performance must be increased by a ratio equal to the reduction in engine size. Turbocharging seems to be the most acceptable solution to meeting the requirements, with high pressure ratios achievable and well documented improvements in efficiency [1,2,3,4,5,6].

Engines found in today's compact sized passenger vehicles are considered small, but are usually more than twice the capacity of the engine designed, developed and tested for this research. Furthermore, the technology needed to replace these larger engines with smaller capacity downsized units already exists. Applying turbocharger technology to smaller engines provides an economically viable solution in the near term to meet both the world's future powertrain and environmental requirements.

### OBJECTIVES

Previous experiments and analysis [7] have found that the extent to which larger engines can be downsized while still maintaining equal performance is combustion limited. Other hurdles such as structural engine design and turbocharger



integration were shown not to restrict engine performance for any odd firing configuration ( $\geq$  two cylinders). Hence, the objectives of this paper are to provide some insight into the relatively unknown combustion effects and limitations for smaller capacity downsized engines, and hence are as follows:

- Define and overcome combustion hurdles in smaller capacity downsized engines to reliably enable the performance limit to be extended to 25 bar BMEP.
- Define engine combustion operating limits and resulting compression ratios (CRs) for varying performance levels, corresponding to varying levels of downsizing for a reduced bore size small engine.
- Determine combustion parameters for a smaller capacity downsized engine over a range of manifold absolute pressures (MAP) up to 270 kPa, engine speeds exceeding 10,000 rev/min and CRs ranging from 9 to 13.

## EXPERIMENTAL ENGINE

The test engine used in experiments was specifically designed and developed at the University of Melbourne. Motivation behind the engine design was to provide a platform to enable the investigation of high levels of turbocharging and downsizing in a small inline two cylinder engine. As no small OEM production engine could be adapted to evaluate this concept, a four valve, inline two cylinder engine was designed and constructed to withstand the high combustion and inertia forces associated with near 300 kPa MAP and engine speeds exceeding 10,000 rev/min.

The 0.43 L twin cylinder in-line arrangement featured double overhead camshafts and four valves per cylinder. Most of the engine components were specially cast or machined from billets. Provisions in the piston design allowed for CR variations, allowing combustion system optimization. Valve motion and exhaust manifold geometry were held constant after initial optimization through modeling and experiments. The conventional PFI fuel delivery system utilized 98-RON pump gasoline, with fuel specifications previously outlined [7, 8].

Due to the engine’s original design objectives, airflow and hence power were limited by an intake flow restriction positioned upstream of the compressor. Further in-depth detail concerning the test engine is documented [7], with general specifications given in Table 1. Figure 1 highlights a design image together with a sectional view of the engine design.

## EXPERIMENTAL ENGINE RESULTS

To put the research into context, experimental engine results at the maximum specific output setting and associated CR of 10 are displayed in Figure 2, with further detail on the test engine

**Table 1. Specifications for the UniMelb ‘WATTARD’ engine.**

|                    |                                                                                                                                                    |
|--------------------|----------------------------------------------------------------------------------------------------------------------------------------------------|
| ENGINE TYPE        | UniMelb ‘WATTARD’ (Prototype)<br>In-line twin, 4 stroke SI,<br>Liquid-cooled,<br>Aluminum head/ barrel/ crankcase<br>Integral clutch/ transmission |
| CAPACITY           | 0.43 L                                                                                                                                             |
| BORE x STROKE      | 69 x 58 mm                                                                                                                                         |
| FIRING ORDER       | Unequal (0°, 180° CA)                                                                                                                              |
| COMPRESSION RATIO  | 9-13:1 with piston modification                                                                                                                    |
| COMBUSTION CHAMBER | Pent roof, central spark plug                                                                                                                      |
| VALVE ACTUATION    | 8-valve DOHC                                                                                                                                       |
| VALVE TIMING       | IVO: 24° BTDC, IVC: 72° ABDC<br>EVO: 57° BBDC, EVC: 9° ATDC                                                                                        |
| INDUCTION          | Garrett GT-12 turbocharger,<br>EMS controlled internal wastegate,<br>Intake flow restricted (Ø 20 mm)<br>Non-intercooled                           |
| FUEL DELIVERY      | Sequential multi-hole PFI                                                                                                                          |
| ENGINE MANAGEMENT  | Motec M4 EMS, M&W Pro 12 CDI                                                                                                                       |
| FUEL               | 98-RON pump gasoline                                                                                                                               |

previously published [7,8,9,10,11,12,13]. To reliably achieve 25 BMEP, the asymmetric (odd) fire configuration required the uneven flow conditions in the intake and exhaust systems to be accommodated in the engine design [12, 13]. In addition to the turbocharged results displayed in Figure 2, the test engine achieved a maximum of 37% brake thermal efficiency in normally aspirated configuration [7, 9]. However, this was only possible after CR optimization to compensate for the higher levels of dissociation, friction and heat losses associated with the smaller cylinder capacity.

Best engine performance, efficiency and CO<sub>2</sub> benefits across normally aspirated, supercharged and turbocharged modes were found to match or exceed the capabilities of typical larger bore engines found in passenger vehicles [7]. The case study performed determined the feasibility of replacing a 1.25 liter normally aspirated engine found in the 2007 Ford Fiesta with the development engine in the turbocharged mode [10]. Results show that the power and hence acceleration performance of the larger engine could be readily matched with the smaller turbocharged unit, with a 66% reduction in engine capacity. Simulated performance over the New European Drive Cycle showed a possible 22% reduction in fuel consumption and CO<sub>2</sub> emissions, including a reduction of 62% at idle conditions. These benefits are a consequence of operating the test engine closer to peak efficiency, together with engine and chassis mass reductions. The reduction in CO<sub>2</sub> would shift the study vehicle well under the 2012 Euro target of 120 g/km.

Across all modes, it was evident that the test engine (~70 mm bore) could operate with considerably higher CR for a given MAP when compared to larger bore (80-100 mm), lower speed

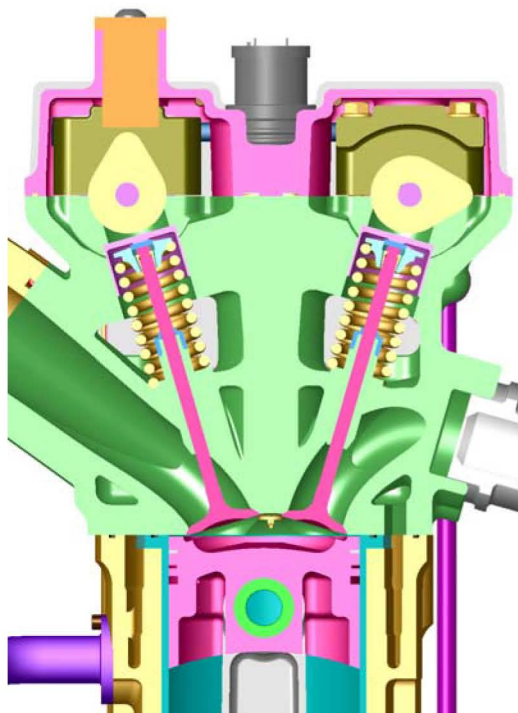
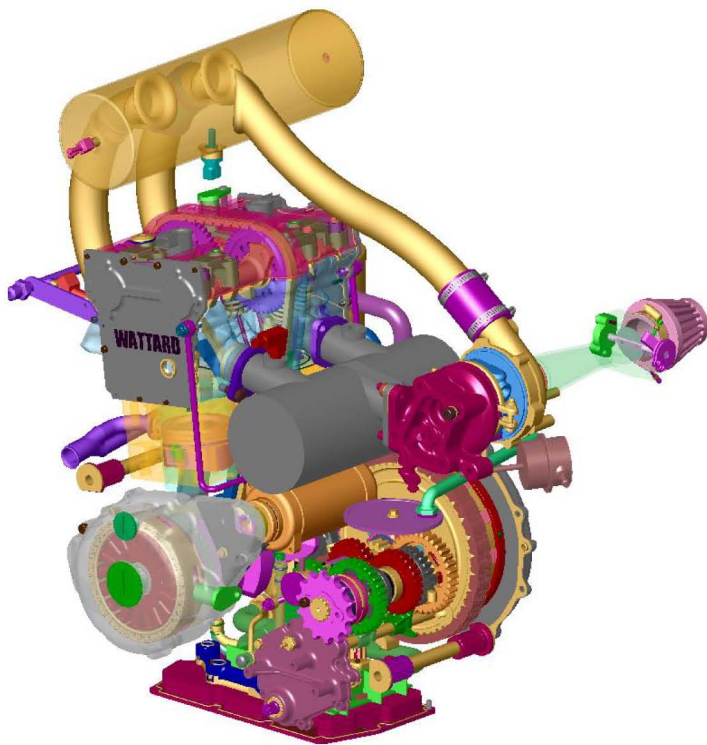


**Figure 1.** Design image of the turbocharged UniMelb 'WATTARD' engine together with a sectional view highlighting the barrel, cylinder head and reciprocating assembly.

engines. This was demonstrated with normally aspirated results showing potential for engine operation at a compression ratio exceeding 13 [9]. However, the dominant performance limiting factor was experimentally found to be abnormal combustion, specifically knock in the end-gas region, with the highest knock intensities deduced to occur on the intake side of the pent roof combustion chamber [7].

## COMBUSTION SYSTEM DEVELOPMENT

Combustion system development is outlined, including mapping various CRs versus performance levels to define engine operating limits. Furthermore, implemented strategies to improve combustion stability at mid-high engine speeds and boost levels are described. As abnormal combustion is not the focus of this paper, abnormal combustion system development and combustion analysis surrounding this phenomenon are not described in this paper. Further detail on this topic relating to this research, including combustion insight into mega knock, is available [7].

## COMPRESSION RATIO OPTIMISATION

Experimental testing included mapping various CRs versus performance levels to define engine operating limits. Performance levels included:

- 150 kPa MAP (wastegate limited),  $16.5 \text{ bar } BMEP_{MAX}$
- 200 kPa MAP (wastegate limited),  $20 \text{ bar } BMEP_{MAX}$
- Airflow limit (turbocharger, restrictor),  $25 \text{ bar } BMEP_{MAX}$

Testing commenced at the highest achievable CR with this piston design (Figure 3), corresponding to 13. When knock was encountered, the first stage of knock control relied on traditional tuning methods involving varying degrees of spark retard and/or fuel enrichment, albeit with the penalty of increased fuel consumption. If knock control could not be achieved through the EMS using these tuning strategies, the CR was then decreased in accordance with the knock severity, with performance levels subsequently retested at the reduced CR to find the limits of operation. Table 2 defines the knock limit (KL) and damage limit (DL) previously published by Rothe et al. [14], used to determine if CR reductions were needed to ensure engine reliability. The knock amplitude (KA) used to quantify limits [14], is defined as the zero to peak pressure of the high pass filtered (Butterworth) cylinder pressure (20 kHz sampling, 1.2 kHz cut-off frequency).

**Table 2.** Knock and damage limits previously defined by Rothe et al. [14] used to define engine operating limits.

|                      |                                                         |
|----------------------|---------------------------------------------------------|
| Knock Amplitude (KA) | Zero to peak pressure of the filtered cylinder pressure |
| Knock Limit (KL)     | 1% of cycles with KA > 4 bar                            |
| Damage Limit (DL)    | 1% of cycles with KA > 20 bar                           |

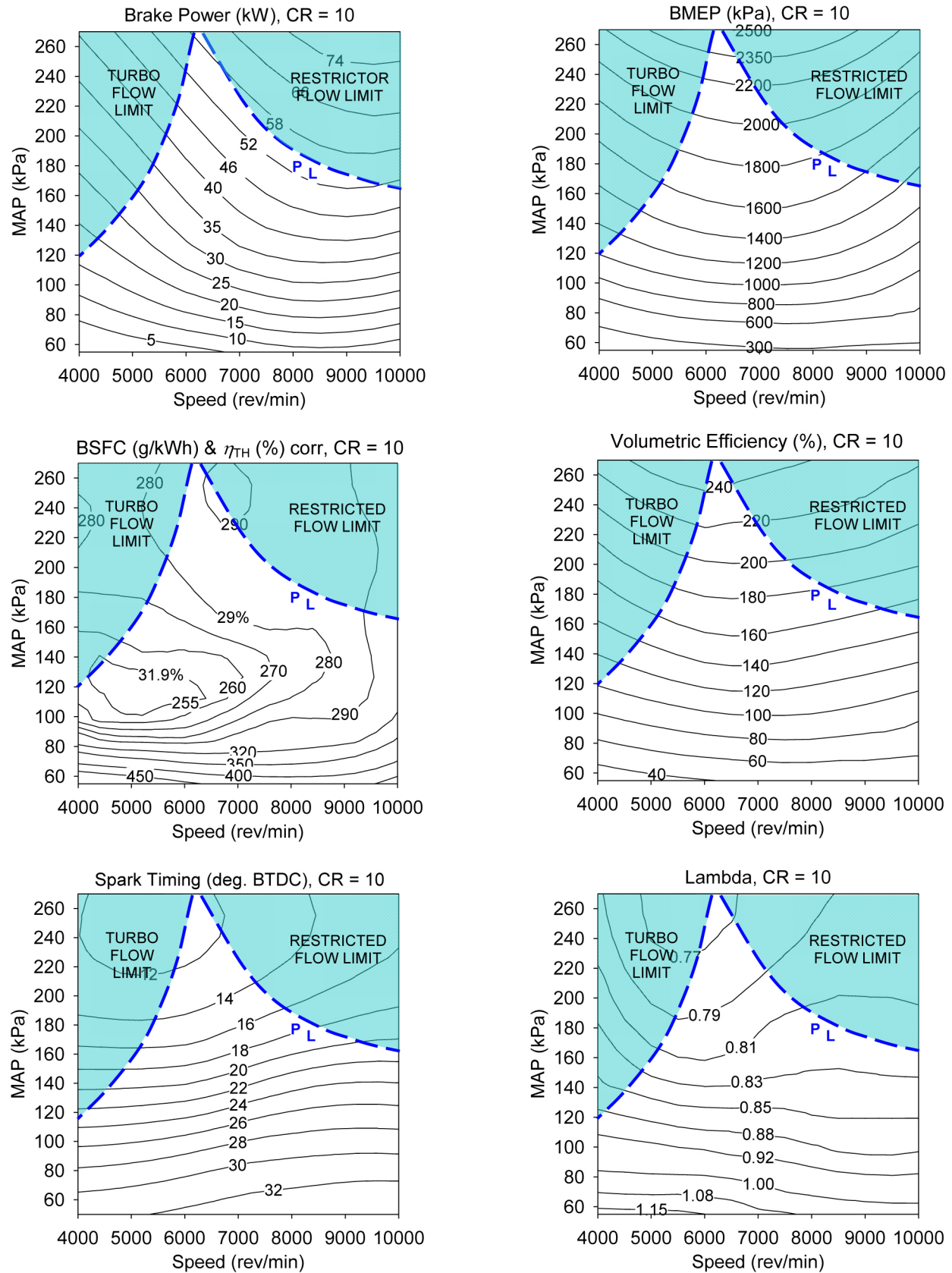


**Figure 2. Turbocharged PFI experimental results for brake power, BMEP, BSFC, brake thermal efficiency ( $\eta_{TH}$ ), volumetric efficiency, spark timing and lambda with varying engine speed and MAP parameters, CR=10. PL is the performance limit line defined at WOT. Airflow limit performance level (25 bar BMEP<sub>MAX</sub>).**

Figure 4 displays the knock and airflow limitations as functions of engine speed, MAP and CR. The cross plots have been constructed from multi-CR experimental data points. These CR values included 9.6, 10, 11, and 13. The resulting piston crown and combustion chamber shapes associated with each CR are displayed in Figure 3. It is noted that squish areas around the periphery of the chamber were maintained to minimize the variation in squish generated turbulence and vorticity, limiting any resulting combustion effects from varying CRs. However, changes to the turbulent length scale were inevitable due to the changes in the combustion chamber dish depth as the CR was altered.

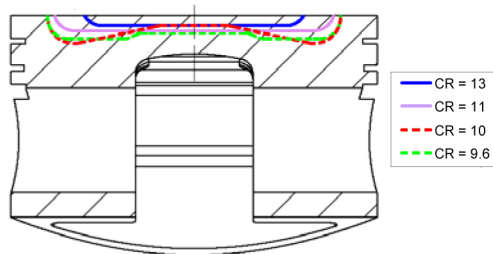


**Figure 3. Piston sectional view illustrating the combustion chamber geometry for varying CRs. Combustion chamber geometry is symmetrical about the piston central axis.**

The cross hatched areas in Figure 4 indicate domains where engine operation was knock limited but could be controlled via tuning strategies to avoid the damage limit. The shaded areas in plots indicate where engine operation was not possible due to airflow limitations or knock levels exceeding the damage limit. Airflow limitations were associated with insufficient compressor delivered airflow or a choked intake restriction. The performance limit (PL) is highlighted by the dashed blue line, corresponding to WOT for a given CR and MAP level.

With knock compensation tuning strategies (varying degrees of spark retard and/or fuel enrichment) together with CR reductions being the only forms of knock control used throughout experimentation, the cross plots displayed must first be put into context. It is noted that further knock reductions are expected with the implementation of various other strategies thus allowing increased MAP and/or CR before exceeding the damage limit. These strategies could include direct injection, cooled EGR, intake charge cooling (intercooling and/or evaporative), variable valve motion, combustion enhancement through chamber design and increases in fuel quality [1, 15,16,17,18,19]. However, these knock preventative strategies were not implemented during experiments due to their added complexity and as such were out of the scope of the project.

Table 3 summarizes CR results displayed in Figure 4, with the estimated highest useful compression ratio (HUCR) determined from the experimental data. Some safety margin was assumed in the estimation to ensure the damage limit was avoided.

**Table 3. Estimated highest useful compression ratio (HUCR) from turbocharged experimental test data operating limits displayed in Figure 4, with some safety margin.**

| Performance Level                                  | HUCR |
|----------------------------------------------------|------|
| 150 kPa MAP (16.5 bar $BMEP_{MAX}$ )               | 11.9 |
| 200 kPa MAP (20 bar $BMEP_{MAX}$ )                 | 10.9 |
| Airflow limit @ 270 kPa MAP (25 bar $BMEP_{MAX}$ ) | 9.8  |

Results from Figure 4 show that is advantageous to optimize the CR to a fixed value, resulting in incremental levels of rising MAP for engine speed increases to maximize power and efficiency, while continuously avoiding the damage limit. Furthermore, specific MAP levels are shown to define the HUCR. At 150 kPa MAP, the HUCR with some safety margin is shown to be 11.9 compared to 10.9 for the 200 kPa MAP case. These CR values are relatively high when compared to larger bore engines with similar boost levels [3, 20,21,22], mainly attributed to the reduced flame travel path to consume the last burning end-gas and the reduced end-gas volume around the periphery of the chamber.

Trends from Figure 4 confirm that for a given MAP and CR condition, the engine is less susceptible to knock at higher speeds [8, 10, 12, 13]. The reduced knock likelihood is a consequence of the increased flame speeds within the combustion chamber (Figure 9), which consume the unburned mass in the end-gas region more quickly. Increasing flame speed decreases the likelihood of knock due to the reduced end-gas residence time within the combustion chamber. It is also interesting to note that spark retard and/or fuel enrichment can be used as methods of knock control for up to 1-2 CR points depending on the knock severity. This can be a useful tool during boosted engine development, as it allows some safety margin to avoid the damage limit without having to reduce the CR. Moreover, this method of knock control allows the operating limit to be extended, thus enabling increased performance. However, the possible efficiency increases due to potential CR and/or MAP increases were found to be offset by the increased enrichment and less optimum spark timing.

The relationship between engine speed, MAP and CR parameters and their effect on the operating limits have been described for this small engine. Experiments highlighted that spark knock is the most dominant factor in limiting the performance of this small engine. However, reductions in end-gas knock severity at similar MAP and CR has enabled the performance operating limits to be extended for this particular small engine when compared to larger units. This is demonstrated with a peak BMEP value of 25 bar recorded at the airflow limit of 270 kPa MAP and a CR of 10. The increased performance of this small engine is attributed to the physical size reduction, particularly the reduced bore size (when

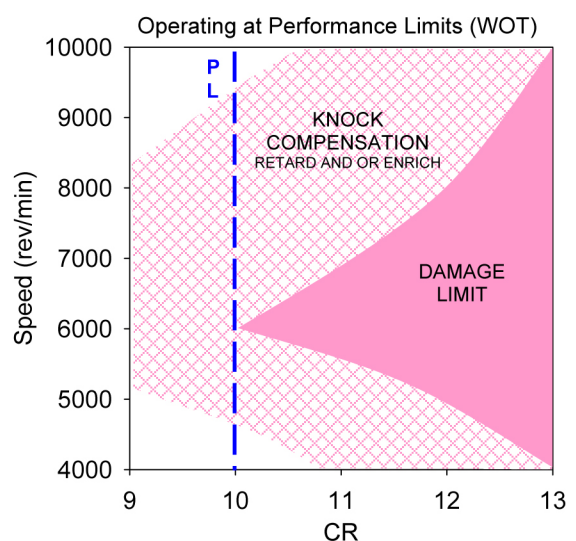
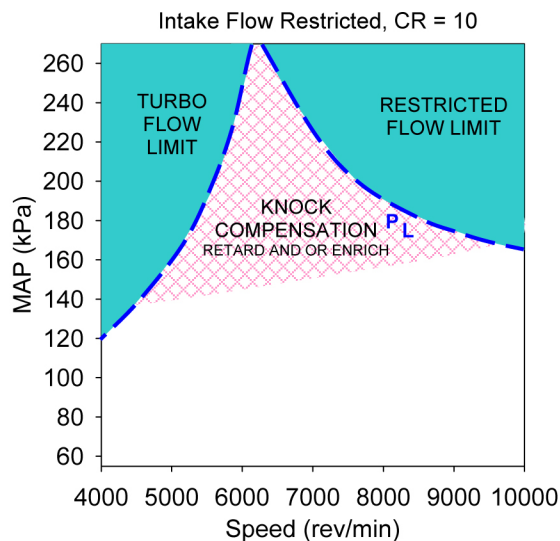
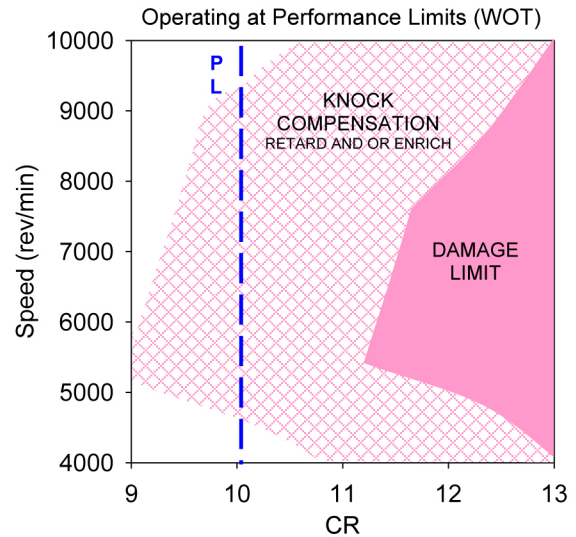
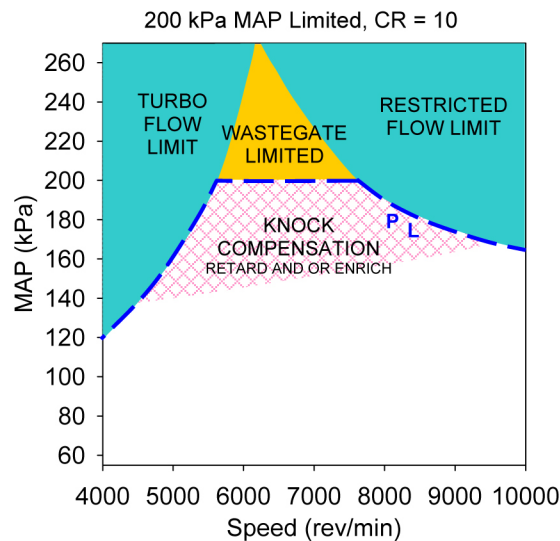
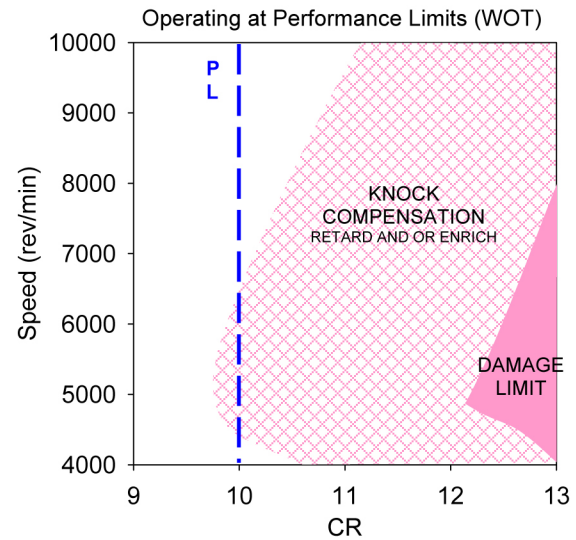
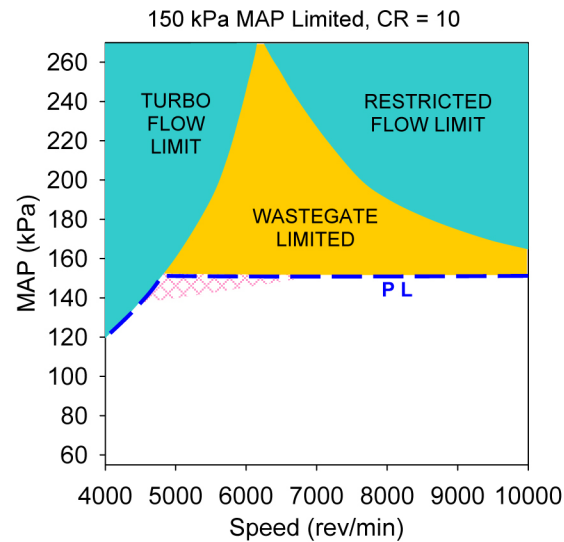


**Figure 4. Knock and airflow limitations versus engine speed, MAP and CR. Cross hatched areas indicate operation with spark retard and/or fuel enrichment to compensate for knock. Shaded areas indicate non-operation due to knock levels above the damage limit or limited airflow. PL is the performance limit line at WOT for a given CR and MAP level. (Left): Engine speed versus MAP domain at CR = 10. (Right): CR versus engine speed domain at the performance limit (WOT). (Upper): 150 kPa MAP limited, 16.5 bar BMEP<sub>MAX</sub>. (Middle): 200 kPa MAP limited, 20 bar BMEP<sub>MAX</sub>. (Lower): Airflow limit, 25 bar BMEP<sub>MAX</sub>.**