

Boosted HCCI for High Power without Engine Knock and with Ultra-Low NOx Emissions - using Conventional Gasoline

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ABSTRACT

The potential of boosted HCCI for achieving high loads has been investigated for intake pressures (P_{in}) from 100 kPa (naturally aspirated) to 325 kPa absolute. Experiments were conducted in a single-cylinder HCCI research engine (0.98 liters) equipped with a compression-ratio 14 piston at 1200 rpm. The intake charge was fully premixed well upstream of the intake, and the fuel was a research-grade (R+M)/2 = 87-octane gasoline with a composition typical of commercial gasolines.

Beginning with $P_{in} = 100$ kPa, the intake pressure was systematically increased in steps of 20 - 40 kPa, and for each P_{in} , the fueling was incrementally increased up to the knock/stability limit, beyond which slight changes in combustion conditions can lead to strong knocking or misfire. A combination of reduced intake temperature and cooled EGR was used to compensate for the pressure-induced enhancement of autoignition and to provide sufficient combustion-phasing retard to control knock. The maximum attainable load increased progressively with boost from a gross indicated mean effective pressure (IMEP_g) of about 5 bar for naturally aspirated conditions up to 16.34 bar for $P_{in} = 325$ kPa. For this high-load point, combustion and indicated thermal efficiencies were 99% and 47%, respectively, and NOx emissions were < 0.1 g/kg-fuel. Maximum pressure-rise rates were kept sufficiently low to prevent knock, and the COV of the IMEP_g was < 1.5%. Central to achieving these results was the ability to retard combustion phasing (CA50) as late as 19° after TDC with good stability under boosted conditions. Detailed examination of the heat release rates

shows that this substantial CA50 retard was possible because intake boosting significantly enhances the early autoignition reactions, keeping the charge temperature rising toward the hot-ignition point despite the high rate of expansion at these late crank angles. Overall, the investigation showed that well-controlled boosted HCCI has a strong potential for achieving power levels close to those of turbo-charged diesel engines.

INTRODUCTION

Homogeneous charge compression ignition (HCCI) engines can provide both high efficiency and low NOx and particulate emissions, but their limited power output remains a significant barrier to widespread implementation. The load limitation typically occurs because the maximum cylinder-pressure rise rate (PRR) increases with fueling rate, eventually causing engine knock. To mitigate these high PRRs with increased fueling, the combustion phasing can be retarded, but the amount of allowable retard is limited by excessive cycle-to-cycle variation in the power output, as measured by the gross indicated mean effective pressure (IMEP_g),¹ and eventually misfire [1,2]. As the fueling is increased, it can become increasingly difficult to maintain sufficiently retarded combustion to prevent run-away knock [3] without having poor combustion stability that can drift into misfire [1,2,4]. Although rapid feedback, closed-loop control systems can help extend this knock/stability limit [1], the maximum load for naturally aspirated HCCI is still low relative to traditional spark-ignition or turbo-charged diesel engines.

Additionally, depending on the fuel-type and operating conditions, the fueling can reach the point that the

¹IMEP_g refers to the IMEP over the compression and expansion strokes only, while IMEP_{net} refers to the IMEP over the entire cycle, including the gas-exchange strokes.

combustion is no longer sufficiently dilute to maintain low combustion temperatures, and NO_x emissions begin to rise [1,3]. For these conditions, the value of using HCCI could be diminished since NO_x aftertreatment would be required.

Intake-pressure boosting is widely used to increase the power output of internal combustion engines, and it offers a means for increasing power while still maintaining the dilute conditions required for low NO_x. However, the application of boost to HCCI has been limited because it can be difficult to control knocking. The knocking propensity of HCCI increases with boost for two reasons. First, the greater charge mass with boost results in a greater pressure rise with combustion, which increases the PRR for the same combustion phasing. Second, boost enhances the autoignition process, resulting in an advancement of the combustion phasing, which would increase the PRR even without the greater pressure rise with boost. Because of these difficulties, most previous attempts to achieve high loads with boosted HCCI used special fuels and/or reduced compression ratios.

Early work on boosted HCCI was presented by Christensen *et al.* [5] who showed that boost could significantly enhance the IMEP, but the combustion phasing became advanced causing knocking. However, they noted that ethanol and natural gas showed less advancement with boost than iso-octane. This behavior allowed them to reach an $IMEP_{net} = 14$ bar using natural gas compared to only about 9.7 bar for iso-octane, at an absolute intake pressure (P_{in}) = 3 bar. Later, Christensen and Johansson [6] extended the high load to $IMEP_g = 16$ bar using natural gas with an iso-octane pilot, combined with exhaust gas recirculation (EGR) and a variable compression ratio ($P_{in} = 2.5$ bar). In a later work from the same research group, Olsson *et al.* [7] achieved a brake mean effective pressure (BMEP) of 16 bar for $P_{in} = 3$ bar absolute. In this work, dual fueling with variable amounts of ethanol and n-heptane was used to control combustion phasing, with the fuel mixture being mostly ethanol for high loads. Ethanol was selected as the less-reactive fuel for this work because iso-octane and gasoline were found to ignite too early at high load. In all these works, the charge was well premixed using port injection, compression ratios ranged from 17 to 19, and NO_x emissions were well below the US 2010 standard of 0.27 g/kWh.

Using a somewhat different approach based on early direct injection (DI) with diesel fuel injectors, Bessonette *et al.* [8] examined the high load limits of HCCI-like combustion for an engine with a compression ratio of 12. Several different fuel blends were investigated, and the study found that a fuel between diesel and gasoline worked best. A BMEP of 16 bar was achieved with both a low-octane gasoline and a low-cetane diesel fuel. NO_x emissions were generally low, but exceeded the US 2010 standard at some conditions, probably due to incomplete charge mixing.

Although the above-mentioned studies did not use conventional liquid fuels, conventional gasoline was used by Kalghatgi *et al.* [9] in an investigation of partially premixed low-temperature combustion that has some similarities to HCCI. In this work, gasoline was directly injected into a diesel engine with about 75% of the fuel injected 11° before top dead center (bTDC) and the remainder injected early (150° bTDC). Due to the relatively long ignition delay with gasoline, substantial premixing occurred, resulting in NO_x and smoke levels that were far lower than for traditional diesel operation. A high load point of IMEP = 15.95 bar (gross or net IMEP is not specified) was achieved for $P_{in} = 2$ bar absolute, but with smoke levels of FSN = 0.067 and NO_x emissions of 0.58 g/kWh, the latter being more than twice the US 2010 limit. In summary, these previous works achieved power levels on the order of 16 bar IMEP, but issues such as non-conventional fuels, dual fuels, high PRRs, or high NO_x emissions limit applicability.

The objective of the current work is to determine the potential of boosted HCCI for producing high power levels using conventional gasoline without engine knock. Maintaining NO_x emissions below the US 2010 standard and high thermal efficiency were also priorities. For this study, the intake charge was fully premixed, which eliminated the potential of local regions with high combustion temperatures that might produce NO_x. A compression ratio of 14 was selected as being sufficiently high for good thermal efficiency, but low enough to mitigate friction losses. With this compression ratio, autoignition of gasoline for naturally aspirated conditions required heating the intake air to temperatures of 130°C or higher (in practical applications, retained hot residuals could be substituted for this intake heat). However, the required intake temperature dropped rapidly with boost due to the pressure-induced enhancement of autoignition. Controlling engine knock under highly boosted conditions required substantial combustion-phasing retard. Fortunately, under boosted conditions, good combustion stability could be maintained with much greater combustion-phasing retard than was possible for naturally aspirated conditions. This allowed high fueling rates to be maintained for boosted operation, as will be discussed in detail later. To achieve the required retard despite the pressure-induced enhancement of autoignition, a combination of reduced intake temperature and cooled EGR was used.

The next section describes the experimental facility, fuel specifications, data acquisition and analysis techniques, and the knock-limit criterion. Following this, the results are presented in two parts. First, engine performance data for various levels of intake boost are presented and discussed. The second part presents an analysis and discussion of the cylinder temperature and heat release rates (HRR) that explain why boosted operation worked well for the conditions

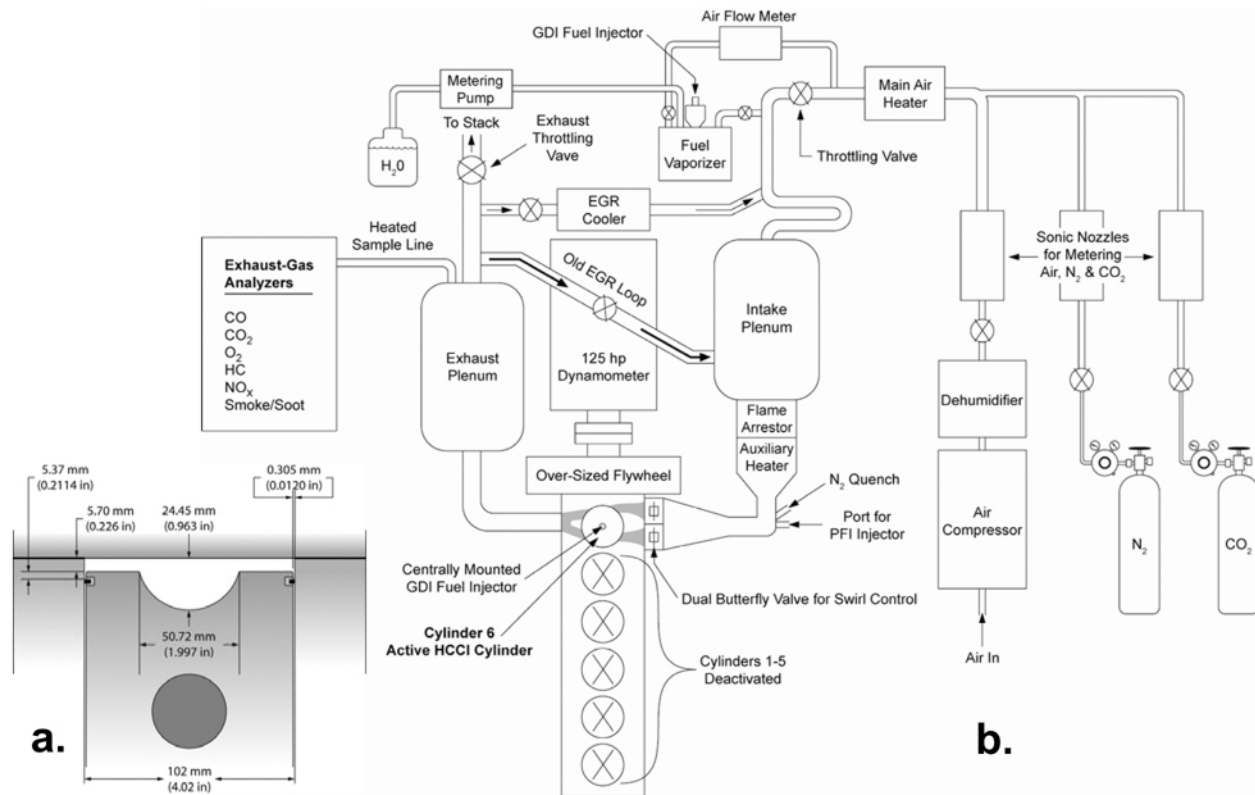


Figure 1. Schematic of the HCCI engine facility (b), and combustion chamber geometry of the CR = 14 piston, at TDC (a).

studied. Finally, in the last section, the study is summarized and conclusions are drawn.

EXPERIMENTAL SETUP & DATA ACQUISITION

ENGINE FACILITY

The HCCI research engine used for this study was derived from a Cummins B-series six-cylinder diesel engine, which is a typical medium-duty diesel engine with a displacement of 0.98 liters/cylinder. Figure 1 shows a schematic of the engine, which has been converted for single-cylinder operation by deactivating cylinders 1-5. The engine specifications and operating conditions are listed in Table 1. The configuration of the engine and facility is nearly the same as for previous studies (e.g. Refs. [2,3,4,10,11]).

<figure 1 here>

TABLE 1. Engine Specifications and Operating Conditions

Displacement (single-cylinder)	0.981 liters
Bore	102 mm
Stroke	120 mm
Connecting Rod Length	192 mm
Geometric Compression Ratio	14:1
No. of Valves	4
IVO	0° CA*
IVC	202° CA*
EVO	482° CA*
EVC	8° CA*
Swirl Ratio	0.9
Fueling system	Fully Premixed
Engine Speed	1200 rpm
Intake Temperature	45 – 146°C
Intake Pressure (abs.)	100 – 325 kPa
Coolant Temperature	100°C

*0° CA is taken to be TDC intake. The value-event timings correspond to 0.1 mm lift.

The active HCCI cylinder is fitted with a compression-ratio (CR) = 14 custom piston as shown in Fig. 1a. This piston provides an open combustion chamber with a large squish clearance and a quasi-hemispherical bowl. Compared to the

broad shallow bowl used in most previous studies, *e.g.* [4,12], this bowl is deeper and has a smaller diameter, thus allowing a broad flat squish region. This change in bowl geometry was made to facilitate safer operation for concurrent testing of a variable valve actuation system (not reported here). Comparison tests showed minimal differences in performance and emissions compared to the CR = 14 shallow-bowl piston (see Ref. [12] for a schematic). A more complete discussion of the comparison between pistons may be found in Ref. [3]. Both of these custom-designed CR = 14 pistons provide a small top-land ring crevice, amounting to only 2.1% of the top-dead-center (TDC) volume, including the volume behind the top piston ring.

As shown in Fig. 1b and discussed in Ref. [10], the engine facility is equipped with two EGR loops and the capability of supplying simulated EGR (combustion products) by adding N₂, CO₂ and H₂O. For the current study, all EGR was supplied using the cooled EGR loop. The EGR cooler (a gas-to-water heat exchanger) is necessary for maintaining the desired intake temperature as EGR levels are increased. In addition, this loop introduces the EGR well upstream of the intake plenum, so the EGR travels with the intake air through a series of bends before reaching the intake plenum, to insure that the intake charge is well mixed. With this configuration, the exhaust pressure must be greater than the intake pressure for EGR to flow into the intake. The required back pressure was achieved by throttling the exhaust flow using the valve shown in Fig. 1b.

The intake air was supplied by an air compressor and precisely metered by a sonic nozzle as shown in Fig. 1b. For operation without EGR, the air flow was adjusted to achieve the desired intake pressure, as measured by a pressure transducer on the intake runner. For the current study, the intake pressure was varied from 100 kPa (simulating naturally aspirated conditions) to 325 kPa. All pressures given are absolute. For operation with EGR, the air flow was reduced from the amount required to achieve the desired intake pressure with air alone, and the valve on the EGR line was opened. The exhaust back-pressure throttle valve was then adjusted to produce enough EGR flow to reach the desired intake pressure. This typically resulted in the exhaust pressure being about 2 kPa greater than the intake pressure. For consistency, the back pressure was maintained at 2 kPa above the intake pressure, even when EGR was not used. The EGR fraction was varied by adjusting the amount of supplied air, and then adjusting the exhaust throttle to maintain the desired intake pressure.

For this study, the intake charge was fully premixed using the fueling system shown at the top of the schematic in Fig. 1b. This fueling system consists of a gasoline-type direct injector mounted in an electrically heated fuel vaporizing chamber

and appropriate plumbing to ensure thorough premixing with the air and EGR upstream of the intake plenum. A positive displacement fuel flow meter was used to determine the amount of fuel supplied. The fuel was a research-grade gasoline supplied by Chevron-Phillips Chemical Co., and its specifications are listed in Table 2.

TABLE 2. Chevron-Phillips Research-Grade Gasoline*

Antiknock Index (R+M)/2.....	87.0
RON	90.8
MON	83.2
Specific gravity	0.746
API gravity	58.2
Carbon, wt%	85.59
Hydrogen, wt%	13.88
Oxygen, wt%	0.00
A/F Stoichiometric	14.79
Lower Heating Value, gas-phase fuel (MJ/kg)	43.17
LHV for stoichiometric charge (MJ/kg)	2.734
Hydrocarbon Type, vol%	
Aromatics.....	22.7
Olefins	4.2
Saturates	73.1
Distillation, °C	
5%	53
10%	59
30%	75
50%	93
70%	115
90%	145
95%	157

**Based on analysis provided by Chevron-Phillips*

Prior to starting the experiments, the engine was fully preheated to 100°C by means of electrical heaters on the “cooling” water and lubricating-oil circulation systems. In addition, the intake tank and plumbing were preheated to 50 - 60°C to avoid condensation of the fuel or water from the EGR gases. An auxiliary heater mounted close to the engine provided precise control of the intake temperature to maintain the desired combustion phasing. Intake temperatures ranged from 45° - 146°C. All data were taken at an engine speed of 1200 rpm.

DATA ACQUISITION

Cylinder pressure measurements were made with a transducer (AVL QC33C) mounted in the cylinder head approximately 42 mm off center. The pressure transducer signals were digitized and recorded at 1/4° CA increments for one hundred consecutive cycles. The cylinder-pressure transducer was pegged to the intake pressure near bottom dead center (BDC)

*Based on analysis provided by Chevron-Phillips

where the cylinder pressure reading was virtually constant for several degrees. Intake temperatures were monitored using thermocouples mounted in the two intake runners close to the cylinder head. Firedeck temperatures were monitored with a thermocouple embedded in the cylinder head so that its junction was about 44 mm off the cylinder center and 2.5 mm beneath the surface. Surface temperatures were estimated by extrapolating the thermocouple reading to the surface, using the thickness of the firedeck and assuming that its back surface was at the 100°C cooling-water temperature [13]. For all data presented, 0° crank angle (CA) is defined as TDC intake (so TDC compression is at 360°). This eliminates the need to use negative crank angles or combined bTDC, aTDC notation.

The crank angle of the 50% burn point (CA50) was used to monitor the combustion phasing. CA50 was determined from the cumulative apparent heat-release rate (AHRR), computed from the cylinder-pressure data (after applying a 2.5 kHz low-pass filter [14]). Computations were performed for each individual cycle, disregarding heat transfer and assuming a constant ratio of specific heats [15]. The average of 100 consecutive individual-cycle CA50 values was then used to monitor CA50 and for the CA50 values reported. The reported PRRs and ringing intensities are computed from the same low-pass-filtered pressure data. For each cycle, the maximum PRR was analyzed separately with a linear fit over a moving $\pm 0.5^\circ$ CA window. Similar to CA50, these individual-cycle values were then averaged over the 100-cycle data set.

A second method of computing the HRR was used for detailed HRR-curve analysis presented in the second part of the results. Here, the heat release was computed in a more refined way from the ensemble-averaged pressure trace (with the 2.5 kHz low-pass filter applied), using the Woschni correlation for heat transfer [15]. Using the ensemble-averaged pressure trace has benefits from the standpoint of reduced noise on the heat-release traces. On the other hand, it can lead to overestimated burn durations if the cycle-to-cycle variations are large. However, for the condition where this was applied, the phasing was fairly stable with the standard deviation of CA50 over 100 fired cycles averaging about 1.2° CA. Moreover, the results of this detailed HRR analysis are mainly used for comparisons of the early part of the heat release, leading up to hot ignition. Since these early autoignition reactions persist for 15° - 25° CA, the relatively small cycle-to-cycle variation in CA50 will have little effect.

Although these detailed HRR curves appear quite smooth when plotted on a scale showing the entire combustion event, small quasi-periodic fluctuations were evident when viewed on the highly amplified scale used for comparisons of the early heat release. This is believed to be an artifact of the

HRR being based on the derivative of the pressure trace (which amplifies noise in the pressure data) and the low-pass filtering process [14]. To mitigate these fluctuations, the HRR data (in Figs. 12b, 13, 14, and 15) were filtered using a piecewise smoothing function in which the value of each point was adjusted based on a weighted average of its original value and the values of its neighbors. For this filtering, the original value is given the greatest weight, and the weighting of the values of the neighboring points is reduced linearly with distance over a specified number of neighbors. The number of neighbors specified determines the amount of smoothing. Care was taken to compare the filtered and unfiltered HRR curves and to adjust the number of neighbors to insure that the filtered curves faithfully reproduced the trends in the unfiltered HRR data. Finally, it should be noted that although the filtering facilitates comparisons between the curves, all observations discussed regarding the comparisons of the HRR between conditions are evident without the filtering, *i.e.* the differences discussed are greater than the noise removed with this filtering.

Charge temperatures during the closed part of the cycle (*i.e.* compression/expansion) are computed using the ideal-gas law in combination with the measured pressure (ensemble-averaged over 100 cycles), the known cylinder volume, and the trapped mass. The trapped mass equals the sum of the supplied air, fuel, EGR, and residuals, the latter being computed by applying the ideal gas law to the clearance volume at TDC-valve-overlap, using a pressure determined from cycle-simulation modeling (using Ricardo's WAVE program) and an estimated residual temperature based on exhaust blowdown to this pressure. The average molecular weight used for the calculation of the charge temperature during the compression stroke corresponds to that of the trapped gases - including fresh air, EGR, retained residuals, and fuel. During the combustion event, and in proportion to the mass fraction burned, the molecular weight in the calculations is gradually changed to that of the measured exhaust composition.² This exhaust molecular weight is then used for the remainder of the expansion stroke and for computing the trapped mass.

Exhaust emissions data were also acquired, with the sample being drawn from the exhaust plenum using a heated sample line (see Fig. 1b). CO, CO₂, HC, NO_x, and O₂ levels were measured using standard exhaust-gas analysis equipment. In addition, a second CO₂ meter monitored the intake gases just prior to induction into the engine. This allowed the EGR fraction of the intake gases to be computed from the ratio of the intake and exhaust CO₂ concentrations.

²Dissociation of the combustion products is neglected since it is expected to be small due to the relatively low peak-combustion temperatures.

RESULTS AND DISCUSSION - ENGINE PERFORMANCE

The goal of the engine performance testing was to determine the maximum load that can be achieved with boosted HCCI while maintaining acceptable knock and NO_x emissions. To accomplish this, the intake pressure was systematically increased, and for each P_{in} , fueling was increased up to the knock/stability limit [1,2,3]. Because increased pressure enhances autoignition, a greater degree of combustion phasing “adjustment” was required as the boost level was increased. A combination of intake-temperature reduction and cooled-EGR addition was used to achieve the necessary combustion-phasing retard.

KNOCK/STABILITY LIMIT

As mentioned in the Introduction, for each P_{in} , increasing the fueling increases the PRR, eventually causing the engine to knock. On the other hand, retarding the combustion phasing reduces the PRR [2] since it slows the HRR by amplifying the benefit of the naturally occurring thermal stratification [16]. Additionally, the larger combustion-chamber volume for retarded combustion reduces the total pressure rise, further reducing the PRR for the same HRR [3]. This reduced PRR allows the fueling to be increased without knock, and the process of incrementally increased combustion retard followed by increased fueling can be continued up to the stability limit, as demonstrated in Ref. [2].

The allowable combustion retard is limited by poor combustion stability as manifested by excessive cycle-to-cycle variation in the IMEP_g. This occurs because the expansion rate increases with increased timing retard, producing a higher rate of charge cooling. This cooling counteracts the temperature rise produced by the early autoignition reactions, decreasing the rate of temperature rise toward the point of hot ignition [4]. As retard is increased, the temperature-rise rate decreases, eventually reaching the point that small random variations in the compressed-gas temperature cause significant variations in hot-ignition timing. This causes large variations in the IMEP_g and can lead to run-away knock or decay to misfire [2,3,4].³

As a result, the maximum fueling rate is limited by the amount of combustion-phasing retard that can be applied with acceptable combustion stability. In practice, it is difficult to precisely determine the maximum load point since operating parameters such as the intake temperature must be adjusted precisely and in small increments to prevent run-away knock or misfire when operating near the knock/stability limit. For

the current work, the fuel was incrementally increased in small steps, amounting to a 2 - 3% increase in IMEP_g, as the maximum fueling rate was approached, and the step size was often reduced if operation became unstable. Therefore, the highest loads reported for the various intake pressures are expected to be very close to the true knock/stability limit, attainable without fast closed-loop feedback controls [1].

Knock-Limit Criterion

The acceptable knock limit for HCCI engines is often defined in terms of a maximum allowable PRR ($dP/d\theta$, where θ is a variable representing °CA). However, this does not correctly reflect the potential for knock under boosted conditions where the cylinder pressure changes significantly. In this work, the correlation for ringing intensity developed by Eng [17] is used as a measure of engine knock.

$$\text{Ringing Intensity} \approx \frac{1}{2\gamma} \cdot \frac{\left(0.05 \cdot \left(\frac{dP}{dt}\right)_{\max}\right)^2}{P_{\max}} \cdot \sqrt{\gamma R T_{\max}} \quad (1)$$

Where $(dP/dt)_{\max}$, $P_{\max}$, and $T_{\max}$ are the maximum values of PRR (in real time), pressure, and temperature, respectively, γ is the ratio of specific heats (c_p/c_v), and R is the gas constant. The ringing is a measure of the acoustic energy of the resonating pressure wave that creates the sharp sound commonly known as engine knock. Based on the onset of an audible knocking sound and the appearance of obvious ripples on the pressure trace, a ringing criterion of 5 MW/m² was selected as the ringing limit. This corresponds to about 8 bar/°CA at 1200 rpm, naturally aspirated. It should be noted that as P_{in} is increased above 100 kPa, $dP/d\theta$ can exceed 8 bar/°CA since the corresponding increased value of $P_{\max}$ in the denominator of Eq. 1 reduces the ringing intensity for a given PRR. At all boost levels tested, perceived engine knock correlated well with the ringing intensity rising above 5 MW/m², giving confidence in this correlation.

ACHIEVING MAXIMUM LOAD WITH BOOST

No EGR Addition

As a starting point for determining the maximum load with intake boost, P_{in} was set to 100 kPa, simulating naturally aspirated conditions. For this P_{in} , intake heating was required to obtain autoignition, even though no EGR was used. The fueling was initially set somewhat arbitrarily to a moderately

³For example, if one cycle ignites a couple of degrees too early, it will knock which increases wall heat transfer. Then, the hotter walls cause the following cycle to be even more advanced, resulting in even harder knock and greater wall heat transfer, and so on. Once started, this run-away knock process is very rapid [3,10], and can lead to engine damage. Similarly, a cycle that is a bit too retarded will not burn as completely, leading to lower residual and wall temperatures, causing the next cycle to be more retarded and burn even less completely, and so on until full misfire.