

A Study Isolating the Effect of Bore-to-Stroke Ratio on Gasoline Engine Combustion Chamber Development

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ABSTRACT

A unique single cylinder engine was used to assess engine performance and combustion characteristics at three different strokes, with all other variables held constant. The engine utilized a production four-valve, pentroof cylinder head with an 86mm bore. The stock piston was used, and a variable deck height design allowed three crankshafts with strokes of 86, 98, and 115mm to be tested. The compression ratio was also held constant. The engine was run with a controlled boost-to-backpressure ratio to simulate turbocharged operation, and the valve events were optimized for each operating condition using intake and exhaust cam phasers. EGR rates were swept from zero to twenty percent under low and high speed conditions, at MBT and maximum retard ignition timings. The increased stroke engines demonstrated efficiency gains under all operating conditions, as well as measurably reduced 10-to-90 percent burn durations. The results were quite non-linear, with the majority of the gains achieved in going from the 1:1 to 0.87:1 bore-to-stroke ratio cases. The further change to 0.75:1 showed significantly diminished returns. Flame speed and chamber geometry estimates were used to project further advantages at reduced bore and constant displacement.

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INTRODUCTION

Among the first questions to be addressed in laying out a new engine are those of total displacement and the number of cylinders. Immediately following from the resulting displacement per cylinder is bore-to-stroke ratio optimization. While these two dimensions can be simultaneously adjusted over a wide range at constant displacement, there is an optimum bore-to-stroke ratio for any particular application. The optimum is determined by four considerations [1]:

- Larger bore-to-stroke ratios allow increased valve sizes and improved breathing, especially at higher engine speeds.
- Larger bore-to-stroke ratios result in lower piston speeds at any given engine speed, thus allowing higher engine speeds.
- Smaller bore-to-stroke ratios reduce the surface area-to-volume ratio, especially when the piston is near TDC, thus reducing combustion heat transfer.
- Smaller bore-to-stroke ratios reduce flame travel distances, improving the combustion chamber geometry, and reducing heat release duration.

The first two considerations are especially important in high rpm, high specific power output applications. It follows directly that high-performance and racing engines have bore-to-stroke ratios greater than one (termed "oversquare"). Typical bore-to-stroke ratios

are between 1.1 and 1.2 to one in high-performance production engines, and may be greater than two to one in high-end racing applications such as Indycar and Formula One [2].

The second two considerations gain in importance when fuel efficiency is paramount. "Square" to slightly "undersquare" engines, with bore-to-stroke ratios between 0.9 and 1.0 to one have been typical in lower speed, fuel efficient engine applications.

With the objective of improved thermal efficiency and reduced fuel consumption, trends in new automobile engine development have included reduced displacement, reduced engine speed, and either turbocharging or supercharging [3,4]. These trends have led engine developers to revisit the question of bore-to-stroke ratio optimization. Reduced engine speed allows longer stroke engines while maintaining acceptable piston speeds. The combination of reduced engine speed and boosted intake manifold pressure allows acceptable engine breathing. The combustion rationale for increasing the stroke and reducing the bore-to-stroke ratio is further summarized as follows.

As vehicular engines are developed for increased efficiency several important design approaches can be identified. Reducing engine speed, reducing displacement, and boosting through turbocharging or supercharging are on the list of favored approaches. Another important lever for maximum efficiency is reduced combustion temperatures, through dilution with exhaust gas or lean operation. Reducing the combustion temperatures reduces

heat rejection during combustion, and improves the direct work extraction during the expansion process by increasing the ratio of constant pressure to constant volume specific heats. Unfortunately, as the flame temperature is reduced the flame speed drops, working against the efficiency improvement, and increasing the propensity to auto-ignition. Methods for increasing flame speed under lean or dilute conditions are important research topics [5].

Flame speed is determined by the combination of mixture strength, flame temperature, and local turbulence. Once the fuel and the desired mixture strength (fresh air-to-fuel ratio and amount of EGR) are determined, the remaining lever is that of enhancing turbulence at the flame front. In the engine operating environment, the turbulence regime of greatest importance is that of the “stretched” or “wrinkled” laminar flame [6,7]. In this regime it can be demonstrated that the turbulence level directly scales with engine speed, or more specifically with piston speed. One way in which turbulence can be enhanced is by using the combination of intake port design and piston motion to generate large-scale tumble flow during the intake process, and recover the energy from the decaying tumble motion during compression as fine scale turbulence at the flame front [8]. It is important as well to keep in mind that the end objective is increased turbulence intensity. Tumble is an important path, but if large scale flow remains during combustion it can be detrimental to flame speed at auto-ignition [9]. In the end, the design optimization challenge is that of maximizing turbulence intensity while minimizing intake flow restriction. Increasing the stroke increases the piston speed at any given engine speed, and therefore increases the turbulence intensity. In other words it is expected to perform favorably in the trade-off between increasing turbulence while minimizing intake port restriction. It holds the further attraction of reducing combustion chamber surface area-to-volume ratio as identified earlier. Several automobile engines have recently been released with bore to stroke ratios between 0.83 and 0.86 to one [10,11,12].

OBJECTIVES

The Southwest Research Institute has recently developed a new single-cylinder research engine that uses production piston, cylinder head, and valvetrain components, and allows the deck height and compression ratio to be easily adjusted. The authors recognized the opportunity this engine would provide to adjust the bore-to-stroke ratio while holding all other variables constant, allowing direct measurement of the effects of this variable alone on fuel efficiency and combustion performance. This was accomplished by changing the stroke and readjusting the deck height at each stroke to maintain a constant compression ratio.

A further question faced by the authors in defining the test constraints was that of whether to conduct the tests at constant displacement or constant bore diameter. While constant displacement was recognized as more representative of the design trade-offs faced in new engine layout, the resulting requirement to modify the valve diameters and port geometries introduces additional variables, the effects of which would be difficult to separate from that of the stroke variation. As a

result it was deemed that maintaining a constant bore diameter would be less problematic. Data analysis could be used to estimate the results of applying the findings at constant displacement.

ENGINE AND INSTRUMENTATION

Southwest Research developed a new single-cylinder research engine designed to use selected, production combustion chambers. A robust but flexibly designed crankcase can accept the cylinder, piston, cylinder head and valvetrain from a selected production engine. The design allows cylinder offset, deck height, and compression ratio to be easily adjusted based on the needs of a particular experiment. A boost system using an electrically-driven supercharger supplies the intake air, and allows cooled exhaust gas recirculation to be utilized. The supercharging system injects the EGR upstream of the compressor, and can use either engine exhaust or a modified gas stream as required (hydrogen enrichment, for example). Cooled, stoichiometric engine exhaust was used in the reported study. A variable exhaust restrictor was used to provide a specified boost to backpressure ratio, to simulate either supercharged or turbocharged operation. The engine is shown in [Figure 1](#) as installed in the test cell.

For this experiment the cylinder head and piston from a production GM two liter engine, with 86mm bore and 86mm stroke was selected as the baseline. The engine uses four valves per cylinder, with a pentroof combustion chamber and direct fuel injection. Good comparisons between port-injected and direct-injected engines have been recently reported by others [13,14].

Cam phasers were used to control both the intake and exhaust valve event timings. An extended length cylinder and connecting rod were constructed to allow the stroke to be incrementally increased. Billet crankshafts were machined as depicted in [Figure 2](#) to provide strokes of 86, 98, and 115mm. With a constant 86mm bore the resulting bore-to-stroke ratios were 1:1, 0.88:1, and 0.75:1 respectively. Each engine build used the same piston and top-end assembly, and the deck height was adjusted to maintain a constant compression ratio of 9.5:1.

Instrumentation included the full array of performance and emission measurement as well as high speed data acquisition. Three hundred cycle averages were used in determining mean effective pressures, peak cylinder pressure, heat release versus crank angle, and knock index. A constant measure of incipient knock was used to identify the limiting conditions at each test point. This measurement was made using computer analysis of the cycle-by-cycle pressure traces, and based on experience, selecting a threshold indicative of end-gas auto-ignition.

TEST MATRIX

The test matrix emphasized high-load, low-speed operating conditions, where the onset of auto-ignition limits performance capabilities. This combination of downsampling, downsizing and boosting results in increased specific output at relatively low engine speeds, with maximum BMEP determined by the conflicting limits of auto-ignition and misfire.



Figure 1. Single cylinder engine (right) and boost/EGR control system (left) as installed in the test cell.



Figure 2. One of the machined billet crankshafts used in the engine. Three crankshafts were machined to provide strokes of 86, 98, and 115mm.

The tests conducted are summarized as follows:

86mm Stroke

- 2000 rpm, spark timing for optimum efficiency (Cases referred to as MBT timing). EGR sweep from zero to onset of misfire.
- 2000 rpm, with spark timing at maximum retard (selected as 25 degrees ATDC 50 percent burn). EGR sweep from zero to onset of misfire.
- 4000 rpm, spark timing for optimum efficiency. EGR sweep from zero to onset of misfire.
- 4000 rpm, spark timing retarded until limiting exhaust temperature was reached. IMEP held constant as EGR was swept from zero to onset of misfire.

115mm Stroke

The entire test matrix conducted with the 86mm stroke was repeated.

98mm Stroke

The 2000 rpm cases conducted with the 86mm and 115mm strokes were repeated. Due to resource limits the 4000 rpm cases were not run with the 98mm stroke.

A stoichiometric air-fuel ratio was maintained, and 87 octane gasoline was used. Under each operating condition the cam phasers were adjusted for minimum specific fuel consumption. At zero EGR the boost pressure was then increased until incipient knock was measured. The backpressure was adjusted to maintain exhaust manifold pressure 10 kPa below intake manifold pressure, representative of a well matched turbocharger. The conditions were repeated with EGR increased in five percent increments until stable combustion could no longer be achieved, as evidenced by increased fuel consumption and hydrocarbon emissions.

At each speed the sequence just described was run first at the optimum, or maximum brake torque (MBT) timing. At 2000 rpm the test sequence was repeated at maximum timing retard; this was defined as the spark timing at which the 50 percent burn point occurred at 25 degrees after TDC. At 4000 rpm the same MBT timing sequence was used. Full load was determined based on a specified maximum torque selected below that achieved at 2000 rpm. The maximum timing retard was based on exhaust temperature limits.

A 2.5 bar, 2000 rpm test point was repeated each day to ensure consistency. The complete test matrix was repeated at the stroke dimensions of 86 and 115 mm, with the clearance height adjusted to maintain a constant compression ratio as discussed previously. Only the 2000 rpm cases were run with the 98 mm stroke; funding limitations precluded running the 4000 rpm test points.

Specific fuel consumption was tracked on a gross indicated (gISFC), net indicated (nISFC), and brake (BSFC) basis. Unless otherwise identified, net indicated performance is reported for this discussion. This is typically the most appropriate basis to use in single cylinder engine tests due to the difficulty of correlated friction and parasitic power requirements to those of a production, multi-cylinder engine installation.

RESULTS AND ANALYSIS

Test Results at 2000 rpm

The 2000 rpm performance data is summarized in [Figures 3](#) and [4](#). In [Figure 3](#) net indicated specific fuel consumption (nISFC) is plotted versus net indicated mean effective pressure (nIMEP). Both the MBT timing data and the data at maximum timing retard are plotted on the same coordinates, with the two conditions separated as indicated by the diagonal line. The individual data points at each stroke were obtained as the EGR was incrementally increased. The same data points are presented in [Figure 4](#), where nISFC is plotted as a function of EGR rate.

Both figures show the ability to increase the load and run the engine more efficiently as the EGR rate is increased. The rightmost points with each stroke in [Figure 3](#) are those at the maximum achievable EGR rate, and thus maximum work or nIMEP. Those in the lower left plots are with the timing optimized for best efficiency (MBT timing). Those in the upper right plots show the increased fuel consumption and increased torque output achieved at maximum ignition timing retard.