SUMMARY

This white paper presents a mathematical analysis of the piston dynamics of three popular versions of the Mitsubishi 4G63 engine, the stock 2.0L, the 2.1L destroked version with an 88mm crankshaft in a 4G64 block, and the stroker with a 100 mm crankshaft in a 4G63 block. Where applicable, charts are included to show the differences between the versions at different RPM's or different crank angles.

The conventional wisdom that strokers make more torque but the 2.0L will rev higher is explained with hard calculations of such factors as piston side loading friction and tension on the rods. The 2.3L stroker has the same side loading friction at 7150 RPM as the 2.0L at 8000 RPM.

The conventional wisdom that the vibration from removing balance shafts is less than the effect of mismatched piston weights is challenged and explained with a mathematical analysis. A 2.3L stroker has the same harmonic imbalance at 7040 RPM as the 2.0L at 8000 RPM.

The effect of the stroker geometry on camshaft selection is examined and supported with charts of common aftermarket 4G63 cams. The analysis shows that stroker engines are more tolerant of aggressive cams than the stock engine. The intake velocity of a stroker at 7000 RPM is the same as a 2.0L at 8000 RPM. Peak piston velocity of a stroker at 7390 RPM is the same as a 2.0L at 8000 RPM.

INTRODUCTION

After spending too many hours researching how to make my 1998 AWD Talon even better than new including reading many tuner posts in the DSM forums I decided to create this document as a payback to the DSM community.

The scope of this document is limited to the differences between common versions of the Mitsubishi 4G63 engine. The calculations are directed toward answering the question "should I stroke my 4G63 or not".

All equations used in this document are documented in Appendix A. The charts are print outs from an Excel project. The Excel file is available as described in Appendix A.

This document was drafted by Maurice Garoutte for the use of the DSM community and is not copyrighted. This document ended up longer and more technical than first planned but for readers who want more please follow the references.

Disclaimer

THIS DOCUMENT IS FOR THE ENTERTAINMENT AND POSSIBLE EDIFICATION OF THE DSM COMMUNITY ONLY. NO 4G63 ENGINES WERE HARMED FOR THE PRODUCTION OF THIS DOCUMENT. DON'T TRY THIS AT HOME. THE USE OR ABUSE OF THE INFORMATION HAS NO WARRANTY EITHER EXPRESS OR IMPLIED. THE AUTHORS LIBILITY FOR THE CONTENT OF THIS DOCUMENT SHALL NOT EXCEED THE PAYMENT FOR THE DOCUMENT, NOTA, ZERO BUCKS. IF YOUR 4G63 THROWS CHUNKS AT 10,000 RPM AFTER READING THIS, DON'T COME CRYING TO ME. YOUR MILAGE MAY VARY. THE OPINIONS EXPRESSED HERE MAY NOT REPRESENT THE OPINION OF THE MANAGEMENT. MANAGEMENT?

TABLE OF CONTENTS

LIST OF FIGURES

POWER vs. TORQUE

A quick search finds 1300 pages on dsmtuners.com site with the words stroker OR 2.3L AND torque.   That topic is a thoroughly beaten, very dead horse.   Without reading all of those pages, it's clear that the conventional wisdom is that the stroker makes more low end/mid range torque but not necessarily more power. That old saw is pretty much true and this paper is focused on the differences in the two versions, not on which is "better".

The next section of this paper is concerned with the RPM limitations of the stroker.   Before getting into the limitations of the stroker version of the 4G63, the horsepower limitations of the 4G63 are examined here. This is pretty much basic engine theory so feel free to skip to the next section.

Intake limitations

The 4G63 is a strong engine capable of handling much more horsepower than the factory spec.   Regardless of how strong the parts, engines making power just from air and gasoline are limited by how much air can get into the combustion chamber. Boost is good. A charge with two bar of boost can burn twice as much fuel as one bar.

For any boost level the limit of how much air can enter the combustion chamber is limited by the intake tract and the size of the intake valves. Back when the author put a 1951 hemi engine in a 1948 Plymouth coupe the hemispherical combustion chamber was state-of-the-art in car engines. See figure 1 at right. Back in 1951 those were some really big valves, angling the valves in from each side allowed bigger valves while a center mounted plug improved flame propagation.

Figure 1 Hemispherical Combustion Chamber

Bigger valves are a good thing because we can't just make the air go faster. When air flow reaches mach one (equal to the local speed of sound) the flow is "choked" increasing the pressure drop across the valve just puts more energy in shock waves but the air flow does not increase. According to the Mechanical Engineers Handbook (19) engines have the best volumetric efficiency with a mach index of 0.45 to 0.5. While a mach index of 1 is a hard wall, the choking effect from high mach flow starts about 0.6. Above a mach index of 0.6 the volumetric efficiency of any engines falls off rapidly.

The state-of-the-art in car engines is now four valves per cylinder with overhead cams as in the 4G63 pent roof engine. See figure 2 at right and compare to the Hemi combustion chamber above. Notice how the two large valves in the hemispherical chamber leave much of the combustion chamber as "not valves". The pent roof design modifies the hemi just enough to fit two more valves in the combustion leaving little area that is not filled with valves.
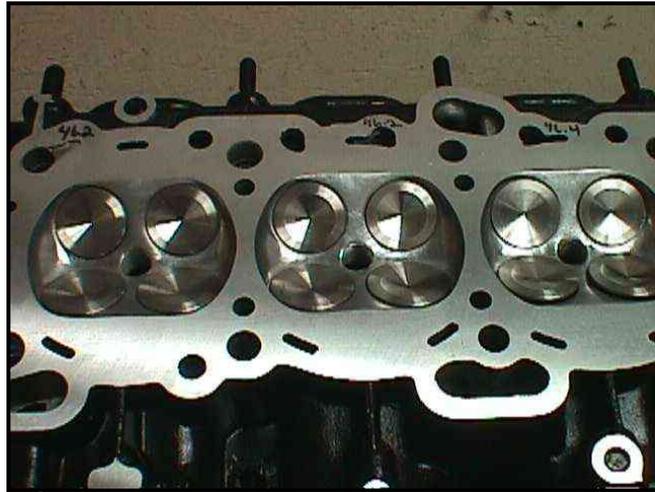
A lot has changed. That 1951 hemi weighed in at 960 pounds and made 180 hp at 4000 RPM from its 331 cubic inches. With a few bolt on modifications the 122 cubic inch 4G63 easily makes 400 hp at 8000 RPM while weighing in at only 300 pounds.

But the speed of sound is still the same as in1951. The hemi has an intake valve mach index of 0.51 at the 4000 RPM where max hp is developed. The 4G63 has the same .51 intake valve mach index at 8000 RPM. When the 4G63 reaches the mach index of .51 it's turning twice as fast as the hemi and with 15 pounds of boost is breathing more than twice as many pounds of air than the 331 hemi at 4000 RPM. Boost is good.

Aside from the history lesson, the message I get from this is that no matter how strong the main caps, or how well designed the pistons, there is a hard performance wall where the intake velocity approaches sonic flow through the valves. The 2.0L 4g63 with stock valves reaches 0.6 mach index at 9600 RPM. The 2.3L stroker reaches 0.6 mach index at 8400 RPM. Trying to spin the 2.3L stroker much past 8400 RPM will just hit the wall quicker as volumetric efficiency drops off. The theory seems to hold up to a check of the top eight DSM's on pump gas (Ludachris and anyone making more power) on http://www.dsmtuners.com/forums/dynosheets.php?&do=pumpgas. The table below shows that three of the eight are 2.0L engines, only Armandovivoni used nitrous.

Table 1 Horsepower Leaders on Pump Gas

| Rank	User	HP	Type	Rank	User	HP	Type
1	Armandovivoni	541	2.0L	5	nanokpsi	501	2.4L
2	Super95awd	521	2.3L	6	FFTEC	477	2.0L
3	greentrbo_95gst	513	2.3L	7	GreddyGST	445	2.0L
4	Mitsutalon	503	2.3L	8	Ludachris	434	2.3L

Bigger Valves Needs Bigger Bore

Where an engine like the 4G63 is limited by the flow through the valves, making the stroke longer will not make much more power. The pent roof head is already full of valves. To get bigger valves the bore has to be bigger. But that's not an option for the DSM world, just another old saw that still cuts.

Conversion of Pressure to Torque

Some of the papers researched for this paper mention that longer rods have better leverage on the crankshaft for a longer time (11). However, for the top of the power stroke where most of the power is made, the shorter rod ratio, and the longer stroke have better leverage on the crankshaft to turn cylinder pressure into torque.

Figure 3 at right depicts two exaggerated examples, one of a short rod, long stroke and one long rod short stroke engine. The crank angle is the same for both examples. The heavy red lines are right angles from the alignment of the rod to intersect the center of the crankshaft. The length of the heavy red lines indicates the relative leverages on the crankshaft for any force from the piston. An interesting thing to notice here is that the side loading of the stroker motor is not all a bad thing. Think of it as the piston leaning on the cylinder wall to get a better push on the crankshaft.
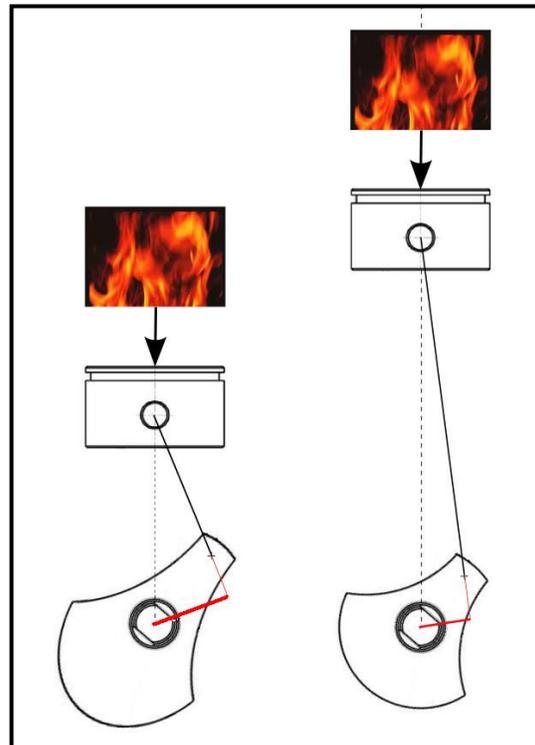
Figure 3 Leverage on the Crankshaft

In the bottom half of the power stroke the higher rod angle of the stroker is the wrong direction and makes less torque for any given cylinder pressure. The good news for the stroker is that the cylinder pressure is less in the bottom half of the stroke by about the ratio of the static compression. For example, with a 10:1 static compression ratio the pressure at the bottom of the stroke will be less than 10% of the pressure at the top of the power stroke. The hotter the fuel burn the greater the pressure drop ratio as the temperature of the gas goes down.

Figure 4 below is a plot of the torque generated from a constant 100 pounds of force on the rod. Friction losses from side loading are neglected. The 100 pounds is held constant for the full power stroke to isolate the effect of the rod ration and stroke. The theoretical 2.3L engine with 1.7 rod ratio is included to isolate the effect of the rod ratio from longer

stroke of the 2.3L stroker.   The plot shows pretty much what the exaggerated examples of Figure 3 above suggests. The longer stroke always has more torque than the shorter stroke. The shorter rods have more torque in the top half of the power stroke and the long rods have more torque in the bottom half.
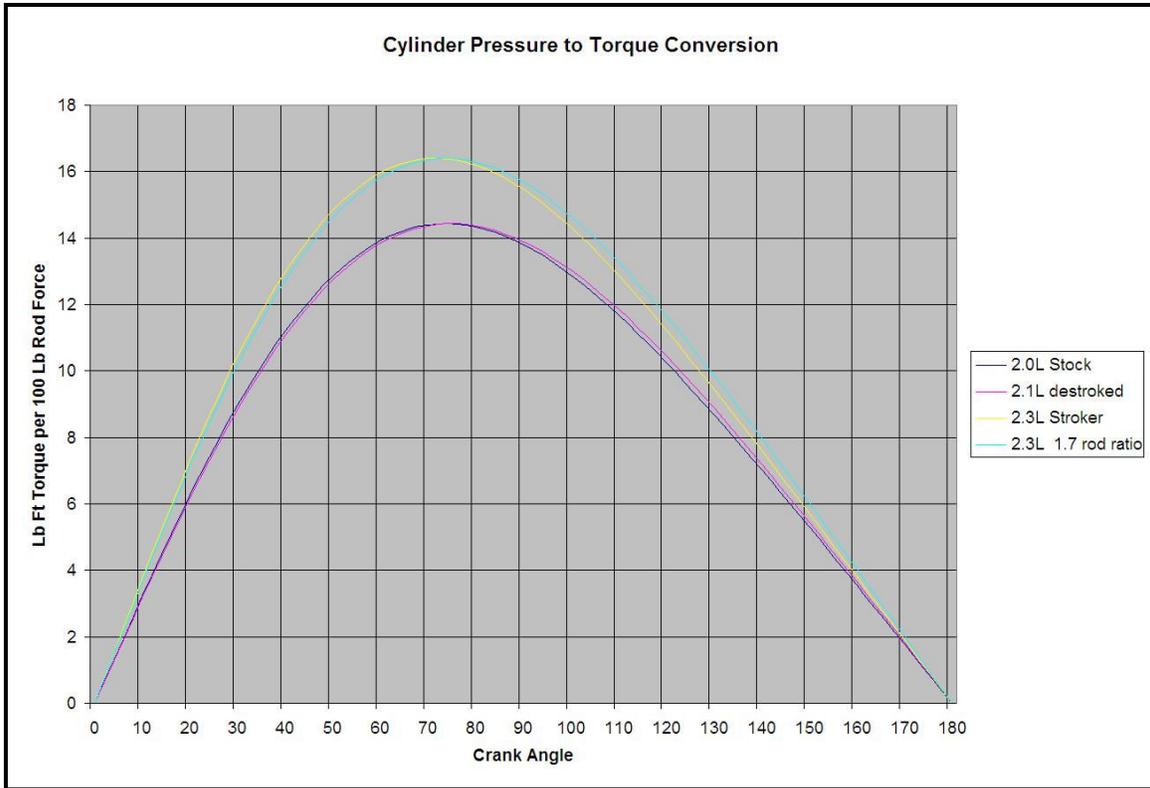

Figure 4 Cylinder Pressure to Torque Conversion

Boost is Good

Well maybe a lot has changed, but the pent roof combustion chamber dates back to before WW II. Figure 5 below is a cross section drawing of a V-1710 Allison engine designed in the 1930s with overhead cams and four valves per cylinder in a pent roof combustion chamber.

The Allison engine powered the P38 Lighting and the early Mustang P51. Later Mustangs were powered by the Rolls Royce Merlin engine.

Both the Merlin and Allison engines were 27 liter V12 engines with 4 valves per cylinder.   The Allison was more efficient and made more power at the same boost with its pent roof combustion chamber compared to the Merlin with its flat topped chamber.

> **Even with the superior pent roof combustion chamber the Allison lost out to the Merlin's better boost design.   Boost is good.**
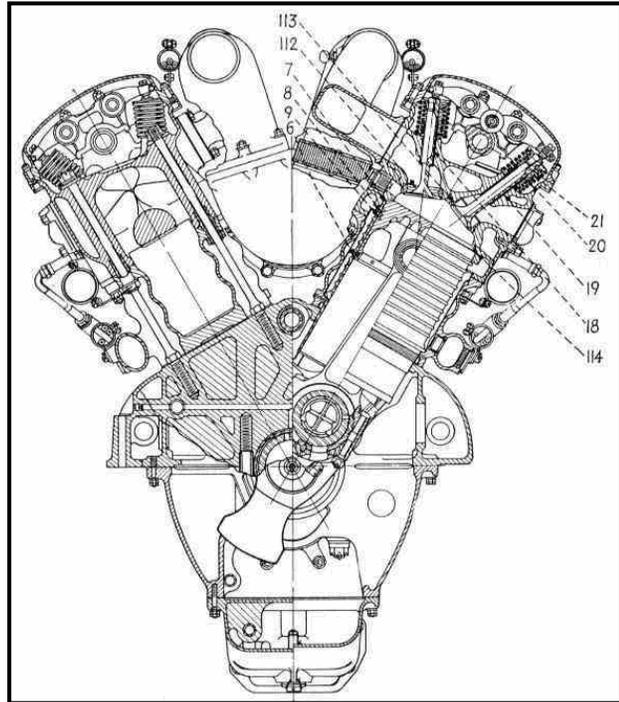
Figure 5 V-1710 Allison Engine

Even with the superior pent roof combustion chamber the Allison lost out to the Merlin's better boost design.   While the Allison had a single stage of turbo boost (Like the 4G63) the Merlin had two stages of supercharging with an intercooler between the stages and an aftercooler before the engine. The Merlin could also control boost from the cockpit taking full advantage of the higher octane fuel as it arrived from the U.S.   Because of the better boost system the Merlin engine is legendary for winning the air war in Europe while the superior Allison design is known today as an upgrade engine in tractor pulling competition. Boost is good.

Heat of Compression

Removing the heat of compression is crucial to performance of the 4G63T. The effect of larger intercoolers is a regular topic in the DSM forums but heat of compression is seldom explained.

Basically, all of the work required to compress air goes into the air in the form of heat. Even with a 100 percent efficient compressor the compressed air will be at a higher temperature than it started. If air is compressed with a piston in a cylinder and the heat of

compression is not removed the final pressure will be more than the static compression ratio by an exponent of 1.4.   This means that with a static compression of 10:1 the resulting pressure will increase by a factor of 10^1.4 or over 25:1 unless some of the heat of compression is removed. With a cold engine most of the heat of compression will be removed from contact with the block and piston and the pressure increase for a 10:1 engine will be "only" about 15 or 20 to one (30).

The table below is from a few quick calculations on the scratch pad of the spread sheet comparing the 8.5:1 CR 4G63 with the latest Corvette engine with 10.9:1 CR. The calculations based on the ideal gas laws shows that with a moderately effective intercooler a 4G63 with three bar of boost will have a lower charge temps than the normally aspirated small block with over twice the charge pressure.

Table 2 Effect of Heat of Compression

Condition	Charge Temp	Charge Pres
Small Block 10.9:1 CR no boost	948	402
4G63 8.5:1 CR 3 bar boost, intercooled	816	867
4G63 8.5:1 CR 3 bar boost **not** intercooled	1457	1354

The Corvette Z06 engine with the 10.9:1 CR is on the ragged edge of detonation with 93 octane fuel.   Higher charge temps or lower octane will cause destructive detonation. The condition of the 4G63 without an intercooler shows the effect of trying to run 25.5:1 CR (3 * 8.5) without removing the heat of compression Boost is good, but only with a working intercooler.

The 4G63 is a turbo engine, not an engine with an added turbo. In the original 2.0L size my Talon was gutless until the turbo spooled. And in the mountains it could barely get out of its own way while off boost. Although mostly ignored in this paper comparing the stroker to the 2.0L, the turbo and intercooler are integral to the engine design.

RPM LIMITATIONS OF STROKERS

Piston Speed

In the research for this paper piston speed is often mentioned as a limiting factor in engine RPM but mean piston speed does not take into account the effect of rod ratio in changing peak piston speed. For example the stock 2.0L 4G63 and the 2.1L destroked version have the same mean piston speed but at 8000 RPM the peak speed for the 2.1L is 0.4 m/s lower and the peak velocity comes one degree later. Figure 6 below shows the difference in piston velocity for three 4G63 versions at 8000 RPM. The chart makes it clear that the increased stroke of the 2.3L has much larger effect on piston than the longer rod of the 2.1L
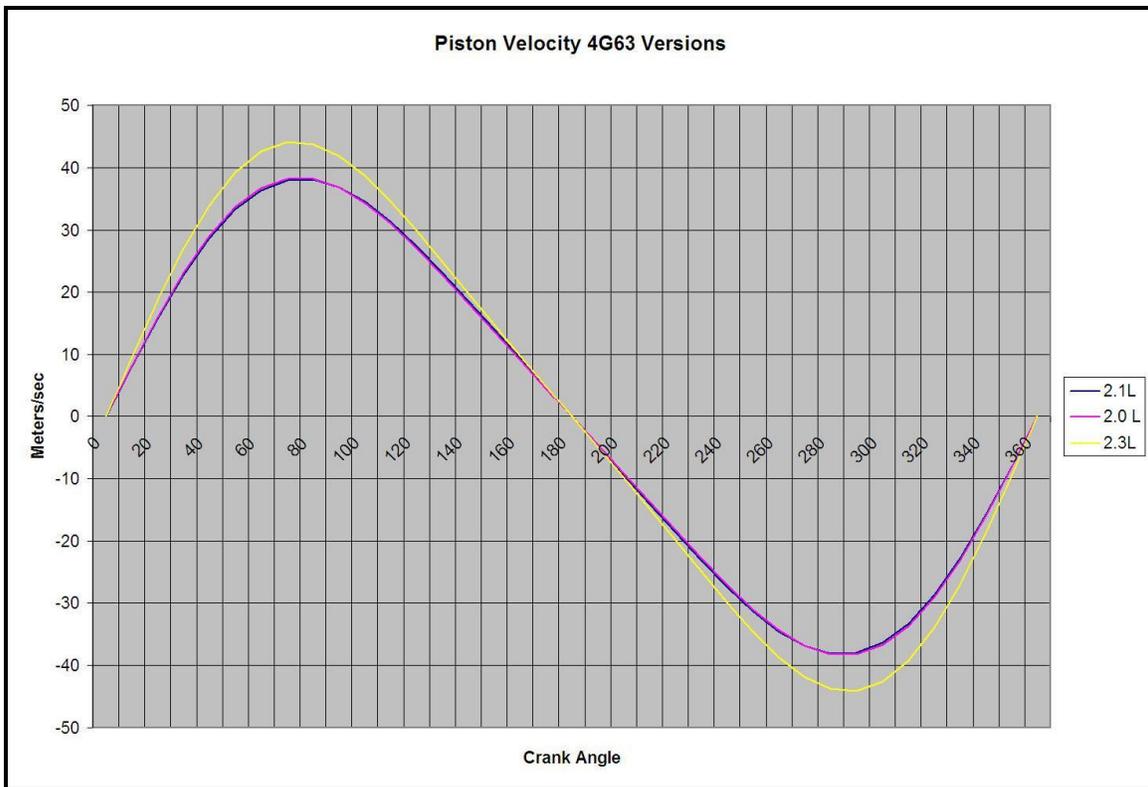
Figure 6 Piston Velocity at 8000 RPM

Piston velocity either mean or peak may not be the best measure of maximum engine RPM. Friction losses from piston speed increase linearly with RPM but forces due to acceleration increase exponentially with RPM. Ref (5) proposes piston acceleration as the limiting factor in maximum RPM, not piston speed. That article lists 100,000 ft/sec^2 (30480 meters/sec^2) as the upper limit where the rings start to flutter. That "limit" is about 13 percent lower than the stock 2.0L 4G63 at the stock Rev limit (35100 meters/sec^2 at 7500 RPM).

Figure 7 below shows the peak piston velocity for three versions of the 4G63 and the theoretical ten meter rod version.   Four things stand out from figure 7:

1. The 2.3L stroker version has significantly higher piston acceleration than the stock 2.0L.
2. The small change from 1.7 rod ratio to 1.8 of the destroked version makes a small but measurable difference.
3. Rod ratio does make a difference. The theoretical ten meter rod version has significantly less acceleration than the stock 2.0L.
4. The relationship of piston acceleration to RPM is exponential as the difference between engines is greater as RPM increases.
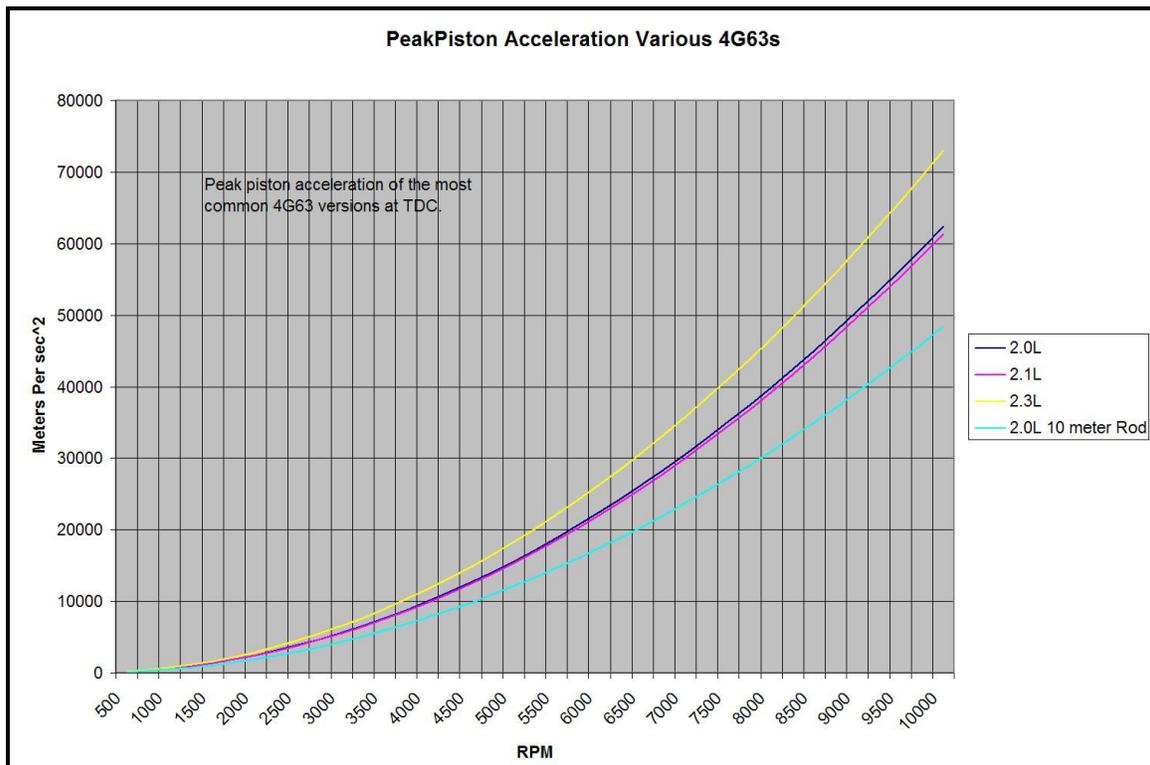
Figure 7 Peak Piston Acceleration 4G63 Versions

Figure 7 shows that the piston acceleration limit comes sooner for stroker than for the stocker.   The 2.3L stroker has about the same piston acceleration at 7400 RPM as the 2.0L stock engine at 8000 RMP.

Rod Tension

Engines using high boost or nitrous injections create very high torque at moderate RPM causing rods to fail in compression.   As the RPM's go up the rods have higher tension loads.   Rod strength in tension is one of the limits of RPM (11). Figure 8 at right shows a rod failure in tension.

Figure 8 Rod Failure in Tension

Ok, Ok figure 9 at right has nothing to do with RPM limits of the 2.3L stroker. But it does show why the word "interference" is used to describe engines. Figure 10 below shows a rod bent from failure in compression.

Figure 9 Gratuitous Image

Rod tension can be calculated in pounds by converting the piston acceleration rates to Gs (where 1G is the acceleration due gravity) and multiplying by the reciprocating mass in pounds.   Rod tension increases linearly with reciprocating mass and exponentially with RPM. Figure 11 below shows the force on the rod for the stock 2.0L 4G63 and three versions with light weight pistons and rods. Note that the peak force for all versions is at TDC and that the lighter than stock weight of the Eagle/Wiseco parts compensates for the higher accelerations of the 2.3L stroker.

Figure 10 Rod Failure in Compression
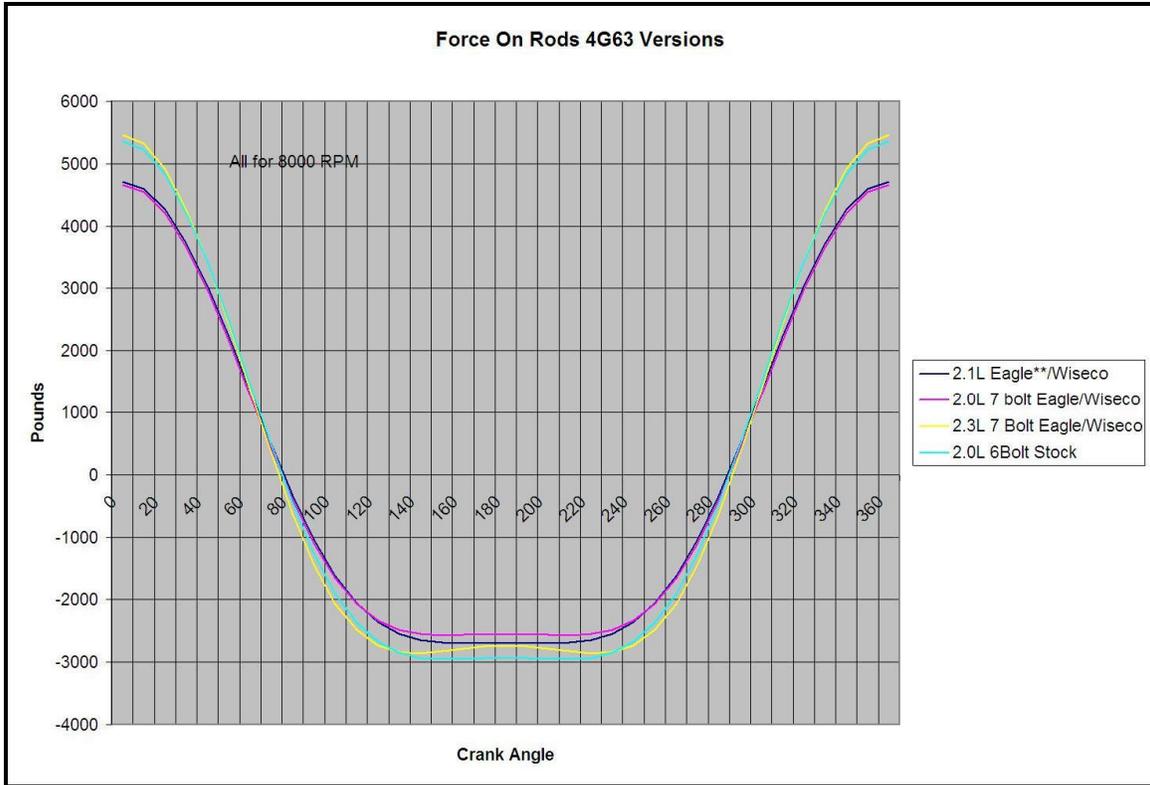
Figure 11 Force on Rods by Crank Angle

While figure 11 above shows the rod forces at different crank angles, Figure 12 below shows the peak rod force by RPM up to 10,000 RPM. By the measure of rod force, the 2.3L stroker is so close to the 2.0L stocker that only one line shows up.
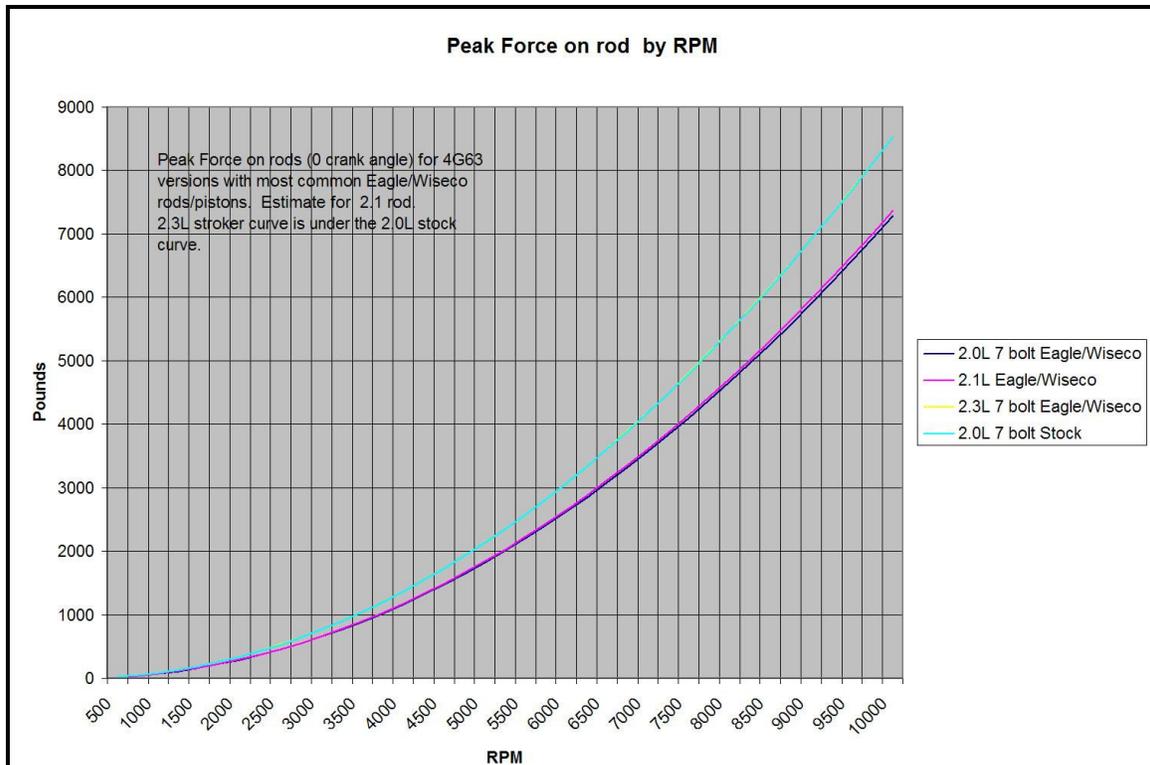
Figure 12 Peak Force on Rods by RPM

Again, by this measure the stroker has a lower rev limit that the 2.0L with the same parts. The 2.3L stroker has the same rod force at just less than 7400 as the 2.0L at 8000 with the light weight Eagle/Wiseco combination.

Friction Losses

Friction losses limit horsepower at high RPM's.   Many of the sites studied for this paper mention that low rod ratios increase side loading on pistons and increase maximum velocity and acceleration.   Friction losses are a triple negative for the stroked 4G63.

1. The longer stroke means higher velocities. Friction losses are about linear with velocity. (19).
2. The lower rod ratio means higher acceleration rates generating more forces on the rod/piston (17).
3. The lower rod ratio means a greater percentage of the rod load goes to side loading. (6, 11) Friction losses are linear with loading (19).

The web has many sites discussing the effects of rod ratio and friction but a (too long) search failed to turn up any hard formula for losses at different rod ratios.

To compare friction losses between the 4G63 versions the calculations here were based on:

1. Friction losses are linear with velocity. The Mechanical Engineers Handbook says that the losses are slightly lower at high speeds but did not give an exponent.
2. Friction losses are linear with loading, which is straight from the literature.

3. Side loading on the piston is proportional to the sin of the rod angle. When the rod is vertical (sin = 0) there is no side loading even with high rod loads. If the rod angle could reach 90 degrees (sin = 1) all of the rod load would be side load.
4. Without a known coefficient of friction the condition for a 2.0L stock 4G63 is set arbitrarily to 100 percent.

In figure 13 below and elsewhere in this paper Side Loading Friction (SLF) is calculated as:

$$\text{SLF} = \text{Sin (Rod Angle)} * \text{Force} * \text{Velocity} * K$$

The absolute values of the components were used in the calculation to simplify the chart by ignoring the difference between left side loading and right side loading. The Constant K was selected to make the maximum SLF to be 100 for a stock 2.0L at 8000 RPM.

With the correct constant, the units of SLF could be horsepower because the components are in the form of pound feet per second. When comparing different versions the percentage difference in SLF corresponds to horsepower lost due to side loading on pistons.

Side loading is difficult to visualize because the components of the calculation are different shaped curves that peak at different times. Acceleration peaks at zero velocity and rod angle peaks when acceleration is low. Figure 13 below shows the components of Side Load Friction (SLF) components for a 2.0L 4G63 with stock rods and pistons.
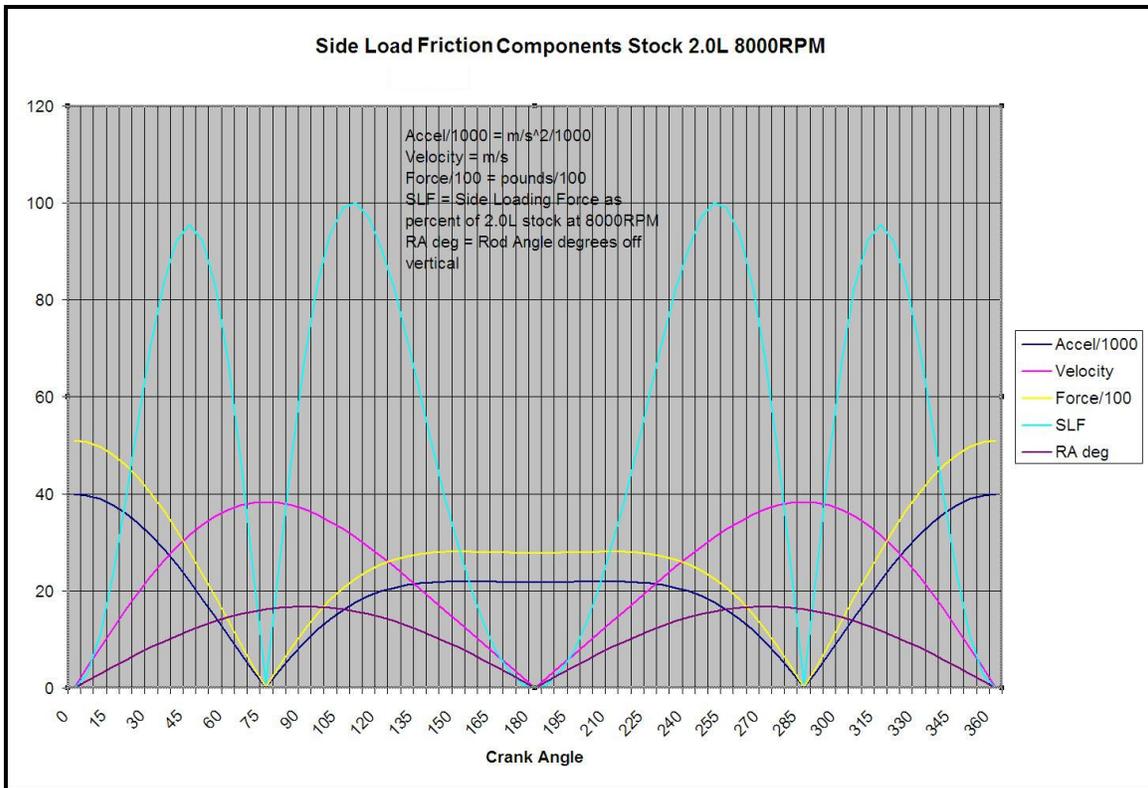
Figure 13 Side Load Friction Components – 2.0L

Wherever the term SLF is used in this paper it can be read as percent of the side load friction losses of a stock 4G63 at 8000 RPM.

Note that SLF accounts only for friction losses due to inertia, and is intended only for comparison of the 4G63 versions (stroked or not). There is of course some friction from the rings even with no side loading, and of course there is a lot of side loading from the forces of combustion.

Figure 14 below charts SLF by RPM for the stock 4G63 and three common versions with lightweight pistons and rods. The calculations were performed at the crank angle with the peak SLF (105 degrees for the stroker and 110 degrees for the others.) Note that the side load friction losses are exponential with RPM and linear with reciprocating mass.

Figure 14 shows clearly one major reason why the 2.1L destroked version revs better than the 2.3L stroker. At 8000 RPM the stroker has the same SLF as the stock 2.0L at 9000 RPM or the 2.1L at 9500 RPM.
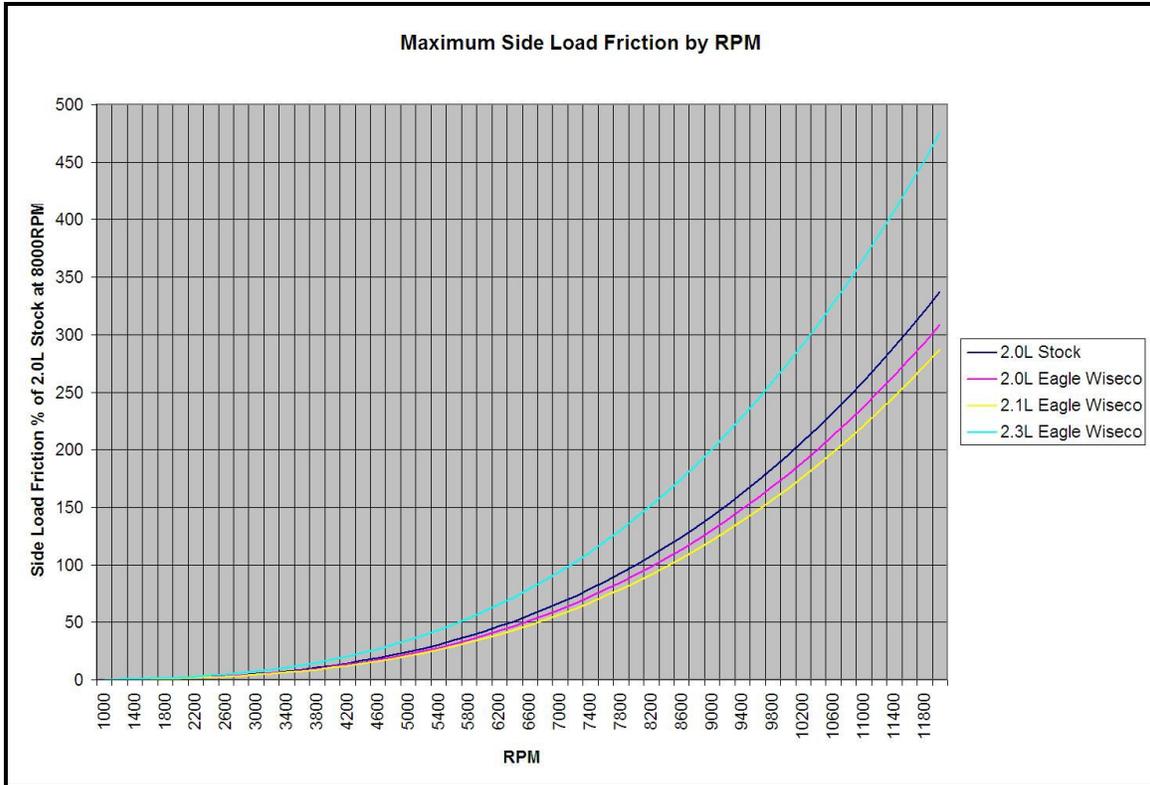
Figure 14 Side Load Friction by RPM

Figure 15 below shows the components of the SLF calculation for the 2.3L stroker at 8000 RPM. Note that compared to the 2.0L stocker in Figure 13, the rod angles are higher, the acceleration rates are higher, and the forces are higher. The three negatives for the stroker result in almost 30 percent more SLF losses than the stock 2.0L at 8000 RPM. You could expect that a NASCAR engine builder would kill for less.
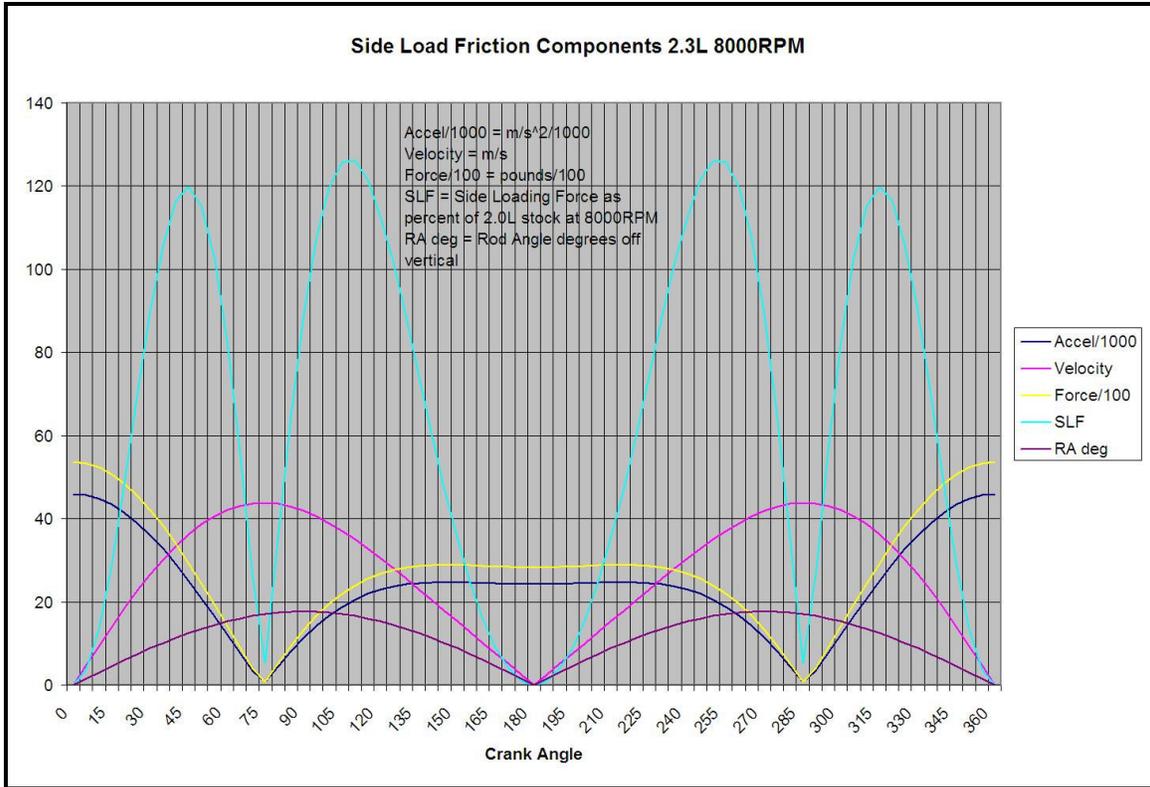
Figure 15 Side Load Friction Components – 2.3L

WHATS AN "IDEAL" ROD RATIO

The stock 4G63 rod ratio is 1.7 which many technical papers describe as "ideal". The destroked 2.1L version has a rod ratio of 1.84 (longer rods and the same stroke) is often referred to as "better" because it can rev higher. The 2.3L stroked 4G63 version has a rod ratio of 1.5 (stock rod length and longer stroke) and is usually referred to as an "unfavorable" rod ratio in the DSM forums.

Longer rods have lower harmonic imbalance and let the intake valves open a little more before reaching peak velocity. Longer rods have less piston side loading which means less friction. For engines designed for top end power, the longer the rod the better. Shorter rods have higher intake velocity at low RPM and better leverage on the crankshaft. For engines needing low end torque, the shorter the rod the better. So the "ideal" rod ratio should be some trade off between the two extremes.

Figure 16 below shows piston acceleration rates for four different rod ratios, all with the same stroke (88mm) and for 8000 RPM. Note that for all four ratios, the acceleration rates at BDC are about half of TDC.   Also the differences between TDC and BDC are less for longer rods. The longest rod has the lowest acceleration at TDC and the highest at BDC.
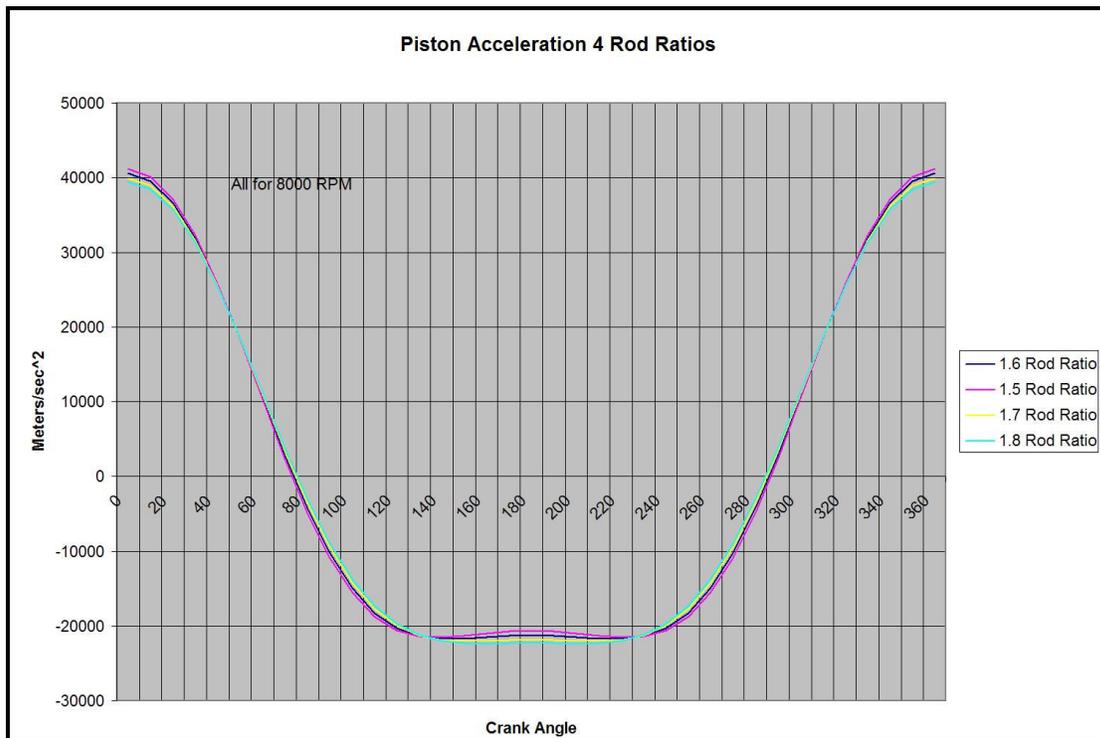
Figure 16 Piston Acceleration 4 Rod Angles

Figure 16 shows the benefit of longer rods for lower harmonic imbalance, but it looks like "the longer the better". Zooming in to the area around BDC shows a better picture of the forces involved.

Figure 17 below is zoomed in to show the variations in Acceleration around BDC. Note that below the rod ratio of 1.7 there is an extra "hump" in the acceleration curve between 140 degrees and 220 degrees. Above the rod ratio of 1.7 the curve will continue to drop lower and lower but will remain a smooth curve without the hump.
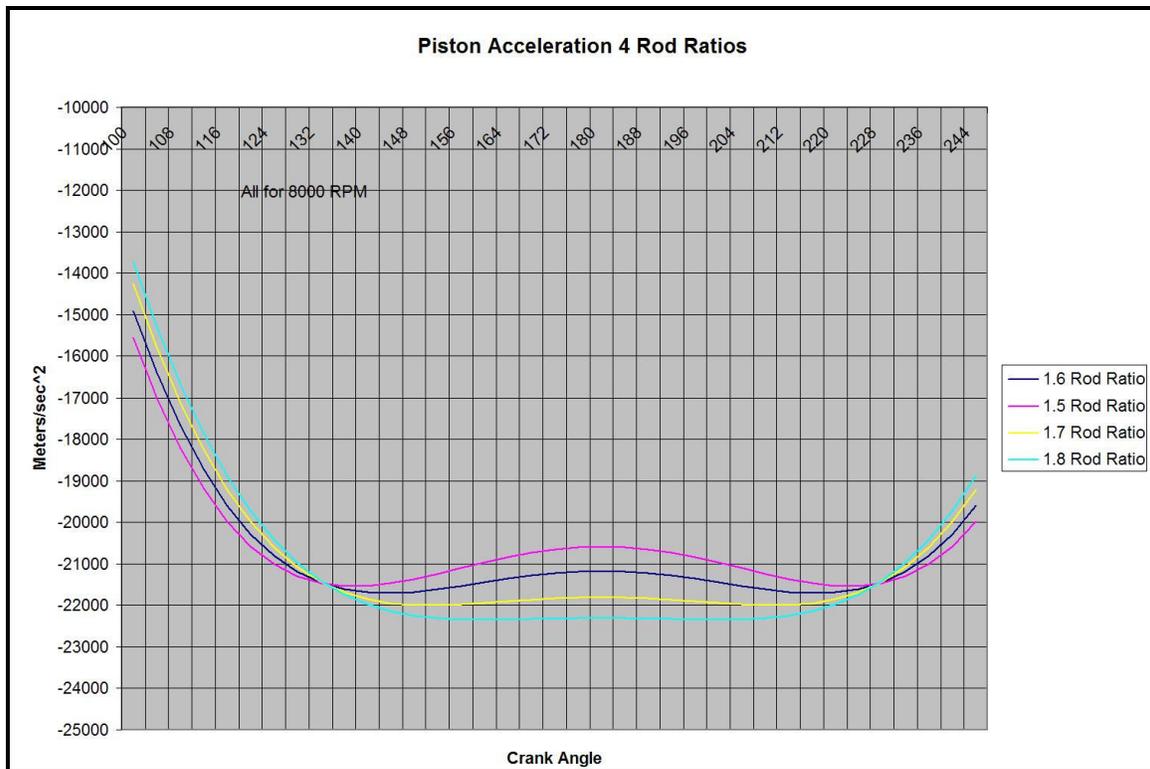
Figure 17 Piston Acceleration 4 Rod Ratios

So there is the "ideal" rod ratio. The 1.7 ratio is the shortest rod that will provide a smooth transition in force at BDC. Longer rods have diminishing returns and require taller blocks. Shorter rods have more high frequency imbalance around BDC.

Note that the little hump in the 1.5 ratio curve at 8000 RPM is about 1000 meters per second per second. That converts to 102 Gs, or with Eagle rods and Wiseco pistons about 146 pounds applied to and removed from the main bearings in about 1.6 milliseconds. That's pretty close to a hammer blow delivered 133 times per second. Smooth transitions are much more important as RPM is increased and as piston/rod weight is increased.

Anyway with 10 meter rods, my 2Gb Talon would never fit under the overpasses. And it would look really silly.

STROKERS AND ENGINE BALANCE

Figure 17 above shows the effect of rod ratio on piston acceleration but the 2.3L stroker doesn't have a shorter rod, it has a longer stroke. Figure 18 below shows how much higher the acceleration is for stroker pistons compared to stock or destroked 4G63 engines.
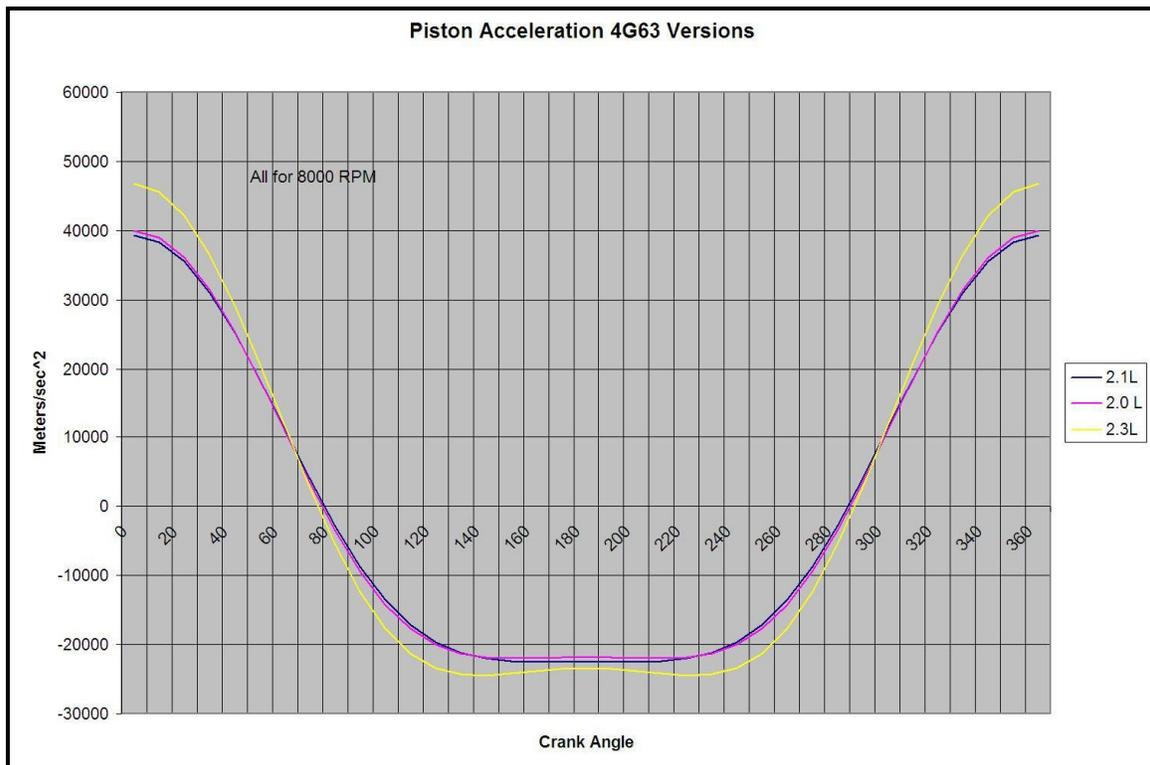
Figure 18 Piston Acceleration 4G63 Versions

Imbalance in Pounds

To get the effect of the imbalance in pounds, the acceleration rates in Figure 16 were converted Gs and multiplied by the mass of light weight pistons and rods. Figure 19 below shows the harmonic imbalance in pounds for a stock 4G63 and three common after market builds. Note that the 2.3L stroker has the greatest imbalance. Even with light weight reciprocating parts, the stroker has more harmonic imbalance than the stock engine. As expected, the 2.1L version is better balanced, about 50% better than the stroker.
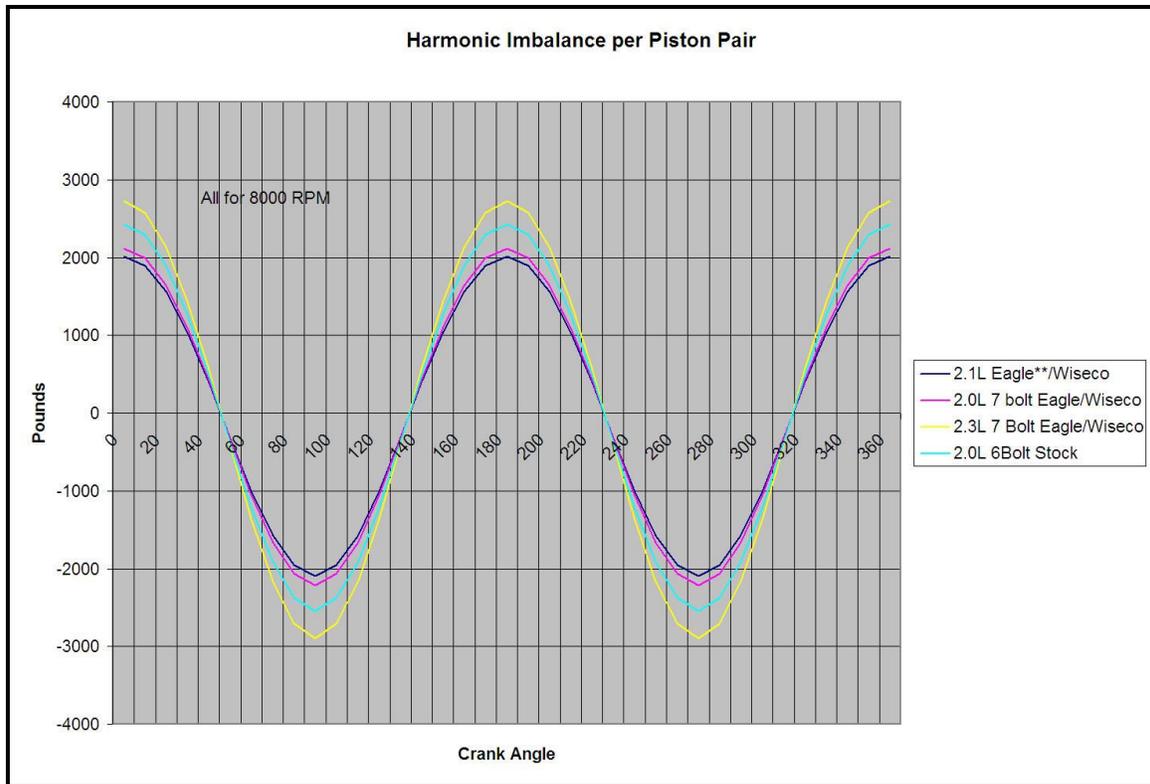
Figure 19 Harmonic Imbalance 4G63 Versions

(**Note that the actual weights of 162mm rods were not available on the web so the rod weight used for the 2.1L version was for Eagles * 162/150.)

The reason for the term "second order harmonic" is clear in the figure above. There are two full cycles of force applied to the main bearings for every one revolution of the crank or two times the frequency of the cranks rotation.

Imbalance from Mismatched Parts

The charts of harmonic imbalance were calculated as the difference in forces between two pistons 180 degrees apart with all else the same. The static imbalance calculations are for the differences in forces between two pistons 180 degrees apart, with different weights, and with the harmonic imbalance subtracted out. The result is just the forces resulting from mismatched weights in the reciprocating parts. Figure 20 below shows the effects of a weight mismatch of 5, 10, and 20 grams on a 2.0L engine at 8000 RPM.
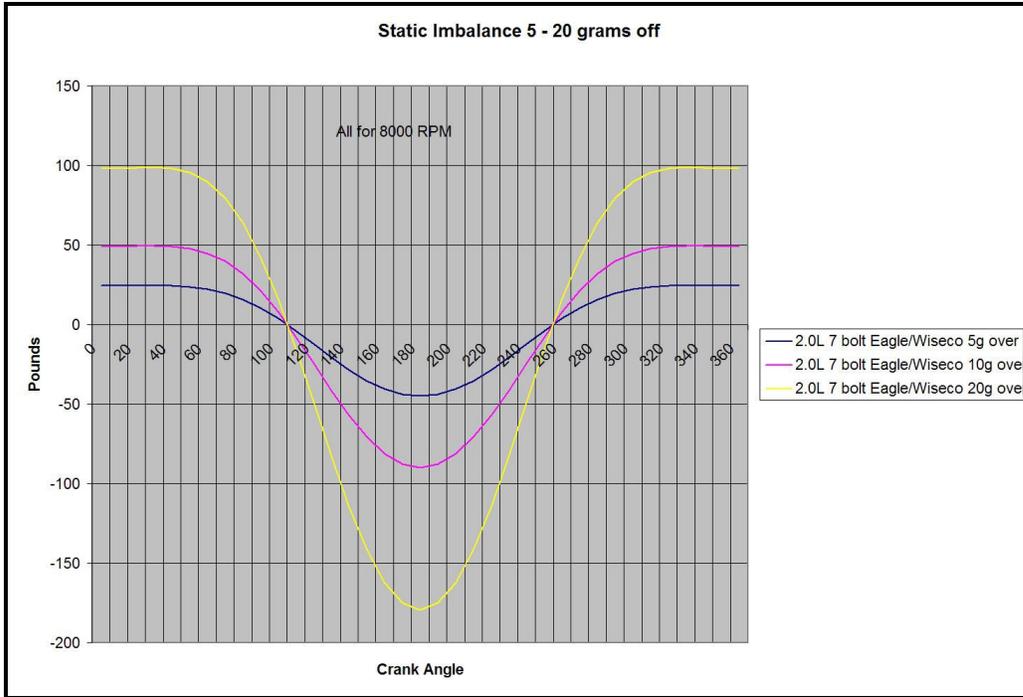
Figure 20 Balance Forces for 2.0L 8000 RPM

Figure 21 below is the same as Figure above except for a 2.3L Stroker.
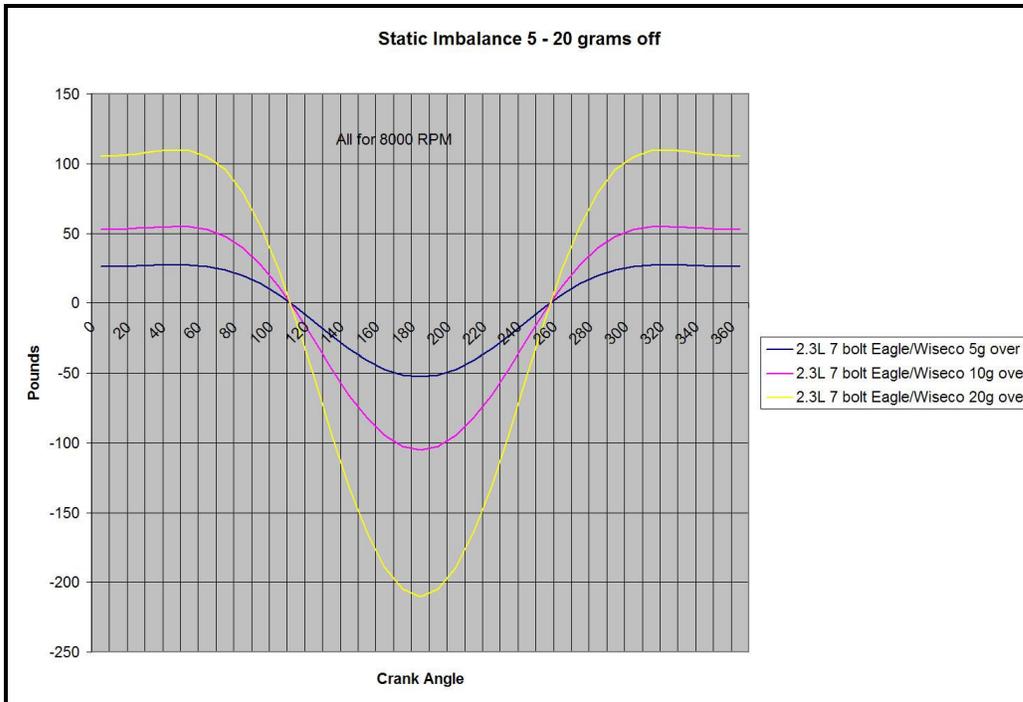
Figure 21 Balance Forces for Stroker 8000 RPM

Note that the imbalance for a stroker is about 20 Percent more than for a 2.0L. As expected with the longer stroke and higher accelerations. Figure 22 below is for 4000 RPM on a Stroker with all else the same as the 8000 RPM chart.
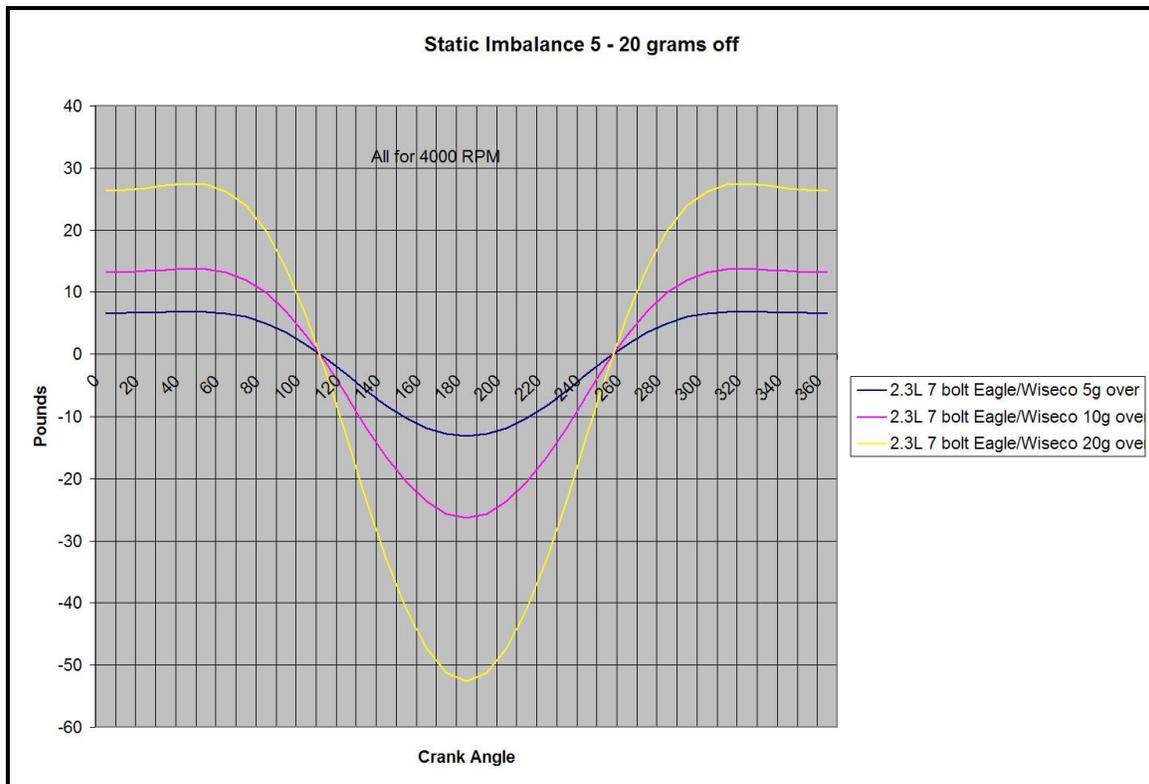
Figure 22 Balance Forces for Stroker 4000 RPM

Note that the vibration from static imbalance is at the same frequency as the crankshaft. Doubling the weight mismatch doubles the resulting force, and doubling the RPM, increases the vibration force by a factor of four.

2.3L strokers are more sensitive to engine balance than the stock 2.0L and the higher the RPM, the greater difference it makes.

STROKE AND CAM SELECTION

Aggressive cams have more overlap time where both the exhaust valves and intake valves are open. At higher RPM's, with higher intake and exhaust stream velocities there is a "ram" effect where the inertia of the intake air keeps air going into the cylinder while the exhaust stream is "scavenged" and continues going out of the cylinder. At lower RPM's the velocity of the two streams are lower and the exhaust flows back into the cylinder during the overlap period and the intake stream stops causing a rough idle and reduced low end torque at best. (5)

Camshafts are designed to meet a goal of higher horsepower or better low end torque by controlling where cam events occur in relation to piston location. However, the cam lobes have no idea where the pistons are, they only know about crank angle. Camshafts designed for a 4G63 engine assume a stroke of 88mm and a rod length of 150mm. Changing either the stroke or the rod ratio changes the pistons velocity and location in relation to the crank angle. There is no reason to expect that a camshaft designed for a 2.0L 4G63 will have the same characteristics in a stroker.

Displacement

When a 4g63 is bored and stroked, the displacement can be increased 16 Percent from 1997 cc to 2323 cc. That 16 Percent bigger engine still breaths through the same intake tract.   More displacement, with the same intake tract means higher velocity and better low end torque with any given camshaft. Figure 23 below shows the significant increase in displacement of the 2.3L stroker compared to the other 4G63 versions.
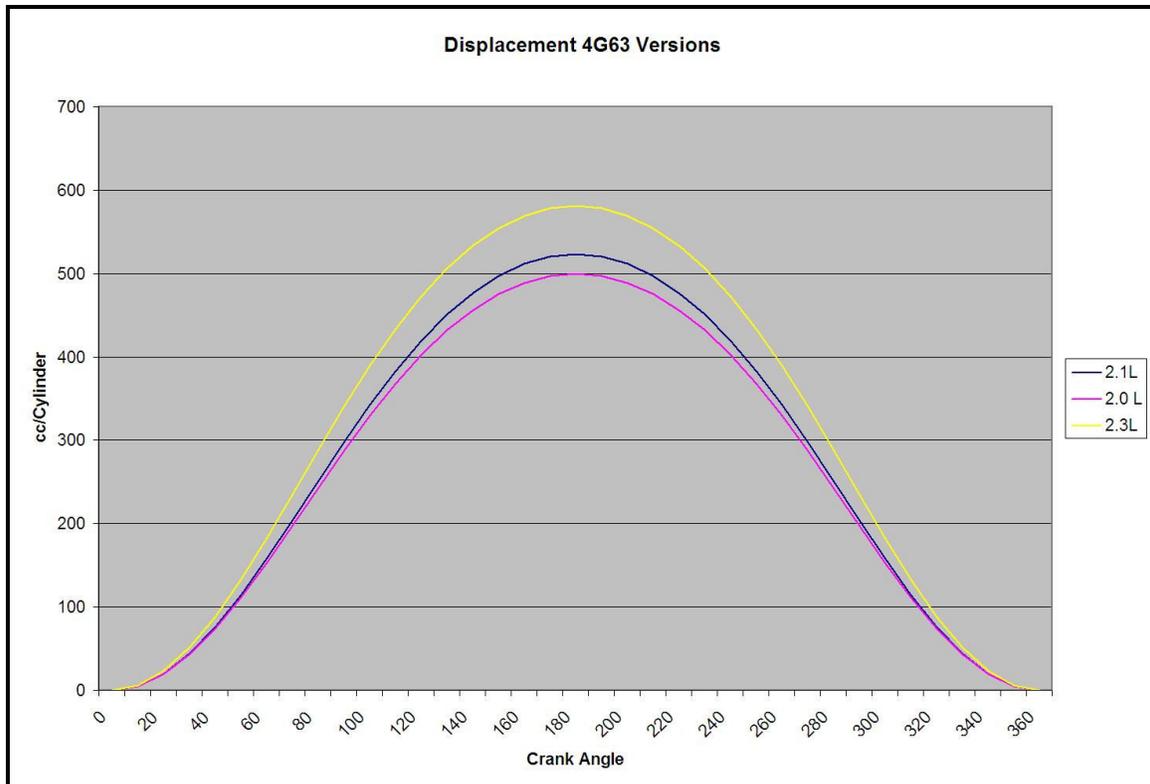
Figure 23 Displacement vs. Crank Angle

Piston Velocity

Another large factor in the velocity of the intake stream is piston velocity. Even with the same displacement, higher downward piston speeds during the intake cycle will cause higher intake velocity. With most of the increase in displacement of the 2.3L due to the longer stroke, the mean piston velocity will be higher than the 2.0L for any RPM. Figure 24 below shows that piston velocities are much higher for the 2.3L stroker. Note that if additional displacement is from wider bores, the piston velocity would be unchanged. All else being equal, longer strokes have higher piston velocity and are more tolerant to aggressive cams at low RPM.
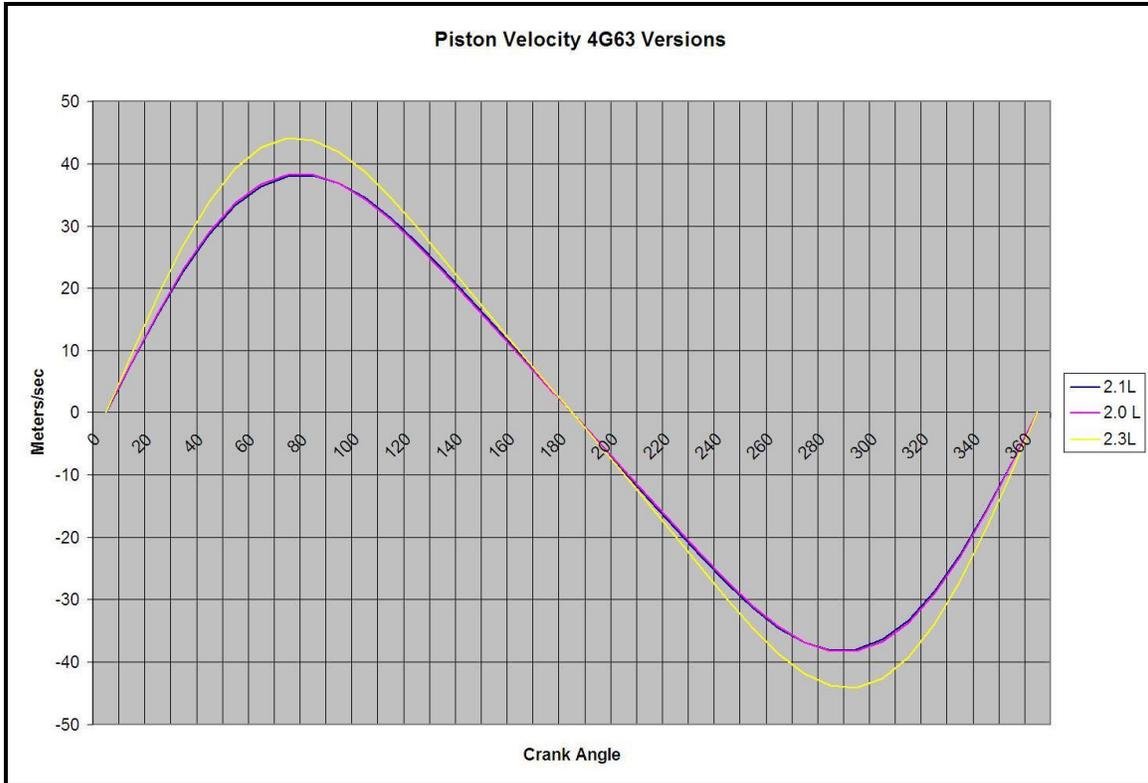
Figure 24 Piston Velocity 4G63 Versions

Rod Ratio and Piston Velocity

While stroke length has the major effect on piston velocity, the rod ratio also has an effect. For any stroke and RPM, the mean piston velocity is the same. However the rod ratio affects the peak velocity. Lower rod ratios have higher peak piston velocities and the peak occurs earlier in the intake cycle. Figure 25 below shows the piston velocities for four rod ratios in the area of peak velocity. The calculations were made with everything the same except the rod ratio.
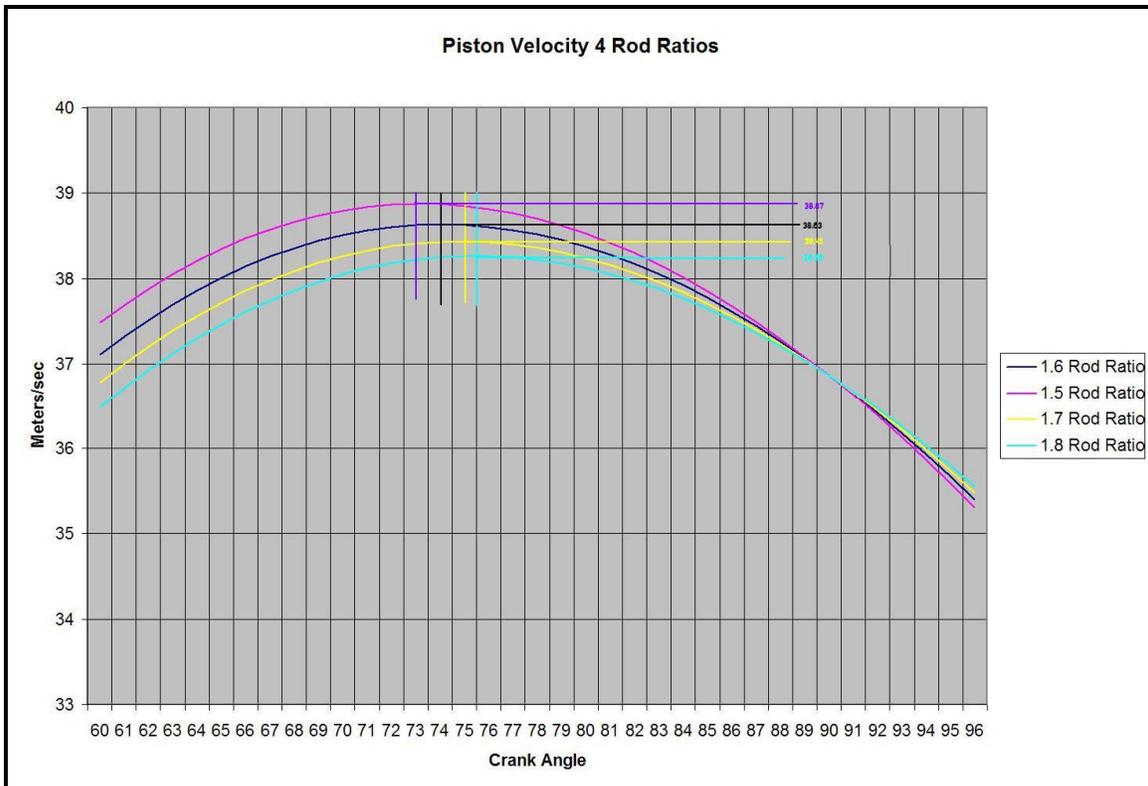
Figure 25 Peak Piston Velocity vs. Rod Ratio

The color keyed horizontal and vertical lines in figure 25 mark the crank angle where maximum velocity is reached and the difference in maximum between the four rod ratios. Note that in figure 25 above the 1.5 rod ratio of the 2.3L stroker reaches peak velocity about 2.5 crank degrees sooner and is about .7M/s higher than the 1.8 ratio of the 2.1L destroked 4G63. The earlier, higher piston velocity of the lower rod ratio provides higher intake velocity at low RPM's and better low end torque.

Another interesting point shown in figure 25 above is that at 90 degrees crank angle the piston velocities are the same for all rod ratios. In the lower part of the stroke the velocities are reversed, the longer rod ratios spend more time on the upper half of the stroke than the lower ratios. Note that the piston is half the way down the stroke at about 80 degrees ATDC depending on the rod ratio. A crank angle of 90 degrees is on the bottom half of the stroke.

Piston Position vs Rod Ratio
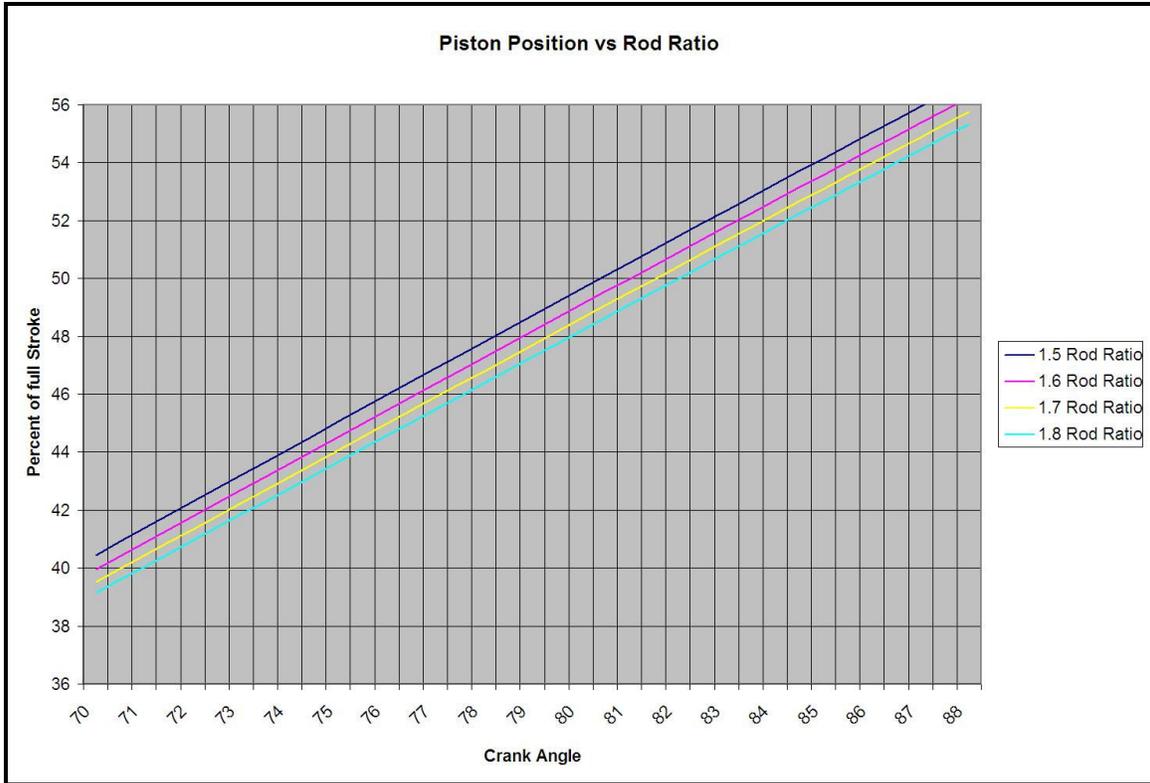
Figure 26 Mid Stroke vs. Rod Ratio

Longer Stroke and Lower Rod Ratio

All else being equal, longer strokes give higher piston velocities and lower rod ratios give higher piston velocities.   The 2.3L stroker combines the two.   Figure 27 below shows the piston velocities for different rod ratios, including the differences in stroke. The chart is for 8000 RPM but the differences will be the same percentage at all engine speeds.
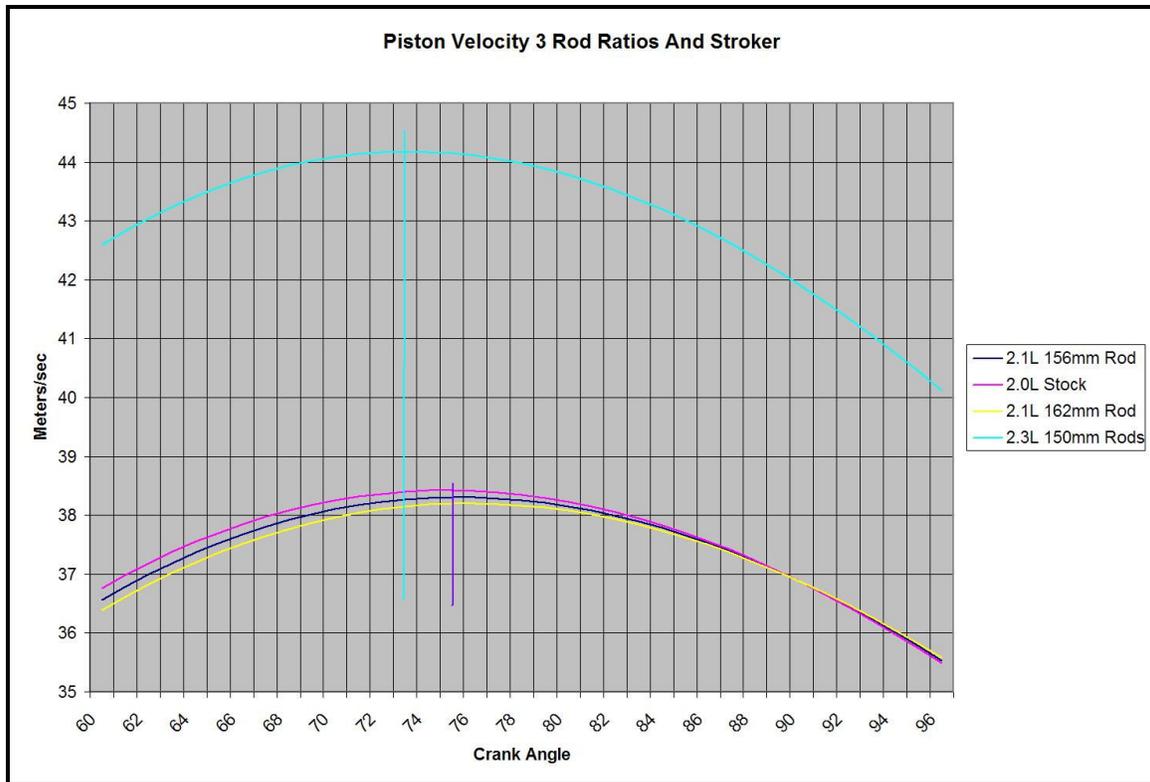
Figure 27 Piston Velocities 3 Rod Ratios and Stroker

The color keyed vertical lines in figure 28 mark the crank angles where the 2.3L and 2.0L stock versions reach peak piston velocity. Note that the 2.3L peak piston velocity is about 15 percent higher than stock and comes 2 degrees earlier. As long as the head and valves flow the air, the stroker will have more torque than the stock 2.0L.

Effective Compression Ratio

The compression ratio that is normally used discussing 4G63 specifications is the (30) Static Compression Ratio (SCR). SCR is simply the sum of displacement and head volume divided by head volume. SCR is a useful number because it doesn't change with the camshaft profile. However in any real 4G63 version, the intake valve doesn't close until several degrees after BDC. The delay in closing the intake valve allows the ram effect in the intake air stream to continue flowing air into the cylinder even while the piston is going up. (There's that velocity effect again). The delay allows more air in the cylinder but compression can't start until the intake valve closes. In any 4G63 version there will be some uncompressed volume because the piston is not at the bottom of the stroke when compression starts. The term Dynamic Compression Ratio (5) is often used for the compression that happens from the time the intake valve closes to piston TDC. However piston position at the valve closing event does not change with engine operating

conditions so this paper will use the less popular term of Effective Compression Ratio (ECR). ECR is affected by the cam profile and the rod ratio. At lower RPM's air is forced back up the intake tract through the open intake valve as the piston moves up. At higher RPM's the velocity of the intake stream still has air going into the cylinder as the intake valve closes. Peak torque occurs about the point where the air stream is just stopping as the intake valve closes. At higher RPM's the volumetric efficiency starts to drop off as the cylinder does not get the full charge of air.

Effect of Cam Profile on ECR

High performance cams delay the closing of the intake valve to get more power at higher RPM's, raising the point of peak torque. The obvious trade off is lower ECR and less low end torque for higher performance cams. Figure 28 below shows the cylinder pressure vs. piston position for the stock cam, and HKS 264, and 272 cams. The 264 and 272 pressures are so close together that only one line shows, but the difference from the stock cam is clearly shown.
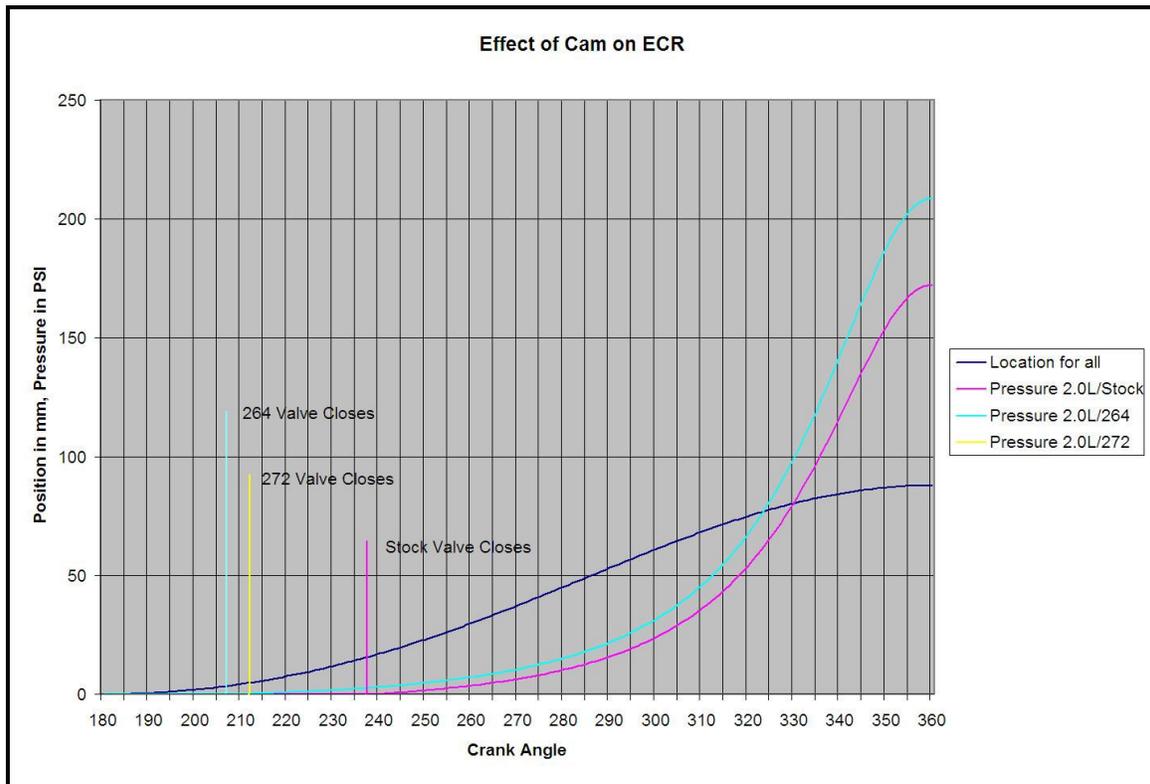
Figure 28 Cam Timing and ECR

Some posts on the DSM forums show the 2G stock intake cam closing 51 degrees ABDC but this paper uses 59 degrees according to the overhaul manual (29). (After production of these charts another "reliable" document was found showing differing cam timing for different codes as listed in the specifications section. Either way the stock intake cam stays open significantly longer than the two popular HKS cams.)

Effect of Rod Ratio on ECR

With lower rod ratios such as the 2.3L stroker the pistons move slower at the bottom of the stroke so have less uncompressed volume when the intake valve closes.   Figure 29 below is zoomed in to near TDC to show that with the 272 cams and 8.5:1 SCR the 1.5 rod ratio of the 2.3L stroker has about 1 percent higher cylinder pressure than the 2.1L destroked version.   The bottom, magenta, line is for the stock cam and 8.5:1 SCR. Figure 29 shows that the lower rod ratios do have a higher ECR, but that the effect is small.
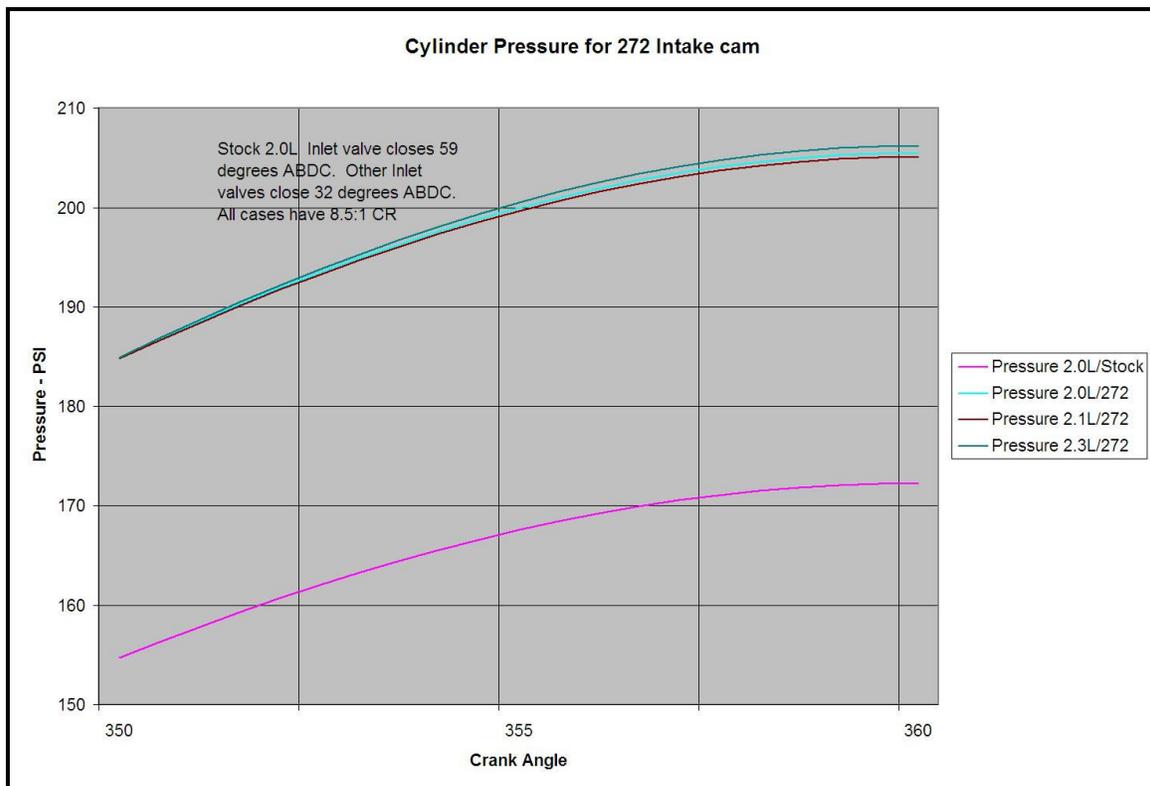
Figure 29 Rod Ratio and ECR

The Differences in ECR for three cams and three 4G63 Versions are summarized in Table 4 below. Note that the differences between cam profiles is larger than differences between rod ratios.

Table 3 Cam and Rod Ratio vs. ECR

4G63 Version	2.0 L		2.1L		2.3L	
	UV	ECR	UV	ECR	UV	ECR
Stock Cam (59°) 8.5 CR	93.7	6.90	100.4	6.87	104.5	6.97
264 Cam (28°) 8.5 CR	21.1	8.14	22.7	8.13	23.3	8.16
272 Cam (32°) 8.5 CR	27.6	8.03	29.7	8.02	30.4	8.05

Rod Ratio and Stroke Effect on Overlap

The effect of rod ratio on piston position is mentioned in six of the references for this paper (5,6,8,11,16, and 17) so it must be a real factor. Right? The paper from GrapeApeRacing (11) was kind enough to call the effect "small".

To show the effect of rod ratio on the period of valve overlap, crank position from 30 degrees before TDC to 30 degrees after TDC was charted in figure xx below. The valve events of intake valve opening and exhaust valve closing were added to the chart for stock 2G cams, HKS 264, and HKS 272 cams. The valve overlap area was then marked for each of the three cam sets. The chart is a little busy but helps us visualize the cam events better than raw numbers in a table.
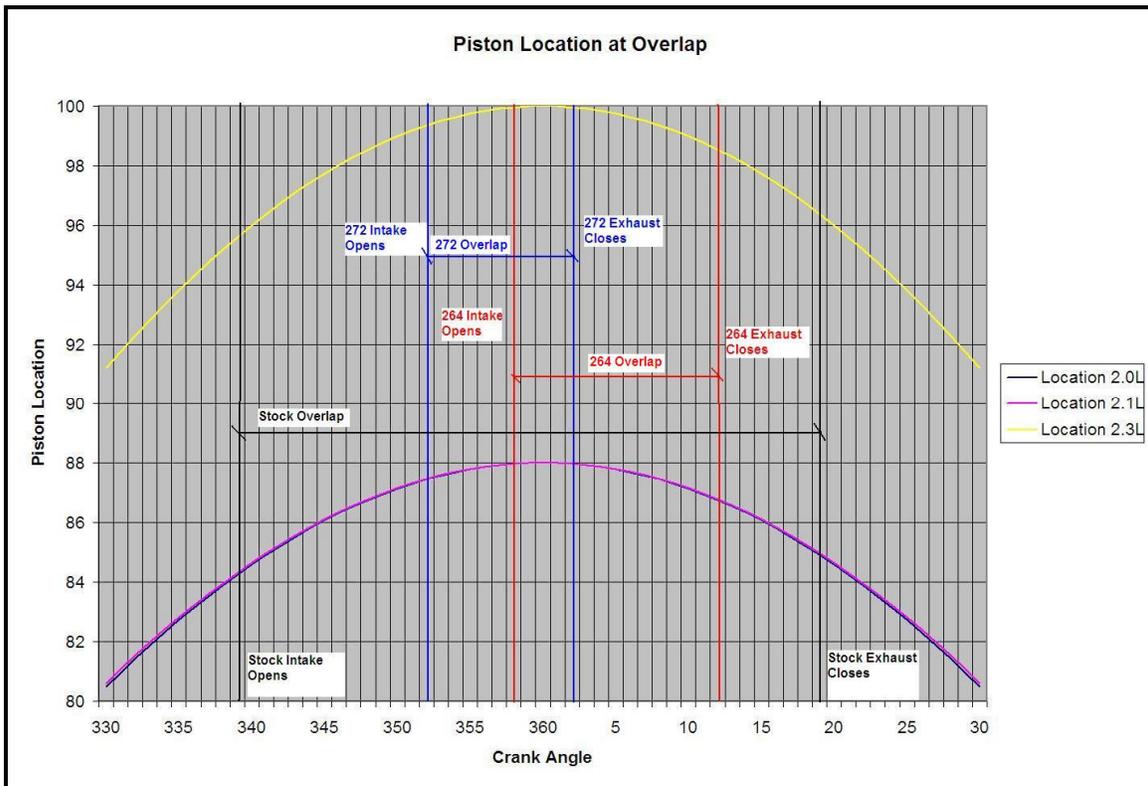
Figure 30 Valve Overlap 4G63 Versions

Figure 30 clearly shows that the rod ratio difference between the stock 2.0L and the 2.1L destroked version is too small to affect cam selection. The 2.1L piston is moving slower than the 2.0 just as the theory says, but the two curves are almost on top of each other.

Equally clear from figure 30 is that the 2.3L stroker piston locations and speeds are significantly different from the stock 2.0L. Where the stock piston with the stock cam

moves up 3.5mm after the intake valve opens, the stroker piston moves up about 4.3mm. That's about 20% more piston motion (and flow).   Cam theory is beyond this author but it looks to me that the 2.3L stroker will respond to cam changes like the 2.0L but at a much lower RPM.

ROD RATIO STROKERS AND ROD SELECTION

Stroking the 4G63 changes the nature of the engine in the direction of more torque at lower RPM. The best trade off between connecting rod strengths in compression and tension for a high revving drag engine may not be the best for the 2.3L. The 2.3L stroker's lower rod ratio increases the bending load on the rods due to the higher rod angle. The table below shows that the 2.3L will have about 14 percent more of the total compression load as bending force.

Table 4 Forces on Rods

Condition	2.0L	2.1L	2.3L
Maximum rod angle at 90 degrees crank angle	16.8	15.6	19.1
Sin of maximum rod angle	.2933	.2716	.33333
Percent of stock rod bending force	100%	92.6%	113.6%

Considerations for rod selection for the 2.3L include:

1. Since the 2.3L stroker will generate more torque but not rev as high as the 2.0L 4G63 connecting rods need to be designed in favor of withstanding compression loads.
2. The 2.3L stroker has a greater rod angle than the stocker causing more bending force on the rods than the 2.0L calling for rods designed to withstand bending forces.
3. The lower rod ratio and longer stroke of the 2.3L increase the inertial loads on the rods at any RPM calling for light weight rods.

The strength of a rod in tension is mostly determined by it's cross sectional area, not the shape. All else being equal, heavier rods are stronger in tension. Resistance to bending force is more strongly influenced by shape. For equal weight, the H beam rod will handle more bending force. The V8 hot rodders have been using H beam rods for years as upgrades. (11)

The old saw still applies. Lighter rods are better until they break, then heavier rods would have been better.

STROKER'S AND PISTON SELECTION

The 2.3L stroker version of the 4G63 makes different demands on pistons from the stock 2.0L.

1. The stroker's higher piston velocity and increased side loading increases the wear on pistons.
2. The lower rod ratio makes the stroker more sensitive to piston slap.
3. The higher piston acceleration rate makes the stroker more sensitive to piston weight.
4. The greater displacement and torque of the stroker puts more stress on the piston than the stock 2.0L. (All else being equal of course.)

Piston selection requires a compromise between the four items above to match the intended purpose of the engine.

Only forged pistons are considered here. The use of forged pistons is all to the good as they are both lighter and stronger than cast pistons. Forged pistons are mostly made from two Aluminum alloys with different silicon content, 4032 and 2618. The table below shows some of the key differences in the two alloys. Most of the data is from reference (32).

Table 5 Piston Material Properties

Property	2618	4032
Percent Silicon	<2%	11%
Density	0.10 lb/in3	.097 lb/in3
Tensile Strength, Yield	54,000 psi	46,000 psi
Modulus of Elasticity	10,400 psi	11,400 psi
Fatigue Endurance Limit	18,000 psi	16,000 psi
Coefficient of Thermal Expansion 68 to 572 degrees F	13.4 uin/in/°F	11.7 uin/in/°F

Because 2618 expands more than 4032, piston to wall clearance for the 2618 piston has to be larger to have the same running clearance at operating temperature. On cold start up the 2618 piston can be expected to have more piston slap than the 4032 piston. Piston slap can be expected to be greater for the 2.3L stroker and slightly less for the 2.1L than for the stock engine.

Pistons forged from 2618 alloy have higher tensile strength and higher fatigue limits than 4032 and are better able to handle the additional stress of higher boost and occasional detonation. Because 2618 pistons are often designed for higher stress, they may be heavier than 4032 pistons.

Pistons forged from 4032 alloy will wear less than the 2618 pistons because the silicon makes the alloy harder. However being harder also means being more brittle. Failure in

a 4032 alloy piston is more likely to be catastrophic, while a 2618 piston may have a "partial" failure. (31)

Pistons forged from 4032 will wear better, probably be lighter and have less "piston slap" than 2618 pistons.   The same old saw is still true. 4032 pistons are better, until they break then 2618 would have been better.

ROD RATIO AND IGNITION TIMING

When ignition timing is just right the spark plug fires long enough Before Top Dead Center (BTDC) so that the burning fuel/air reaches maximum combustion pressure about 15 to 20 degrees After Top Dead Center (ATDC) where the piston has started down and there is enough leverage on the crankshaft to turn the pressure into torque. If maximum pressure is too early the piston and rod can't go down, however detonation can make at least the top of the piston go down. If maximum pressure is too late, it's also too low and power is lost. (35,36)

Getting the maximum cylinder pressure to arrive at just the right time is a major goal of tuning. Some of the tuning variables affecting ignition timing are: (36)
1. Higher RPM needs more degrees advance for the same time for the flame front to expand to maximum.
2. Higher boost pressure causes the flame front to move faster.
3. Richer fuel/air mixture causes the flame front to move slower.
4. Higher octane fuel causes the flame front to move slower.
5. Higher cylinder temperatures are more likely to cause auto-ignition.
6. Higher octane fuel tolerates higher temperatures without auto-ignition.
7. Peaks closer to TDC are higher and more likely to cause detonation.
8. Peaks after optimum ATDC are less likely to knock and make less power.

The longer stroke and lower rod ratio of the 2.3L stroker has higher velocity ATDC and reaches the point where pressure can be converted to torque 2.25 degrees sooner than the 2.0L engine. Theoretically the stroker should tolerate more ignition timing advance (or lower octane) better than the stock 4G63.

Figure 31 below is a plot of the Lb Ft of torque developed for each 100 pounds of piston pressure zoomed in to the area where maximum cylinder pressure is reached. The red line highlights how much earlier the stroker reaches the same leverage on the crankshaft as the stock engine. Note that the longer rod ratio of the 2.1L causes that same leverage to be reached slightly later.
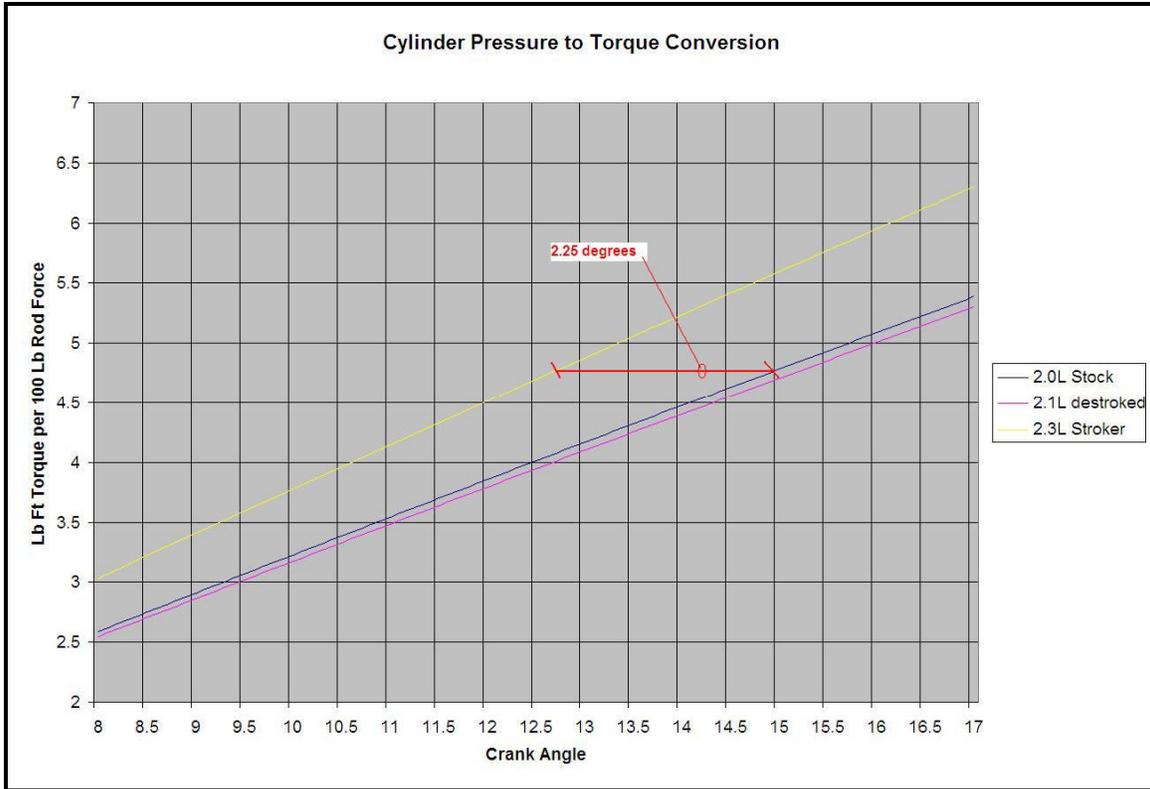
Figure 31 Torque Conversion at Max Pressure

HARMONIC BALANCE OF IN LINE FOURS

The 4G63 engine has two pairs of pistons 180 degrees apart on the crankshaft. One pair goes up while the other pair goes down. If the speeds and acceleration rates of the two pairs of pistons were the same, the engine would be perfectly balanced. However, because the rod angle changes, the velocity of the piston changes throughout the stroke. With any reasonable rod length the piston velocity is higher near TDC than at BDC. Figure 32 below shows the differences in piston acceleration rates for the most common 4G63 versions and a two liter with 10 Meter rods as an extreme example of long rods.
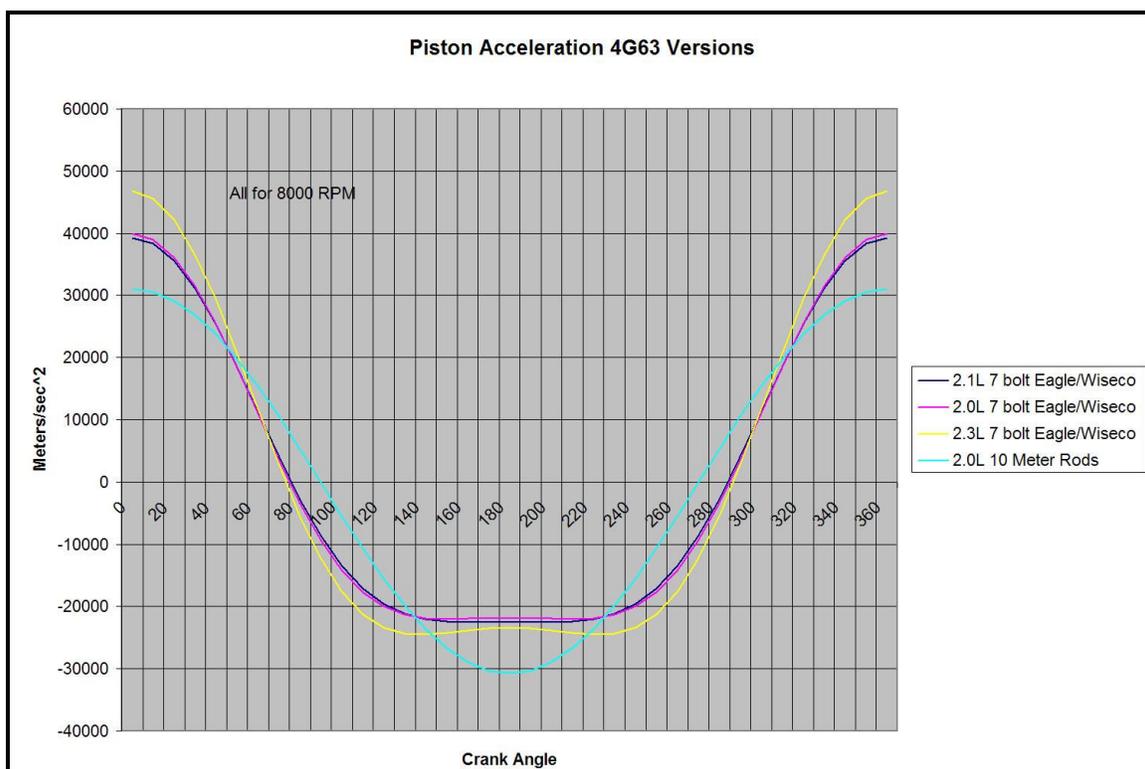
Figure 32 Piston Acceleration at 8000 RPM

As shown in the figure above the acceleration rate for the very long rods are the same at TDC and BDC, about 30,000 meters per second per second for 8000 RPM. For the real versions of the 4G63, the lower the rod ratio, the more difference in acceleration rates between TDC and BDC. The difference in acceleration rates (In Gs) times the weight of the reciprocating mass results in the imbalance forces of in line four cylinder engines. The up and down forces are equal twice a revolution as they pass through zero, causing the imbalance to be at two times the engine speed.

Figure 33 below shows harmonic imbalance in pounds force for three common 4G63 versions and the theoretical 10 meter rod engine. To get the Imbalance in pounds the acceleration rate was converted to Gs and multiplied by the reciprocating mass in pounds. The forces on two pistons 180 degrees apart are added algebraically to derive the

imbalance force in pounds for one piston pair.   Note that there's no imbalance with the 10 meter rods but that for the real engines, the lower rod ratios have significantly more imbalance.   The 2.3L stroker has about 800 pounds more imbalance force at 8000 RPM than the 2.1L destroked engine.
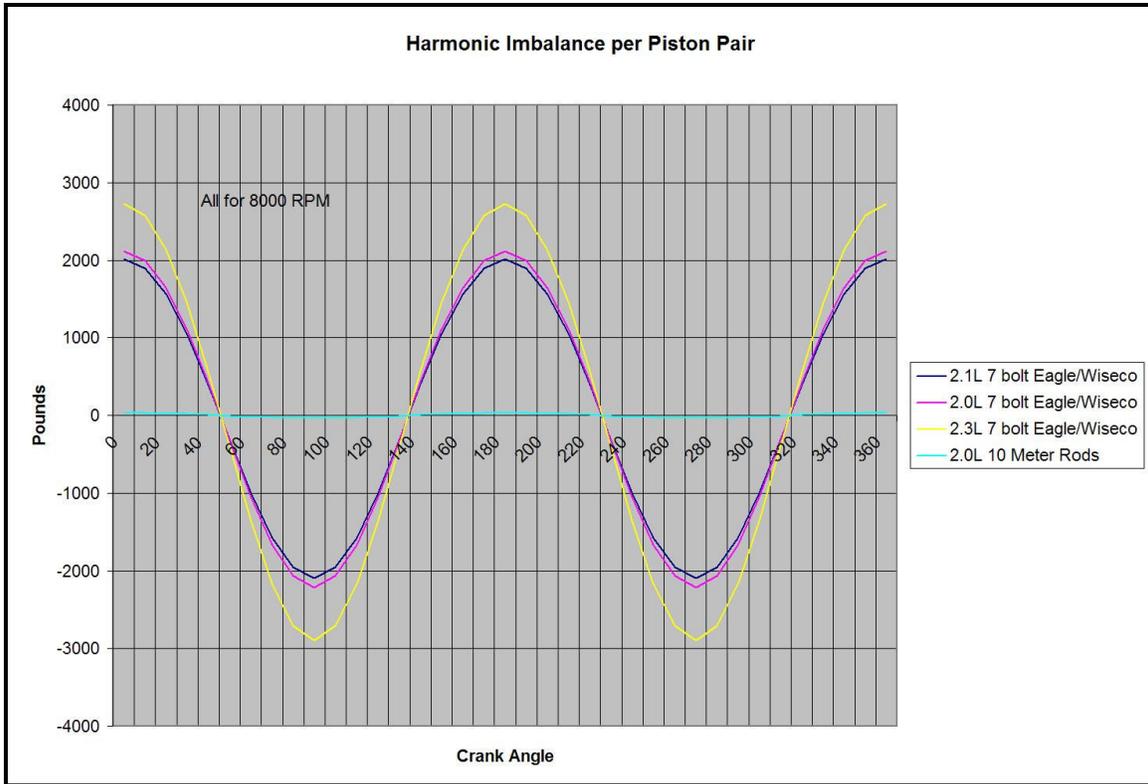
Figure 33 Harmonic Imbalance 4G63 Versions

The harmonic imbalance is inherent in the design of in line four cylinder engines and can not be compensated for by matching the weights of pistons or balancing the crankshaft. The imbalance gets worse with longer strokes or heavier pistons causing a "maximum" size of about 2 liters for in line fours.   Mitsubishi pioneered the use of balance shafts to offset the natural imbalance and licensed the technology to Porsche.(24) Figure 33 above shows the imbalance force from one pair of pistons. The 4G63's two pairs of pistons would have two times this imbalance without balance shaft compensation.

The 4G63 Mitsubishi engine includes "silent shafts". There are two eccentric shafts counter-rotating at two times the engine speed and registered so that both shafts "up" and "down" forces are additive to counter the up and down forces of the piston imbalance. Since the two shafts counter-rotate, the left to right forces of the two shafts are balanced to zero. See Figure 34 at right (1).   The two
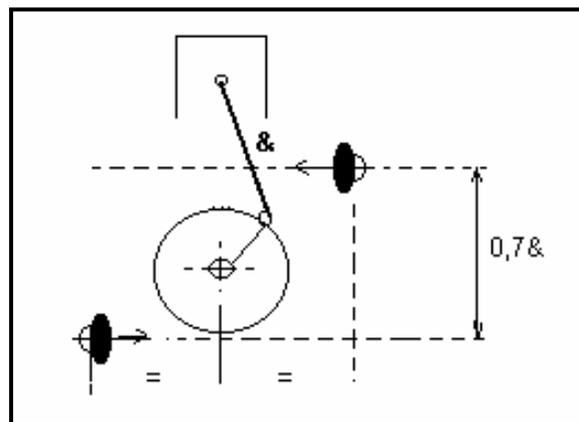
Figure 34 Mitsubishi Silent Shafts

shafts are positioned at different heights to also counter the rotating forces from combustion.

The balance shafts do not decrease the imbalance loads on the crankshaft. They apply counter forces to the engine block to decrease the block's vibration. Lowering the block vibration is all to the good. Such critical items as the oil pump and valve train can be influenced by the vibration. (3) In the DSM community its "common knowledge" that removing the balance shafts frees up horsepower. However, common sense seems to indicate that shaking a 3200 pound car should require some horsepower. But this paper is already way too long.

This would be a good place to mention that it's not a good idea to just remove the balance shaft belt, stopping one balance shaft and leave the other going. The one shaft still rotating will have nothing to counter its left to right force and will cause the engine to vibrate left and right in addition to up and down.

At the risk of speaking heresy in the DSM community, it needs to be said that the boxer four of the Subaru is inherently balanced. The boxer's two pairs of pistons are exactly 180 degrees from each other so the two pistons approaching BDC have forces that exactly cancel each other. The other pair (approaching TDC) also exactly balance each other. The boxer engine is inherently balanced without regard to rod ratio. Ok now you can flame me.

CONCLUSIONS

After all of the math and charts, go figure, the conventional wisdom still holds up. The 4G63 stroker is more suited for street use with better low end/mid range torque. The 2.0L version will still rev higher and if pushed hard, can make as much power.

Although the stroker motor may not make more power with the same big turbo as a 2.0L, it will spool the big turbo up faster, making a high horsepower daily drive more bearable.

Camshafts will not perform the same on strokers as 2.0L engines. Strokers will tolerate more aggressive cams than the 2.0L with fewer low end negative effects.

Strokers are (even) more sensitive to the weight of the reciprocating parts than 2.0l engines. Piston and rod selection should take the acceleration rates into account.

Strokers are more sensitive to balance shaft removal and hard motor mounts than the 2.0L engine. A stroker with balance shafts removed, solid motor mounts, and full weight rods and pistons should shake the mirrors.

The original high flowing head design of the 4G63 makes the stroker version still useful at stock rev limits.

2.3L Stroker Pros
1. More Displacement. Almost 18% more displacement.
2. More low/mid range torque.
3. More tolerant of aggressive cams.
4. Faster spool up.
5. Higher effective compression ratio.
6. More tolerant of timing advance and lower octane fuel.

2.3L Stroker Cons
1. Higher native harmonic imbalance.
2. More sensitive to engine balance.
3. Lower RPM potential from higher piston friction from side loading and velocity.
4. Higher tension loads on rods, both in tension and bending.
5. Volumetric Efficiency drops off at lower RPM's than 2.0L

Summary of Calculations

The 2.3L stroker is about eight sevenths of the displacement of the 2.0L engine, and the OE red line is 7000 RPM. Going into this study my early guess was that the stroker would perform at about the same at 7000 RPM as the 2.0L at 8000 RPM.   Table 6 below summarizes key differences between the three versions of the 4G63 engine. Where possible in Table 6 the RPM's of the 2.1L and 2.3L versions are listed to achieve the same performance as the 2.0L engine at 8000 RPM.

Table 6 Summary of Differences

Measurement for Comparison	Engine Version		
	2.3L	**2.0L**	**2.1L**
Bore (mm)	86	85	87
Stroke (mm)	100	88	88
Displacement (cc)	2350	1997	2092
Piston Weight (grams) for force calculations	336	336	336
Rod Weight (grams) for force calculations	550	550	594
Rod Length (mm)	150	150	162
Rod Ratio, (Rod Length/Stroke)	1.5	1.7	1.84
RPM for 282 CFM@100% VE (~400HP@ 2.2 bar)	6800	8000	7650
RPM for 39939 m/sec^2 peak piston acceleration	7390	8000	8070
RPM for 38.25 m/sec peak piston velocity	6940	8000	8040
RPM for 2155 lbs peak harmonic imbalance	7040	8000	8200
RPM for 4663 lbs tension on Rod at 0 Crank Angle	7390	8000	7957
RPM for reference side load friction on pistons	7150	8000	8450
RPM for 0.51 mach index flow thru intake valves	7000	8000	8000
Percent of bending force on rod (referenced to stock)	114%	100%	93%
Crank degrees past TDC for peak piston velocity	73	75	75.5

The rule of thumb that a stroker at 7000 RPM is pretty much the same as a 2.0L at 8000 is supported by the calculations listed in table 6. Not exactly the same but close enough for me to plan my Talon upgrade.

RECOMMENDATIONS

This section in a formal report would list recommendations for the reader based on the facts presented in the body of this document and conclusions that follow from the facts. But this paper is for the DSM tuners audience. No recommendations here. Not by me. Your Mileage May Vary.

Changing the stroke of the 4G63 engine from 88 mm to 100 mm changes the nature of the engine. Whether the nature of the stroker is 'better' is strictly a personal opinion.

DSM tuners are invited to read this document as an aid in understanding what will most closely meet your individual goals.

Or, if the equations and charts are just too much information the three-step analysis used by the author might be simpler.

1. Hmmm torque good.
2. Me stroke Talon.
3. Make tires happy.

Happy tuning.

REFERENCES

(1) http://www.fordscorpio.co.uk/tech2_3_2.htm
   Good graphic of the operation of balance shafts including Mitsubishi "silent shafts".

(2) http://www.aussiemachine.com/balance.htm
   An Australian engine balancing firm's web site discusses the effect of engine imbalance.

(3) http://www.centuryperformance.com/balancing.asp
   More about the effects of engine imbalance.

(4) http://www.csgnetwork.com/automotiveconverters.html
   A site with many useful automotive converters and calculators.

(5) http://victorylibrary.com/mopar/cam-tech-c.htm
   Cam Timing vs. Compression Analysis. Has a good general description of engine dynamics.

(6) http://victorylibrary.com/mopar/rod-tech-c.htm
   Connecting Rod vs. Stroke Analysis. Explains the pros and cons of different rod ratios.

(7) http://victorylibrary.com/mopar/crank-bal-c.htm
   Crankshaft balance factors. Explains the effects of balance and reciprocating vs. rotating mass.

(8) http://www.eng-tips.com/viewthread.cfm?qid=85349
   A forum about engine dynamics with equations and examples of forces in various competition engines.

(9) http://www.engineersedge.com/engine_formula_automotive.htm
   A collection of engine related formula.

(10) http://www.arp-bolts.com/Catalog/Catalog.Images/ARPCatalog.pdf
   ARP's catalog including technical data about engine dynamics.

(11) http://www.grapeaperacing.com/GrapeApeRacing/tech/connectingrods.pdf
   Technical data about various connecting rods.

(12) http://www.menet.umn.edu/~groep004/Senior_Project.pdf
   A document describing the design process for a single cylinder engine. Many of the concepts described are relative to engines in general.

(13) http://www.vka.rwth-
aachen.de/StudLehr/Lehre/Formelsammlungen/FormelsammlungICE1uICEF.pdf
  A collection of engine related formula.

(14) http://www.mid-lift.com/TECH/TECH-Definitions.htm#PISTON%20VELOCITY
  A Glossary of engine related terms.

(15) http://www.pro-race.com/faq.htm
  A description of the operation of and need for harmonic balancers.

(16) http://e30m3performance.com/tech_articles/engine-tech/rod-ratio/kin2.htm
  Rod Ratio Kinematics. Excellent description of the effect of rod ratio on engine
  dynamics. Includes a downloadable Excel spread sheet to generate charts
  describing piston dynamics.

(17) http://www.miata.net/garage/KnowYourCar/S11_Piston.html
  Mazda MX-5 Piston Acceleration And Piston Velocity. Written specifically for
  the Miata but includes general formula in Excel friendly format for several
  aspects of piston dynamics. This page is the source for several formula used in
  this white paper.

(18) http://www.iantaylor.org.uk/papers/Esslingen1998.pdf
  A paper on the effects of lubricants on engine friction.

(19) http://www.ongcreports.com/HR%20MANUAL/mech_eng_handbook.pdf
  The Mechanical Engineers Handbook. (Read it!)

(20)Http://www.paddocktalk.com/news/html/modules.php?op=modload&name=News&f
ile=article&sid=28546
  An article about formula one engines with data on piston dynamics.

(21) http://www.bmwworld.com/engines/p83.htm
  Information about BMW's Formula One engine with data on piston dynamics.

(22) http://www.nd.edu/~caschenb/Main2.pdf
  Study of the 4 Stroke Gasoline Internal Combustion Engine. History and basic
  operating theory of engines.   Good basic information about engine theory.

(23) http://www.hotrodders.com/forum/383-vs-350-a-48278-
4.html?s=a08013f08a96d45df041285cafb525f8
  A hotrodders (V8 guys) discussion of stroking small block V8s.

(24) http://www.answers.com/topic/balance-shaft
  Balance shaft information and history from answers.com

(25) http://www.empirenet.com/pkelley2/DynamicCR.html
Defines the difference between static and dynamic compression ratio. Includes link to Visual Basic program to calculate different combinations.

(26) http://www.rbracing-rsr.com/comprAdvBMW.htm
Interactive dynamic compression calculator accounting for cam events, rod angles boost pressure, and altitude.

(27) http://not2fast.wryday.com/turbo/compression/compression.shtml
Page has a more complex calculator of dynamic compression including deck milling and piston shape.

(28) http://www.symuli.com/vw/camp1.html
This page has a straightforward description of cam timing and terminology.

(29) http://www.4g63t.net/2005_FSM/GR00001500-11B.pdf
4G63 Overhaul manual. Lots of hard data on how the 4G63 is put together.

(30) http://en.wikipedia.org/wiki/Compression_ratio
Wikipedia article on compression ratio includes the effect of heat of compression.

(31) http://www.stealth316.com/2-pistonguide.htm
Piston upgrade guide for DSM's cousin the stealth.

(32) http://www.jepistons.com/pdf/2006-sportcomp1.pdf
Technical data from JE about piston alloys.

(33) http://www.wallaceracing.com/machcalc.php
Intake port mach index calculator. Assumes one valve per cylinder so enter RPM/2 for the 4G63

(34)http://www.jcmmachine.com/PDF%20files/JCM%20Tech%20Report%20ch%204%20to%209.pdf
Everything you ever wanted to know about gasoline and the combustion process from the performance engine builders viewpoint.

(35) http://www.jcmmachine.com/PDF%20files/JCM%20Tech%20Report%20ch3.pdf
Normal and abnormal combustion. What really goes wrong to cause those holes in your piston?

(36) http://www.innovatemotorsports.com/resources/myths.php
Ignition timing myths. Good description of mixture, octane, pressure, and ignition timing on power and detonation.

APPENDIX A – FORMULA USED IN CHARTS

The Excel Worksheet used in this white paper is available from http://kidzuku.com\4G63PistonForces.zip. Please download and unzip for local use. The spreadsheet was not (NOT I say) prepared for commercial use. Some familiarity with the idiosyncrasies of Excel will be required to get results from the spreadsheet.

Basics for all calculations

Reference (17)		2.0L	2.1L	2.3L
Stroke mm	S	88	88	100
Bore mm	B	85	87	86
Connecting Rod Length (+/- 0.05mm)	R	150	162	150
Number of Cylinders	N	4	4	4
Rod Stroke ratio	R/S	1.70	1.84	1.5
Engine Displacement (cc)	D*N	1997	2092	2323
Crank Arm mm	S/2			
RPM	T	Revolutions per minute of the Crank		
Picton Velocity	V	This is the speed of the Piston's movement measured in meters per second		
Piston Acceleration	A	The rate of change in velocity, measured in meters per second per second		
Crank Angle	K	Crank angle in degrees after TDC		

To calculate the velocity (V) of the piston when the crank is at a particular degree of rotation (K) ATDC and at a given RPM (T) the formula is: (17)

Piston Velocity at a given crank angle
V=(T*ATAN(1)/7.5)*((S/2)/1000)*SIN(((ATAN(1)/7.5)*K/6))*(1+COS(((ATAN(1)/7.5)*K/6)) /(SQRT(((R/(S/2))^2)-SIN(((ATAN(1)/7.5)*K/6))*SIN(((ATAN(1)/7.5)*K/6)))))

The formula for calculating the acceleration rate (A) of a piston at a given crank angle after TDC (K) is (17)

Acceleration in Meters per second per second at K
=((T*ATAN(1)/7.5)^2)*((S/2)/1000)* ((1-
COS(4*((ATAN(1)/7.5)*K/6)))/(8*(SQRT(((R/(S/2))^2)-
SIN(((ATAN(1)/7.5)*K/6))*SIN(((ATAN(1)/7.5)*K/6))))^
3)+COS(2*((ATAN(1)/7.5)*K/6))/(SQRT(((R/(S/2))^2)-
SIN(((ATAN(1)/7.5)*K/6))*SIN(((ATAN(1)/7.5)*K/6))))+COS(((ATAN(1)/7.5)*K/6)))

The formula for the location (W) and displacement (D) of the piston for a given crank angle (K) after TDC is: (17)

Location = W = (((S/2)+R)-((S/2)*COS(RADIANS(K)))-SQRT(R^2-((S/2)*SIN(RADIANS(K)))^2))

Displacement at K = D = (+PI()*(B^2)*W/1000)/4

To calculate the imbalance force in pounds and effective compression required a few more formula to be developed and tested.

Gs or G forces	G	Non-dimensional, 1 G = normal earth Gravity
Force	F	Force in pounds
Harmonic Imbalance	HI	Force due to harmonic imbalance in pounds
Static Imbalance	SI	Force due to mismatched weights in pounds
Total Imbalance	TI	Static Imbalance plus Harmonic Imbalance
Reciprocating Mass	RM	Recip. mass in pounds (grams *0.0022046)
Swept Volume	SV	Displacement of one cylinder (cc)
Head Volume	HV	Volume of combustion chamber (cc)
Static Compression Ratio	CR	Using full swept volume (SV + HV)/HV
Compressed Volume	CV	Displacement at point of intake valve closing
Uncompressed Volume	UV	Swept Volume – Compressed Volume
Effective CR	ECR	Using compressed volume (CV + HV)/HV
Boost Pressure	BP	Cylinder pressure as intake valve closes
Intake valve closing angle	KIC	Crank angle where the intake valve closes
Cylinder Pressure at K	CPK	Pressure at K in same units as Boost Pressure

G forces:
$G = A/9.80665$

Reciprocating Mass (10):
$RM = PistonWeight + RodWeight/3$

Force:
$F = G * RM$

Harmonic Imbalance:
$HI = F_{K1} + F_{K2}$ Where the crank angle at F_{K1} is 180 degrees from the force calculation at F_{K2}.

Total Imbalance:
$TI = HI_{RM1} + HI_{RM2}$ where the Harmonic Imbalance calculations for HI_{RM1} and HI_{RM2} are the same except for the reciprocating mass, and the crank angles are 180 degrees apart.

Static Imbalance:
$SI = TI - HI$

Swept Volume:
$SV = (PI()*((B/2)^2)*S/1000)$

Head Volume:
$HV = SV/CR$

Static Compression Ratio:
CR = Entered value of static compression ration accounting for bore, stroke, and piston shape. The advertised CR is assumed to include a nominal .040 thick head gasket.

Compressed Volume:
CV = Displacement at K of KIC. (Compression starts at intake valve closing.)

Uncompressed Volume:
$UV = SV - CV$

Effective CR:
$ECR = CV/HV$

Boost Pressure:
BP = entered boost pressure PSI absolute where 14.7 = normally aspirated and 29.4 equals two bar. Cylinder pressure is corrected for PSI gauge, where two bar reads 14.7 PSIG.

Intake Valve Closing Angle:
KIC = From cam specification referenced to a K of 0 degrees = TDC
A specification of 28 degrees ABDC is a KIC of 208. (ABDC + 180)

Cylinder Pressure at K:
$CPK = (((CV + HV)/ (MIN(CV,D_K) + HV))^{1.23} * BP) - 14.7$
Where:

      D_K = Displacement at crank angle K
      The exponent of 1.23 accounts for heat of compression.
      Minus 14.7 corrects the absolute units back to gauge pressure.

To calculate rod angles and effective lever length to calculate the effect of rod angle on torque a few more formulas had to be developed.

Rod Angle	RA	Calcs in radians outputs in degrees
Crank Angle	CA	Calcs in radians, outputs in degrees
Rod Length	RL	In mm
Stroke/2	SH	Half of the stroke in mm
Rod Journal Offset	RJO	Rod Journal offset from Centerline mm
Effective Lever Length	ELL	Lever on Crank adj for CA and RA, feet

Rod Angle Formula
Sin CA = RJO/SH (Sin = Opp/Hyp)
RJO = SH * Sin CA

Sin RA = RJO/R (Sin = Opp/Hyp)
RA = ArcSin RJO/RL
RA = ArcSin((SH * Sin (CA))/RL)
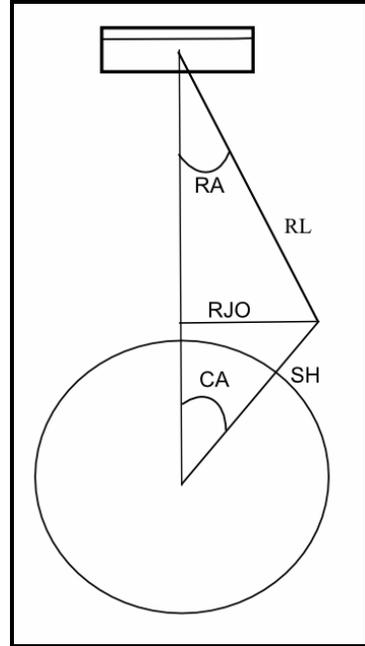
Figure 35 Rod Angle Sketch

Effective Lever Length Formula
ELL = Cos(90 - RA + CA) * SH
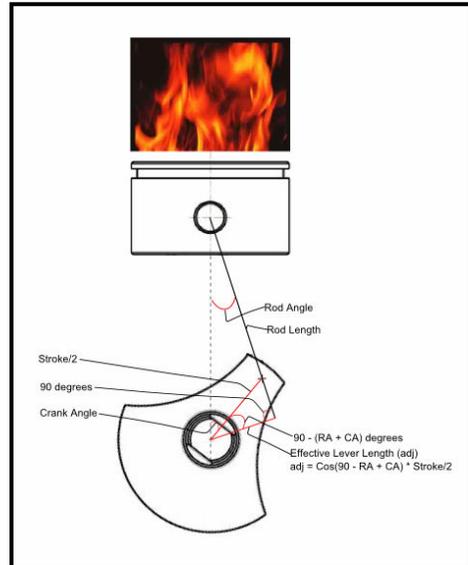
Figure 36 Effective Lever Length Sketch

APPENDIX A – SPECS USED IN THE CALCULATIONS

PUT THE ENGINE SIZES AND PISTON ROD SPECS HERE.
TALK ABOUT NO HARD DATA FOR 2.1L RODS.

APPENDIX C - DEFINITIONS

CAM OVERLAP (14): Refers to the time when both, the intake and exhaust valves of a given cylinder are open at the same time, measured in CAM DEGREES. Specifically, it is when the INTAKE valve begins to open, as the EXHAUST valve is just closing. The piston is still rising UP, chasing the exhaust valve closed, and the intake valve is opening to meet it. Increased overlap is needed on higher RPM's to provide additional time for the cylinder to "purge" itself of leftover gases from the prior combustion process. Physically, as explained in Lobe separation, the overlap cycle is the opposite perspective of the same consequence to cam design. They effect the engine's "running" compression ratio. You are merely saying the same thing from opposite perspectives. OVERLAP is a term not used much anymore, but was very much the perspective of cam development and comparison by professional engine builders during the pivotal cam development era of the late 1960's and 70's, where many of the breakthroughs of cam design were established. The transition to "lobe separation" lingo was the result of marketing by cam companies, beginning in the mid to late 1970's, who needed to boost cam sales and horsepower on LOW compression engines, resultant from the change in Detroit's gas guzzling car production, and ending the Muscle Car era as we knew it. Reduced CAM OVERLAP is INCREASED LOBE SEPARATION. This reduced cam overlap increased net effective cylinder pressure (NECP) within the engine, often boosting factory engine horsepower by in effect increasing the running compression on these milder, low compression engines. The greater OVERLAP was not needed on these low compression engines since they were not going to operate at high RPM's. Among advanced cam analysis in professional racing engines, cam overlap is still a critical perspective and term to be documented and used, particularly with cylinder head flow analysis.

NOTE: *It is important to understand that you need to increase the static compression ratio of pistons (i.e., 9:1 up to 12:1) because a cam of greater overlap will lose "running compression" (NECP). Increasing the piston's compression ratio replaces this lost operating pressure from the increased cam overlap. (Assuming you have cylinder heads capable of breathing at these higher valve lifts and RPM's.) You cannot increase cam overlap and duration on a low compression engine without changing the pistons for higher static compression, and still achieve full potential. Nor can you install larger compression pistons without changing to a cam of higher overlap (and duration), since now you will have too much "running compression" (NECP) and likely burn pistons, valves and create other problems from excessive heat, pre-ignition and detonation. The two run hand in hand. The only exception to these necessities being chosen only for higher RPM operation would be a large cubic inch engine with a large stroke crankshaft; as this increased crank stroke increases piston velocity and increases airflow through the heads at lower RPM's.*

Compression ratio(30)

The compression ratio is a single number that can be used to predict the performance of any engine (such as an internal-combustion engine or a Stirling Engine). It is a ratio between the volume of a combustion chamber and cylinder, when the piston is at the bottom of its stroke and the volume when the piston is at the top of its stroke. The higher the compression ratio, the more mechanical energy an engine can squeeze from its air-fuel mixture. Literally, high ratios place increased oxygen and fuel molecules into a reduced space, thus allowing for increased power at the moment of ignition. Higher compression ratios, however, also make detonation more likely.

The ratio is calculated by the following formula:

$$\mathrm{CR} = \frac{\frac{\pi}{4}b^2 s + V_c}{V_c}$$

, where

b = cylinder bore (diameter)

s = piston stroke length

Vc = volume of the combustion chamber (including head gasket). This is the minimum volume of the space into which the fuel and air is compressed, prior to ignition. Because of the complex shape of this space, it usually is measured directly rather than calculated.

Due to pinging (detonation), the CR in a gasoline/petrol powered engine will usually not be much higher than 10:1, although some production automotive engines built for high-performance from 1955-1972 had compression ratios as high as 12.5:1, which could run safely on the high-octane leaded gasoline then available. Recently, with the addition of variable valve timing and knock sensors to delay ignition timing, one worldwide manufacturer is building 10.8 CR gasoline engines that use (U.S.) 87 octane fuel.

In engines running exclusively on LPG or CNG, the CR may be higher, due to the higher octane rating of these fuels.

IC racing engines burning methanol and ethanol often exceed a CR of 15:1.

In engines with a 'ping' or 'knock' sensor and an electronic control unit, the CR can be as high as 13:1 (2005 BMW K1200S)

In a turbocharged or supercharged engine, the CR is customarily built at 8.5:1 or lower.

In an auto-ignition diesel engine, the CR will customarily exceed 14:1--and over 22:1 is not uncommon.

Fault finding and diagnosis

Measuring the compression pressure of an engine, with a pressure gauge connected to the spark plug opening, gives an indication of the engine's state and quality.

If the nominal compression ratio of an engine is given, e.g. as 10:1, the measured pressure in each cylinder of common automotive designs can be roughly estimated in pounds per square inch as between 15 and 20 times the compression ratio, or in this case between 150 psi and 200 psi, depending on cam timing. Purpose-built racing engines, stationary engines etc. will return figures outside this range.

If there is a significant (> 10%) difference between cylinders, that may be an indication that valves or cylinder head gaskets are leaking, piston rings are worn or that the block is cracked.

If a problem is suspected then a more comprehensive test using a leak-down tester can locate the leak.

Because cylinder bore diameter, piston stroke length and combustion chamber volume are almost always constant, the compression ratio for a given engine is almost always constant, until engine wear takes its toll.

Dynamic Compression Ratio

The calculated compression ratio, as given above, presumes that the cylinder is sealed at the bottom of the stroke (BDC or bottom dead center), and that the volume compressed is the actual volume.

This is not true: intake valve closure (sealing the cylinder) always takes place after BDC, which causes some of the intake charge to be compressed backwards out of the cylinder by the rising piston at very low speeds; only the percentage of the stroke after intake valve closure is compressed. This "corrected" compression ratio is commonly called the "dynamic compression ratio".

This ratio is higher with more conservative (i.e., earlier, soon after BDC) intake cam timing, and lower with more radical (i.e., later, long after BDC) intake cam timing, but always lower than the static or "nominal" compression ratio. The actual position of the piston can be determined by trigonometry, using the stroke

length and the connecting rod length (measured between centers). The absolute cylinder pressure is the result of an exponent of the dynamic compression ratio. This exponent is a polytropic value for the ratio of variable heats for air and similar gases at the temperatures present. This compensates for the temperature rise caused by compression, as well as heat lost to the cylinder. Under ideal (adiabatic) conditions, the exponent would be 1.4, but a lower value, generally between 1.2 and 1.3 is used, since the amount of heat lost will vary among engines based on design, size and materials used, but provides useful results for purposes of comparison. For example, if the static compression ratio is 10:1, and the dynamic compression ratio is 7.5:1, a useful value for cylinder pressure would be $(7.5)^{1.3}$ × atmospheric pressure, or 13.727 × 14.7 psi at sea level, or 201.8 psi. The pressure shown on a gauge would be the absolute pressure less atmospheric pressure, or 187.1 psi. From this, we can see that the two corrections for dynamic compression ratio affect cylinder pressure in opposite directions, but not in equal strength. An engine with high static compression ratio and late intake valve closure will have a DCR similar to an engine with lower compression but earlier intake valve closure.

HORSEPOWER: (14) In its *purest* definition is: 1 HP= 33,000 Foot-Pounds of WORK Per Minute. In measurement, it is TORQUE multiplied by RPM divided by 5252. The essence of how horsepower is derived is based upon an understanding of the difference between WORK and TORQUE. FORCE is the common denominator of the two. WORK is DIRECT FORCE operating across a distance; while TORQUE is RADIAL FORCE operating around an axis. The formula of these is: POWER = FORCE x DISTANCE ÷ TIME. To calculate power, both DISTANCE and TIME must be derived, using FORCE across both of these. Where engines are involved, the rotational values of the crankshaft by its diameter is needed. To generate a formula for the TORQUE, the DISTANCE aspect is measured in RPM (revolutions per minute), times the RADIUS, times Pi (3.1416), times 2 (or 6.2832.). TORQUE is measured as Pounds of Force Times Distance. POWER is measured as FORCE times Distance per Minute. The 5252 value in our equation comes from 33,000 divided by 2x Pi (6.2832). Therefore: HP = Torque x RPM ÷ 5252. Something worth noting, is that at 5252 RPM, both TORQUE and HORSEPOWER will be equal. Beneath this value, Torque will be numerically greater, while above it Horsepower will be greater.

PISTON VELOCITY (14): This is a relatively self descriptive term, but its real influence with engine performance is often not fully understood. Piston velocity, or "speed," is the result of two things: (1) Crankshaft stroke, and (2) RPM's (obviously). Piston speed is THE SOURCE for response characteristics of cylinder head air flow through the ports, in addition to the cubic inch *volume* each cylinder has with every stroke. These are two crucial elements to place in proper context: (a) Airflow VOLUME, and (b) airflow VELOCITY. Cylinder volume in conjunction with piston speed determines all of the tuning characteristics and specification needs within a given cylinder head's design, as well as the fundamental camshaft specs to meet these needs. These two elements can offset each other when choosing cylinder head packages and/or camshafts. In other words, if you have a large cubic inch small block engine, its increased stroke will achieve high torque at lower RPM's compared to a conventional size small block engine, but it can use larger port heads and greater camshaft specs that would more typically be chosen for a high winding engine of smaller cubic inches.

NOTE: Torque is often the term that is used to explain what a large stroke engine sees. Although it is true that the operating pressures created with each power stroke is applying leverage onto a longer "arm" for an engine having a greater stroke, and thus "torque" forces upon the crank are increased, the simple truth with what is happening in the cylinder heads with airflow is that torque is a symptom from the increased PISTON VELOCITY that the stroke generated. The engine literally thinks it is going 8,000 RPM when the tach reads 6,800 RPM, because airflow and volume demand are greater at lower RPM's. There is a second dynamic to the piston's influence on the cylinder heads, dictated by a ratio of connecting rod length divided by crankshaft stroke. Known as the rod ratio this determines how quickly the STARTING and STOPPING of PISTON SPEED occurs. But the ultimate piston speed is still determined by our two main factors (crank stroke and RPM's). The formula for determining piston velocity in FPM (feet per minute) is this: STROKE x 3.1416 x RPM ÷ 12 = FPM. If you use this formula when comparing different engine strokes, you can see how the RPM's are affected. TO COMPARE, pick your first stroke combination and use the foregoing formula. Then save the FPM value in your calculator and begin working backwards from this sum. Only now substitute the division symbol for a multiplication symbol, and divide where there is a multiplication symbol. SKIP over the "RPM" in this second step working backwards while changing the STROKE value to your second choice (i.e., 4.50" first calculation compared to 4.00" for second comparison), and your final answer will be the different RPM "effect" created in the ports by the different stroke. The answer to a comparison of a 4.50" stroke running at 6,000 RPM, compared to a 4.00" stroke is 6,750 RPM. In other words two identical engines of the SAME SIZE cubic inches having their only difference being stroke, will have a 750 RPM increase being needed for the 4.00" stroke to have the same port velocity as the 4.50" stroke. Or more simply stated: the 4.50" stroke engine only needs to turn 6,000 RPM to make the same power that the 4.00" stroke engine can make at 6,750 RPM. When you consider that the stroked engine will also have more cubic inches (unless the cylinder bores were much smaller to keep the comparison the same size), the extra VOLUME dictates that PORT passages would also need to be increased, as well as VALVE LIFT. This is why it is easy to exceed the cylinder head capabilities when going into big stroke engines; not to mention all the increased work needed by larger induction and exhaust. When these limits are reached with such engines, the RPM's stop dead in their tracks, although you've created a "torque monster" at lower RPM's; and to think, it all starts with "piston velocity."

ROD RATIO (14) : Refers to the "connecting rods" between the piston and the crankshaft; but specifically, the mathematical division of a crankshaft's STROKE by the connecting rods LENGTH. Ranging anywhere from 1.3:1 (not good) to more than 2.0:1, the effects of rod ratio on an engine's performance and specifically its torque curve are significant. Rod ratio directly affects how the piston speed accelerates from its extreme positions of TDC and BDC (as well as slows down in approaching these points). The piston velocity still reaches its same peak value, as dictated by the crankshaft's stroke; but this acceleration toward this is determined by rod ratio, with the smaller values of rod ratio making these quicker, and the higher values slower, respectively. One distinct down side of lower rod ratios, is their inherently higher thrust of the piston's skirt into the cylinder wall. These require more piston skirt clearance compared to a similar stroke engine using a longer connecting rod. Of course, point of fact: if you have two engines of the same model with the same stroke, but different length connecting rods, then of course the difference must be made up with either a different Compression height piston, and/or a greater deck height block. It is commonly accepted that connecting rod ratios of 1.70:1 to 1.90:1 are preferred for most applications. Deviations above and below these values usually have some other negative offset. The higher the rod ratio, the higher the shift in RPM's for peak torque to occur.