"Special 300 HP" Engine Considerations
For 1962 to 1967 327 CID, 300 HP Corvette Engines

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Overview:
Many owners of 327/300 HP Corvettes desire "more power", but do not want to alter external appearance or the smooth, low speed idle characteristics of this engine. A Flint-built 327/300 produces excellent low end torque, but only about 225 SAE net horsepower at the crankshaft, which is about 190 horsepower at the rear wheels with SAE air density correction. Power peaks in the 4000-4500 range and then drops off with maximum useable revs of approximately 5000.

Pocket porting/port matching/chamber relieving the cylinder heads with careful multi-angle valve seat work and retarding the OE camshaft four degrees will increase peak power to about 265 net HP with only a modest loss of low end torque and extend the useable rev range to 5500+, however, some owners desire more.

The author has completed several Corvette engine system engineering projects that produced excellent results and considerable test data, which led to increased insight into vintage Corvette engine performance. As machined by GM, 461 and later cylinder heads produce isentropic flow efficiencies of about 44 percent on the inlet side and 41 percent on the exhaust side, and the ratio of exhaust to inlet flow is about 0.65. Because the exhaust port is shorter and of simpler geometry than the inlet port, traditional cylinder head modifications yield about a 20-30 percent flow increase on the exhaust side, but only a 10-12 percent increase on the inlet side. The resulting inlet/exhaust (E/I) isentropic flow efficiencies are in the range of 50/60 percent with an E/I flow ratio of approximately 0.80.

Early 327 camshafts trace their design back to the small port, small valve heads of the 1957 era, and their flow was "balanced" with an E/I flow ratio close to 0.75, which responds well to equal duration on both sides. This is reflected in the early base engine hydraulic lifter and Duntov mechanical lifter camshafts, each of which uses the same effective lobe on both sides. The Duntov cam exhaust lobe has a .004" taller clearance ramp than the inlet lobe, but above the tops of the clearance ramps, the inlet and exhaust lobe eccentricities are identical.

Introduction of the large port 461X heads in 1961 yielded a considerable increase in inlet flow that "unbalanced" the E/I ratio (to about 0.65) resulting in a restrictive exhaust port relative to the inlet port, and this situation responds well to an earlier opening exhaust valve, which is represented by longer exhaust than inlet duration with a relatively early phased exhaust point of maximum lift (POML).

The new E/I flow ratio was not taken into account with the next two camshaft designs - the "30-30" and L-79 camshafts, each of which is a "single pattern" camshaft that has the same lobe on both sides, but was incorporated into the new for 1967 base engine camshaft - 3896929. Relative to the previous 3733431 camshaft, the "929" has 6 degrees more exhaust than inlet duration at .050" tappet lift, and the exhaust point of maximum lift is advanced 3.5 degrees. This has the effect of opening the exhaust valve 6.5 degrees earlier while maintaining essentially the same exhaust closing point as the previous camshaft.

A good sports car road engine should have the broadest possible torque bandwidth while producing more top end power and useable revs than a comparable sedan engine, and to achieve this goal, valve timing must be carefully matched to inlet and exhaust flow characteristics; and because traditional cylinder head modification techniques significantly alter the E/I flow ratio, a different valve timing scheme is called for. Since head work significantly increases the E/I flow ratio, optimum valve timing requires less exhaust than inlet duration.

With the above knowledge the author designed a new camshaft that is specifically tuned to the 300 HP engine with "massaged" heads - one that will retain the smooth 500 RPM idle in neutral at approximately 18-19" Hg. manifold vacuum while increasing peak power and revs to Special High Performance (SHP) engine levels with equivalent cylinder head preparation. There will be some loss of low-end torque, but not as much as the L-79 configuration.

It cannot be overemphasized that this engine configuration is very carefully system engineered to achieve desired results, and deviation from recommended specifications and parts may result in poor performance. The remainder of this paper discusses specific considerations, specifications, and parts that are required to attain design objectives.

Compression Ratio:

Since the Special 300 HP camshaft has a “late phased” inlet event with a very late closing inlet valve, it requires a relatively high static compression ratio - in the range of 10.0-10.5:1, but because the inlet valve closes very late, dynamic compression ratio is well below 8:1. One can never rely on "specified" compression ratios. Variation in cylinder block deck height and head gasket choice can result in a wide range of actual "as-built" compression ratio. For this reason, it is very important to measure deck clearance and know head gasket compressed thickness in addition to net piston and cylinder head chamber volumes in order to accurately compute and manage the as-built compression ratio within the specified range.

The nominal Chevrolet blueprint deck height - crankshaft axis centerline to block deck - is 9.025", and the sum of nominal crankshaft throw radius (1.625"), connecting rod length (5.700"), and OE piston compression height (1.675") is 9.000", so if all components are machined to nominal blueprint dimensions, deck clearance - the distance the machined piston crown is below the block deck at TDC - should be .025", however this is often not the case.

As machined by Flint, OE block decks are often high, and the typical range is up to .015" above the nominal 9.025" dimension. Also, it is common for the right and left side deck heights to vary by up to .010". Since .010" variation in deck height changes static compression ratio by about 0.25, it is very important to have accurate dimensions for block deck height and deck clearance.

Prior to block disassembly, the deck clearance of all eight cylinders should be measured and the results recorded. This data will tell you if the block deck is parallel to the crankshaft axis by observing variation down each bank of cylinders, reveal any difference between the two banks, and provide a starting point to compute the new compression ratio using the measured deck clearance, new piston volume and compression height, head chamber volume, and gaskets of different thickness as required to achieve the target range. Better machine shops have tooling that facilitates actual measurement of block deck height after the block is disassembled, and I recommend this procedure in addition to measuring deck clearances prior to block disassembly. The crank throw radius, rod length, and piston compression height are limited to tight tolerance ranges - plus or minus one thousandth or less - so most deck clearance variation from the nominal .025" is in the block deck height dimension.

Prior to cylinder head installation, a final check of deck clearance must be accomplished and a suitable compressed thickness head gasket chosen to achieve the target compression ratio range.

Of course, blocks with original Flint broached decks should not be "decked" in the field in order to retain the original die stamped ID data and tooling marks, and head gaskets are available in sufficient range to minimize side to side variation in final compression ratio, however, if the left deck is higher than the right deck, it may be machined to equal the right deck without removing any critical stamped identification characters or original tooling marks that are visible when the engine is assembled.

Head gasket compressed thickness availability ranges from .015" to over .050". The thinnest gaskets are of the OE shim type, and most modern shim-type gaskets have an organic coating, which aids sealing. Block deck and head surfaces should be checked with a machinist's bar and .0015" feeler gage, and if flat, a shim-type gasket should seal. Composition gaskets will tolerate some warp - typically up to .003". Gaskets may also be doubled as they were on SHP/FI engines beginning as a running change during the 1962 model year. As a result, nearly any total gasket compressed thickness can be achieved in approximate .005" increments. The following is a partial list of currently available head gaskets in order of specified compressed thickness. Additional offerings are on the referenced manufacturers' Web sites.

Brand	Part number	Type	Bore dia.	Comp. thick- ness spec.	Notes
Felpro	1094	Shim, coated	4.100"	.015"	polymer coating
Victor Reinz	1178SSB	Shim, st. steel	4.100"	.020"	stainless steel
ROL	31600	Shim, coated	4.090"	.024"	polymer coating
GM	3830711	Shim, painted	4.080"	.026"	possibly discontinued
Victor Reinz	5746	Composition	4.100"	.026"	
Felpro	1043	Composition	4.080"	.039"	
ROL	HG31000HT	Composition	4.125"	.045"	

Felpro: http://www.21cgt.com/FMWebCatalog/default.htm
ROL: http://www.rolmfg.com
Victor Reinz: http://www.napaonline.com/MasterPages/NOLMaster.aspx?PageId=0

Do not be overly concerned with "quench clearance" - the distance between the piston crown and head surface, which is the sum of deck clearance and compressed head gasket thickness. According to Taylor ("The Internal Combustion Engine in Theory and Practice", two volumes, MIT Press), the benefits of additional detonation resistance are lost once quench clearance exceeds .005 times bore diameter, which is .020" for a 4" bore, and this is well below Chevrolet's minimum recommended .035-.040". Quench clearance up to about .060" is perfectly acceptable for the specified static compression ratio range, and .050-060” quench clearance is typical for Flint-built engines. Minimum quench clearance is only important for racing engines where compression ratio is pushed to the absolute limit of available fuel octane.

The following Web-based compression ratio calculator is a very useful tool:

http://www.csgnetwork.com/compcalc.html

Pistons and Rings:
In order to preserve the quiet operation of the 300 HP engine, a snug fit, cast piston is required, and the three following cast hypereutectic pistons are acceptable candidates. The KB157 should be evaluated first in the compression ratio calculator once deck height is determined. The alternatives will yield lower compression at a given deck clearance and are candidates if the block has been decked. The Federal Mogul piston is sold under both the Sealed Power and Speed Pro brands. The OE cast pistons have .060" pin offset, but these replacements have no pin offset, which eliminates the need for "double" valve clearance notches. Note that the KB pistons have .003" greater than OE 1.675" compression height.

Brand or Mfg. Basic part no. Comp. height Net volume Available oversizes, thousandths

OE 3799491 1.675” -4.5 cc Std. size, .060” pin offset
Keith Black KB157 1.678" 0.5 cc 20, 30, 40, 60
Speed Pro H660CP 1.675" -5 cc 20, 30, 40, 60
Keith Black KB156 1.678" -7 cc 20, 30, 40, 60

Keith Black pistons have some special design features. The high placed top ring runs hotter than the lower placed top ring of the OE or Federal Mogul pistons and requires greater than OE end gap. It is absolutely necessary to follow the specific ring gap guidelines to avoid breaking a piston due to ring butting. Detailed instructions and specifications are on the Keith Black web site:

Keith Black pistons: http://kb-silvolite.com/index2.php

Additional information on the Federal Mogul manufactured pistons can be found in the Federal Mogul online catalog:

Speed Pro and Sealed Power pistons: http://www.21cgt.com/FMWebCatalog/default.htm

All the above pistons use a standard width ring set (5/64", 5/64" compression, 3/16" oil), and a standard tension, moly-faced ring set is recommended. Keith Black pistons may require filing the top ring to achieve the minimum end gap specification. These engines were originally equipped with chromium-faced top rings, but molybdenum-faced top ring sets have been OE since the seventies. The chrome-faced rings offer improved abrasion resistance, but are slow to break in and are not necessary with modern air filtration.

Block and Cranktrain:
Normal block preparation should be accomplished including measuring main bearing saddle alignment. Align boring should only be done if absolutely necessary. Bore cylinders as required and hone to achieve piston manufacturer specified clearance and wall finish for moly-faced rings. Install new cam bearings as required. Chevrolet specifications for main and rod bearing clearance cover a wide range with very small minimums. For road engines I recommend .0015" main bearing clearance and .0020" rod bearing clearance, plus or minus .00025". A one thousandth undersized or oversized bearing half may be used with a one standard bearing half to maintain clearances within a .0005" range, if needed.

Through 1967, 300 HP engines have the same forged steel crankshaft as SHP/FI engines, but without the Tufftride surface hardening treatment that was applied to SHP/FI crankshafts. Grinding the journals will remove this surface hardening and is not recommended for SHP/FI crankshafts. Such is not the case for 300 HP crankshafts, however, only grind if necessary. Minor surface imperfections can often be removed by polishing the journals. Crankshafts should be checked for straightness, dimensional conformity with specifications, and qualified by Magnaflux inspection prior to any other work.

The OE 327 connecting rods are a critical weakness in all small bearing 327s. The first design used through 1965 is the least durable and are known to break at the bolt seat. The second design that went into production in 1966 has additional material adjacent to the bolt seat, which is an improvement. I do not consider the first design to be acceptable for an engine designed to rev over 5500. The second design is marginally acceptable for higher revs, but must be qualified by Magnaflux inspection. Second design rod durability can be improved with "race preparation" as outlined in the Chevrolet Power Manual and other sources, however, this is a labor intensive task and may end up costing nearly as much as a high quality set of aftermarket rods. I recommend Crower Sportsman rods. They have proven durability and at 585 grams are only marginally heavier than the 570 grams OE rods. Most high strength aftermarket rods are heavier, which may complicate balancing. In 2007 Eagle introduced a new small bearing rod, SIR5700SP, which has a specified weight of 590 grams. This SAE 4340 steel rod with 3/8" cap screws appears to have the required durability and can be purchased from retail sources for as little as half the cost of a Sportsman set. You can download the Crower and Eagle catalogs from their web sites.

www.eaglerod.com
www.crower.com

KB pistons have pin retainer grooves and spiral pin retainers are supplied, however, for road engines that are intended for long life I recommend pressed pins. Floating pins require rods that are configured for pin oiling and require careful attention to rod bushing clearance and pin endplay. Pressed pins eliminate these variables and potential problems. Floating pins are convenient on racing engines that are disassembled frequently for inspection.

If pressed pins are used, merely discard the KB-supplied pin retainers. Do no use them with pressed pin rods.

Balancing:
Rebalancing the cranktrain is required due to changes in rod and piston/pin masses with the new components. Precision balancing is necessary for engine smoothness. Equalizing all rotating and reciprocating masses to within one gram and adjusting the crankshaft end masses to within one gram to balance the first order rocking couple will result in very smooth engine operation, and owners of so-balanced engines also report elimination of the annoying "shifter buzz" that is common on vintage Corvettes. Flywheel and front damper balancing is also highly recommended. The flywheel should be balanced separately, then the clutch cover/pressure plate assembly should be bolted to the flywheel and the assembly balanced with any mass added or subtracted from the clutch cover. The clutch cover and flywheel should then be clearly marked to indicate correct indexing in order to maintain proper orientation upon installation. OE flywheels and clutch covers are stamped with an "X" to indicate assembly indexing.

Cylinder heads:

Proper cylinder head preparation and attention to detail is critical to achieving design performance objectives. The heads must be pocket ported and port matched to the inlet and exhaust manifolds and chamber overhang eliminated. Pocket porting/port matching only involves material removal up to about 1.5 inches above the valve seat on the "long side" radius, less on the "short side" and no more than one inch beyond the manifold interface. The objective is to improve port flow efficiency as expressed by the isentropic flow coefficient, not port size. Other than removing any obvious casting flash or severe roughness, the interior of the port should not be modified, and "polishing" is neither necessary nor desirable. Additional flow improvement can be achieved with a "three angle valve seat" on the cylinder head and a top cut off both valves to eliminate any unused portion of the valve seating surfaces. Also, radiusing the bottom of the valve circumference can aid flow, particularly on the exhaust side. Inlet/exhaust valve seat widths of .040/.060" will provide good longevity. Head and valve seating surfaces should be ground at a 45-degree angle and the valves lapped in. Details have been in publication for over 30 years in the Chevrolet Power Manual, "How to Hot Rod Small Block Chevys", and various books by David Vizard. Refer to these books for additional and important details.

Valve timing was specifically designed for the standard 1.94/1.5" valve set. A slight improvement in top end power at a slight expense to low-end torque can be achieved with larger 2.02/1.6" valves; however, I do not recommend the larger valves. Heads with larger valves are known to develop a crack between the valve seats, but this is rare with the 1.94/1.5" valve set. If the original valves have sufficient margin for grinding and no more than .0005" stem wear they may be reused. Exhaust valve stem wear is often greater than inlet valve stem wear, and since exhaust valves are subject to thermal fatigue, apriori replacement is often warranted.

If valve replacement is required, OE quality is sufficient, however better quality valves are available. The Federal Mogul Speed Pro S2323 1.5" exhaust valve is formed from 21-4N (21% chromium, 4% nickel) stainless steel and should be very long lived. Stainless steel is not at all necessary on the inlet side, and the Speed Pro 1.94" 8440 alloy steel with flash chromed stem will be very durable. "Race flow" valves are also available from the aftermarket that typically have undercut stems, but the author does not believe that these will offer a significant improvement at this configuration's level of power.

The OE integral cast iron valve guides with valve stem O-ring seals and valve spring shields are a fairly simple, reliable, and effective oil control method, and if the valve guides are not excessively worn, the same method can be used, however, I recommend Viton O-ring seals as they will outlast the standard nitrile O-rings at least two to one before they harden, crack, and allow oil to leak past.

There are several commercial valve guide rebuilding systems including cast iron guide inserts that effectively restore the guide to original specification. Use the valve guide manufacturer's recommended seals as different materials and valve guide technologies have different oil flow requirements. Also verify that valve stem material or surface treatment is compatible with valve guide material choice.

The cylinder heads should be mounted to the bare block after boring and any chamber overhang identified and ground away by beveling. This is particularly important on the exhaust side. Only remove the minimum material by bevel grinding to eliminate obvious overhang at the widest portion of the chambers. This will increase chamber volume slightly. Published data varies, but the OEM machined 461 heads with the 1.94/1.5" valve set are typically in the range of 61 cc. The later 462 heads with the standard valve size have slightly larger chambers, about 1-2 cc, because the small quench area on the spark plug side of the chamber was eliminated. Head chamber size should be measured after the heads are assembled. Use the compression ratio calculator to determine “best fit” – which head goes on which side to achieve he narrowest assembled CR range. Then grind chambers as necessary to bring down the highest computed CRs so the t total variation from highest to lowest is no more than 0.1. Head surfacing will reduce chamber size about 0.15cc for each .001" of material removal, which is 1.5cc for a .010" cut; however, heads should be surfaced only if they do not measure sufficiently flat to seal with the planned gasket type.

Heads that were originally machined for the larger 2.02/1.6" valve set include a chamber relief formed by a 2.40" or 2.34" diameter cutter centered on the inlet valve guide to unshroud the larger inlet valve. This adds about 1-2 cc to chamber volume. Installing larger inlet valves without performing this relieving operation will likely reduce flow relative to the standard 1.94" inlet valve.

Special valve seat hardening or hard seat inserts are not necessary for normal road use with unleaded gasoline. The insulating benefit of a thin lead oxide film on the valve seats is formed early in an engine's life if leaded fuel is initially used, so blending in 100LL avgas or leaded racing gasoline in the first few hundred miles of operation may offer a slight benefit that can last many thousands of miles.

Camshaft and Lifters:
Valve timing for this engine configuration was optimized to create the broadest SAE net torque bandwidth from 2000 to 6500 RPM while maintaining OE idle characteristics using accurate models of all components (including front end accessories and exhaust systems) in the Engine Analyzer 3.0 simulation program. Optimum duration at .050" tappet lift turned out to be about 225 degrees on the inlet side and 205 degrees on the exhaust side. Design preference was to use OE lobes since they are well proven in service, and the author has done a detailed analysis of their dynamic behavior. The closest OE lobes are from the inlet side of the 3896962 camshaft that was used on the L-46 and L-82 engine options from 1969 to 1979 and the exhaust lobe from 3896929 that was the base engine camshaft from 1967 to 1979.

Most aftermarket camshaft vendors manufacture camshafts from blanks that are received with the indexing pin installed and all machining accomplished except for final lobe grinding. The lobes are ground on mechanical machines that follow an oversized "master lobe" to grind the lobe to final contour. There are literally hundreds of small block master lobes available, and the cost of a custom camshaft that requires a new master lobe would be prohibitive. Crane Cams offers a reproduction of the 929 camshaft, but not the 962 camshaft, however, they offer a reproduction of the 3863151 camshaft that was used on the L-79 engine option from 1965 to 1968. This lobe has two degrees less duration at .050" tappet lift than the 962 camshaft, 222 versus 224 degrees and the same lobe on both sides. The 929 camshaft uses different inlet and exhaust lobes, and the exhaust lobe duration is 202 degrees at .050" tappet lift. The final decision for the Special 300 HP camshaft design was to use the L-79 lobe on the inlet side and 929 camshaft exhaust lobe on the exhaust side with indexing optimized for maximum average torque in the 2000-6500 RPM rev range while maintaining the same effective overlap as the 929 camshaft. Final specifications are as follows:

Type: Hydraulic flat tappet
.050" tappet lift timing points, deg.: IVO -4.5 BTDC, IVC 46.5 ABDC / EVO 43.5 BBDC, EVC -21.5 ATDC
.050" tappet lift inlet/exhaust duration, deg.: 222/202
Point of maximum lift (POML), inlet/exhaust, deg., lobe separation angle (LSA): 115 ADTC/123 BTDC,

119 deg.
Gross lobe lift, exhaust/inlet: .29807/. 27333"
Heel to toe length, base circle diameter inlet/exhaust: 1.53223", 1.23416" / 1.53133", 1.25800"

What I call "POML" is commonly referred to as "centerline", however, since both these lobes are asymmetrical, I prefer using more precise terminology because the actual "centerlines" of the lobes do not correspond to the POML.

Best warranty coverage is assured by using Crane supplied lifters. Their part number for the set is 99277-16.

Crane Cams has given assurance that they can grind the camshaft with the specified dimensions, POMLs, and LSA. Crane supplies camshafts to GM Performance Parts, which is reasonable assurance that their products are manufactured to acceptable quality standards including dimensional conformance and lobe hardness, however, in order to verify that this custom camshaft was manufactured to specification, measure the heel to toe dimension on all lobes prior to installation, and verify lobe indexing on at least one inlet and one exhaust lobe. Camshaft indexing checks must be made with a mechanical lifter. The first two articles produced for the prototype engines met specifications.

Valvetrain:
Only OE or OE equivalent valve train components are recommended to include, pushrods, rocker arms, and valve springs. Beginning in 1967, which was the first year for the 929 base engine camshaft, a new small block valve spring was introduced in production, 3911068, which features a slightly higher rate than the earlier valve spring. This valve spring was used on nearly all subsequent Gen I small block engines (including Z-28 and LT-1 mechanical lifter engines), exceptions including some later engines that had exhaust valve rotators, which required a different spring design. The Federal Mogul Sealed Power equivalent is VS677, and the Dana Clevite equivalent, which can be purchased at NAPA is 2121150. Use any of these three springs. Do not use higher rate aftermarket springs! Specifications are 76-84 pounds @ 1.70" and 197-203 pounds @ 1.25" with a free length of 2.03" and coil bind at 1.15". The nominal rate is 267 pounds/inch, and a .030" shim will alter installed height force by about 8 pounds.

Expectations are that this engine configuration will provide useable power to 6500 RPM, which will push the limits of lifter pump up speed, and attention to valvetrain assembly detail is necessary. Valve springs should be checked for conformity with specifications and shimmed using .030” and .015” shims as necessary to achieve a nominal installed height of 1.64” on the exhaust side and 1.67” on the inlet side to a tolerance of +.005/-.013”. At the typical 1.44:1 peak lift rocker ratio that is achieved on these engines, maximum inlet/exhaust valve lift is 0.429/0.394". The exhaust valve is slightly lighter, but exhaust lobe peak negative acceleration is slightly greater, which offset to some degree. The nominal .030” lower installed height on the exhaust side will yield peak lift spring restoring force equal to the inlet side, and with the minimum installed heights specified above, the springs will compress to approximately 1.23" at maximum lift, which provides .080” coil bind margin. The low installed exhaust spring height may result in the bottom of the spring being above the spring pocket edge if the OE spring shields are not installed. If such is the case, use the OE spring shields even if a positive type seal is used. Typical valve spring shims can also be used beneath the OE spring retainer, so this is another option.

Current OE replacement rocker arms are of the "self-aligning" design. The stamped rocker arm has a slot that centers on the valve stem. These are acceptable for use with any hydraulic lifter camshaft, but cannot be used for mechanical lifter camshafts. The OE pressed-in rocker arm studs are adequate if the heads have no history of pulling studs. Screw in studs can provide insurance if there is any doubt. Do not use pushrod guide plates unless the heads are modified to accommodate them, and under no circumstances use guide plates with self-aligning rocker arms. Pushrod alignment in early heads is provided by the pushrod bores. Heads that were originally equipped with guide plates have larger pushrod bores and require special pushrods that are hardened along their entire length. Early heads retrofitted with guide plates must have their pushrod bores enlarged and the proper hardened pushrods. Otherwise, the guide plates will gall and even bend the pushrods. The original non-self-aligning rocker arms without guide plates are perfectly adequate.

Existing OE type valvetrain components (except lifters) are suitable for reuse if they do not show excessive wear. Mating surfaces should appear burnished with no obvious wear or galling. A good visual inspection with a magnifying glass should reveal any excess wear that demands part replacement.

Rocker nut preloads vary widely, and the author knows of no minimum specification, however, I suggest replacing any nuts that have less than 10 lb-ft breakaway torque at engine teardown. Loose rocker nuts can also be peened to tighter preload by placing a nut flat on a hard surface and striking the opposite flat with a hammer. A little practice may be required.

Lifter Preload:
The shop manual specification for lifter preload is 3/4 to 1 turn down from zero lash. Since the rocker arm stud has 24 threads per inch, (and taking into account the measured 1.37:1 lash point rocker ratio) one turn down from zero lash preloads the lifter approximately .072", which is more than sufficient to compensate for normal valvetrain wear over the life of the engine. Additional valve train limiting speed can be achieved by "zero lashing" the lifters according to the Chevrolet Power Manual. This procedure calls for only 1/8 to ¼-turn lifter preload. Since high valve train limiting speed is an objective, I recommend ¼-turn preload. If greater preload is used and valvetrain limiting speed is below expectation, the lifters can be readjusted to less preload.

Induction System:
The camshaft has been engineered to perform well with the OE inlet manifold and carburetor. A modest increase in top end power at some expense to low end torque is achievable with higher flowing inlet manifolds such as the Z-28/LT-1 manifold and some current aftermarket manifolds. The aluminum 327 SHP manifold offers some improvement over the production cast iron 300 HP manifold. "Single plane" manifolds are specifically not recommended. OE carburetors have adequate flow and OE jetting is likely adequate, however, chassis dyno testing with a wide band O2 sensor is recommended to determine if improvement is required. Carburetors are often rich at WOT, low revs and lean out somewhat as revs increase. Ideally, the carburetor should provide an A/F ratio in the range of 12.5-13.0:1 at high revs. If the ratio leans out beyond 13.5:1, the engine may have more propensity to detonate.

Cruise mixture can be tested on an inertia dynamometer by starting a top gear run at about 1500/16" and increasing throttle opening as revs build to end at about 3500/12". A little practice may be necessary. For these runs the power is meaningless as the intent is to gather A/F ratio data at typical cruise revs and load. A cruise A/F ratio in the range of 14.0-14.5 is usually adequate. Richer mixtures waste fuel and a leaner average mixture may place the leanest cylinders into a misfire condition, which can usually be felt by the driver as engine roughness or hesitation at cruise or light acceleration.

Some 300 HP configurations have a restrictive air cleaner, which includes the early C2 "dual snorkel" air cleaner. Later 300 HP configurations have an open element air cleaner, which is much less restrictive. Owners with the early, restrictive air cleaners should consider one with less restriction, such as the same year SHP air cleaner or an aftermarket open element air cleaner that can be easily integrated to the original PCV connections without permanent modification.

Always use a quality, name brand cellulose air filter element. Both the vintage OE urethane foam elements and aftermarket non-cellulose elements have poor particle filtration performance, and published tests show that heavily advertised aftermarket non-cellulose elements do not flow more air than typical name brand cellulose elements of the same size.

Exhaust System:
The Special 300 HP camshaft's low overlap is designed specifically for the OE exhaust manifolds and modest to medium vehicle exhaust system backpressure. Headers are specifically not recommended and will provide no material benefit. The C2 exhaust system with 2.5" manifolds and 2.5" pipes is very efficient and will only generate about 3 psi backpressure at the power level of this configuration, which minimizes exhaust pumping loss. The smaller 2" manifolds on 1966 and later engines will increase backpressure slightly. The 2" pipes used on 1962 models, even with 2.5" manifolds, will increase backpressure to about 4.5 psi and reduce peak power by about three percent relative to the C2 2.5" manifold/pipe system. From 1962 to 1965, 300 HP engines with manual transmissions were equipped with 2.5" manifolds and 2.0" manifolds with Powerglide. After 1965, all small block configurations were equipped with 2" manifolds.

Oiling System:
The original C2 300 HP 4 quart oil pan (5 quart total system with filter) is adequate for occasional excursions to the 6500 RPM design speed. Also, use a standard volume, standard pressure oil pump. Do not use a "high volume" oil pump or increase the oil pressure beyond the normal 40-45 psi hot at 2000 RPM specification. Excess volume or pressure increases parasitic power loss and oil temperature and is not necessary for a road engine. The installed oil pump can be inspected, and if there is no evidence of excess wear, it may be reused. If the pump is replaced, use a standard volume, standard pressure OE replacement pump, and be certain that the pump pickup is identical to the OE pickup, which is unique to some Corvette configurations including all C2s. Pump performance can be optimized by dressing the housing (and longest gear, if necessary) down to achieve .002-.003" end play on both gears.

If sustained high rev operation (over 5000 RPM) or high dynamic loading (road racing) are anticipated as normal use, install the SHP/FI oil pan and windage tray, and optionally install a 55-60 psi relief spring in the standard volume OE or OE replacement pump. Standard volume pumps with the higher pressure relief spring were OE on mechanical lifter engines beginning in late 1963 and carried forward to the last 327 mechanical lifter engines in 1965.

Ignition System and Spark Advance Map:
The OE Delco single point distributor will provide sufficient spark energy and a reliable spark to 6500 revs if careful attention is paid to assembly clearances and a high breaker arm tension (28-32 oz.) point set is used. The shaft should fit snugly in the housing bushings and the breaker plate should fit snugly on the housing without wobble. Shaft end play should be shimmed to achieve .002-.007", minimum preferred. If distributor end play is at the high end of the range with available shims, add a .010" shim and dress the thickest installed shim on a flat surface with 280-400 paper wetted with mineral spirits a suitable

amount to achieve the lower part of the range. Tight end play reduces spark scatter. All distributors should be disassembled, cleaned, clearances thoroughly checked, upper bushing grease well replenished, and assembled to achieve the above specifications, and the vacuum and centrifugal spark advance curves (which together form the spark advance map) should be verified for compliance with OE spec or similar as described below. It is a good idea to measure the spark advance map prior to beginning the project, so the baseline is known.

Since idle characteristics will be the same as the OE 300 HP engine (500 RPM @ 18-19" Hg. in neutral), a vacuum advance control (VAC) that provides full vacuum advance of 16 crankshaft degrees at no more than 15-16" Hg. is required. This is met by most OE VACs; 1966 and 1967 300 HP engines have a 12" VAC, which is acceptable, but might increase the propensity to detonate at part throttle acceleration. The NAPA/Echlin VC1802 (0 deg. at 8", 16 deg. at 15") meets the above requirement. It is stamped "B22". It can be cross referenced on other brand web sites, and all, including Delco brand, will likely have the same stamped ID. Test the VAC to ensure that it meets specifications before installing, even if it is new.

Total wide open throttle (WOT) spark advance - the sum of initial timing and maximum centrifugal advance - should be in the range of 34-38 degrees. Total OE centrifugal advance on 300 HP engines varies by year from 24 to 30 degrees, which dictates initial timing in the range of 4-14 degrees. If low rev detonation is evident on 24 degree distributors, it can be mitigated by grinding the advance slot to increase centrifugal to greater than 24 degrees and reducing initial timing an equal amount. The OE centrifugal curves are relatively slow, and quickening the curve to the ragged edge of detonation will increase low-end torque. Spring kits are available from aftermarket vendors, and the low dynamic compression ratio of this configuration should tolerate a fairly quick centrifugal advance curve - similar to the '64-'65 SHP/FI engines, which is all in (24 deg.) at 2350 RPM.

Total idle spark advance is the sum of initial plus full vacuum advance and, depending on the initial timing, should be in the range of 20-30 degrees. The VAC signal source must provide a full manifold vacuum signal under all operating conditions, including idle. Convert any "ported" VAC signal lines to full time, which includes engines originally equipped with K-19 exhaust emission control equipment.

Final centrifugal and initial timing calibration is best achieved by road testing, and keep in mind that higher inlet air temperature and/or coolant temperature increases the tendency to detonate, so the worst case is usually a hot summer day. Along with proper carburetor calibration, an optimized spark advance map will have a significant impact on torque bandwidth and top end power bandwidth in the upper 30 percent of the rev range (approximately 5000-6500). The OE calibrations are a good starting point, but can likely be improved to the point of meaningfully increasing torque and power bandwidths. Individual engine build characteristics such as compression ratio along with operating conditions to include typical ambient pressure and temperature and available fuel octane are all variables in the equation. Time and budget spent "fine tuning" both the spark advance map and carburetor calibration will pay dividends.

The following parts references will aid your distributor setup efforts.

Brand Part no. Description Remarks

Mr. Gasket 2820 Dist. shaft shim kit 2-.010", 1-.020", 1-.050"
Moroso 26140 Dist. shaft shim kit 2-.010", 2-.020", 1-.053"
Mr. Gasket 925A Dist. curve kit 3 sets of springs, weights...

Moroso 72310 Dist. curve kit 3 sets of springs, weights...

Curve kits include three spring sets. The OE and aftermarket springs can be mixed as required to achieve the best centrifugal curve. The installed springs do not have to be matched, but avoid one very light and one very stiff spring. The curve kits include weights, but there are reports that some aftermarket weights are not sufficiently hardened and the pivot holes wear. Use the OE weights whenever possible.

Most other distributor/ignition parts except the cam lubricator and breaker plate are available in the traditional aftermarket ignition components brands (GM still offers the plastic seal, 1950569), and all these brands are now owned by Standard Ignition Products, so they are the same parts. Also, most Delco brand parts are now purchased from Standard or other sources, not manufactured by GM.

Replacement Parts:
With the exception of the camshaft/lifters, pistons, connecting rods, and some small parts I recommend OE or OE equivalent name brand replacement parts. When these cars were built, GM manufactured many internal engine parts, but since the nineties GM has sold most of their parts manufacturing operations and buy from outside suppliers. Though many OE replacement parts are still available from GM Parts Division, they are likely manufactured by outside suppliers such as Federal Mogul and Dana Corporation. Both Federal Mogul and Dana offer extensive lines of OE replacement parts, many of which are very likely the exact same parts sold by GM, but the replacement brand parts are available from many sources with good price competition. Brand names such as Sealed Power and Felpro are Federal Mogul brands. Clevite, Perfect Circle, and Victor Reinz are Dana brands. You can find most part numbers by searching available online catalogs and cross-references and since Dana owns NAPA, they carry the complete line of Dana brands.

www.federalmogul.com
www.danacorp.com

Assembly and Break-in:
A thin film of "assembly lube" (and I emphasize thin) should be applied to cam lobes, lifter heels, and other valvetrain mating surfaces. Pistons rings, bearings, and other mating components should be assembled with API service category CJ-4 oil, and a bottle of GM EOS or equivalent, which is rich in ZDDP anti-wear additive should be added to the crankcase. Prefill the oil filter before installing. Using an engine pre-oiler to rotate the oil pump with a gage attached to the oil pressure signal port and/or watching for oil at the rocker arm nozzles while slowly rotating the engine is a good idea.

Original GM camshafts were Parkerized and did not require a "cam break-in" with the modest force OE valve springs. Crane also Parkerizes their cast iron camshafts, and this is usually evident by a dull gray finish on the lobes. Notwithstanding the above, to ensure long-term durability I recommend a cam break-in be accomplished by starting the engine and running it at 2000-2500 revs, no load for at least 20 minutes. Since assembly lubricants have solids that will be trapped by the oil filter, change the filter after cam break-in and replenish the crankcase as required. Research over the years indicates that new engines can exhibit a high rate of particle generation during break-in, but then it drops off dramatically until end of life. For this reason, change the filter again at about 500 miles, but leave the original oil with the EOS in the engine. A complete oil and filter change should then be accomplished at 1000-1500 miles and from that point on every 5000 miles or no more than one year for engines that see less than 5000 miles annual service. Continue to use API service category CJ-4 oil.

Fast initial start is ensured by prefilling the carburetor bowls with gasoline, and static timing the engine. Set the balancer notch at the chosen initial timing value on the timing tab (not TDC) and rotate the distributor housing until the points just open using an ohmmeter. This will place the initial timing within about a degree of your initial specification. Prior to shutting down the fully warmed up engine on the last run before work begins, set the fast idle cam in the normal cold start position and adjust the fast idle screw to achieve about 2400 RPM. This will initially yield about 2000 RPM on the first cold start after assembly, which will allow you to inspect the engine for leaks or other issues during cam break-in without having to monitor the throttle.

No special procedures or precautions other than the above are required if the engine is expected to sit for months, or even years, before it begins normal operation, however, the engine must be stored in a non-condensing humidity environment to preclude the formation of internal corrosion and the exhaust and inlet openings should be covered to prevent dust entry. If new valvetrain components are used (pushrods, rockers and balls) and valve lifter preload was set at a quarter turn or less, re-accomplish the procedure at 1000-5000 miles and thereafter every 30,000-60,000 miles for either used or new components. If the original valvetrain components were installed in their original locations, little valvetrain run-in wear will occur, so the initial recheck can be deleted, and in either case, if one-half turn or more was used at assembly, no further adjustment is usually necessary.

Operating conditions and fluids
Idle: Approximately 500 RPM in neutral at approximately 18-19" Hg. manifold vacuum
Ignition timing: (see above discussion under Ignition timing and spark advance map)
Maximum recommended engine speed: 6500 RPM
Coolant: Hybrid organic acid technology (HOAT) type antifreeze such as Zerex G-05 mixed 50/50 with distilled water
Thermostat, normal operating temperature: 180 degree F thermostat, 180-230 degrees F with 15 psi relief cap
Engine oil: API service category CJ-4, SAE 15W-40 for cold starts down to 10F (5W-40 for cold starts below 10F)
Hot oil pressure: 40-45 psi @ 2000 RPM
Fuel: Commercial unleaded premium gasoline, 91 Pump Octane Number (PON), minimum

Depending on head work, assembly detail, ignition map, air-fuel ratio optimization, and air cleaner and exhaust system restriction, the expected beginning of the 80 percent torque bandwidth is no more than 2000 RPM with the top of the 80 percent torque bandwidth at over 6000 RPM. Expected peak SAE net power is in the range of 290-310 horsepower in the 5500-6000 RPM range with a useable power bandwidth to about 6500. Peak torque values (lb.-ft.) should be slightly higher at near 4000 RPM. Dyno-jet 248 inertia chassis dyno data with SAE air density correction will be about 0.85 of these values to account for drivetrain and tire loss. - near 270 lb.-ft. and 260 horsepower, which is about one-third more top end power than OE with 1000-1500 more useable revs without altering the engine's docile character. During chassis dynamometer tests, sufficient external fans and dwell time between pulls should be established to prevent the viscous fan clutch from tightening. I estimate that fan clutch tightening will reduce peak torque readings by about 15 lb-ft and peak power by about 10 horsepower.

Duke Williams, Engine System Engineer Revision 2, February 2008