

A Tale of Two Camshafts  
by Duke Williams  
with Dennis Clark, Mike McCagh, and John McRae

*[Editor's note: This article was originally published in the Fall 2010 "Corvette Restorer". This revised edition is presented with the permission of the authors and NCRS.]*

In the last few years I have completed several vintage Corvette engine system engineering projects. The first two involved a '65 L79 and a '65 L76. The owners wished to maximize power without affecting normal operating behavior or original visual appearance. Using the Engine Analyzer 3.0 engine simulation program with head flow data provided by the owners, I developed a configuration that I named *327 LT1*, which consists of "massaged" heads, the 1970 LT1 camshaft, and upgraded connecting rods to ensure a bulletproof bottom end. The original rods, particularly the pre-'66 version without the additional material adjacent to the bolt seats, are the 327's Achilles heel.

One engine was tested on a laboratory dynamometer and the other on a Dynojet inertia-type chassis dynamometer approaching 300 SAE corrected rear wheel horsepower at 6500 revs with 80 percent of peak torque at 2000 (which was a primary requirement) 90 percent at 2500, and a useable power bandwidth to 7200. Given the success of these projects, I turned my attention to the 300-horsepower engine, which is the most commonly installed engine of the 1962-68 period. Base 350s from 1969 to 1974 will show similar improvements, but the highly restrictive single catalyst exhaust system beginning in 1975 will mostly negate internal engine flow improvements due to excessively high exhaust backpressure that greatly increases exhaust parasitic pumping power.

## **Engine Performance**

How many times have I heard guys say, "I want more power"? But peak power is only available at the top of the rev range, and we spend 99.9 percent of our driving time below this range. There's an old expression variously attributed to engineers from both GM and Mercedes Benz: "People buy power, but they like torque." How true! An engine with high low-end torque has high low-end power and will feel very responsive in normal driving—no waiting for the engine to "come on the cam." An excellent comparison within the 327 Corvette engine family is the 300-horsepower version and the 365-horsepower L76. The 327/300 is very pleasant to drive in normal traffic with its docile manners and stump-pulling low-end torque while the 365-horsepower engine is notoriously torque-shy. Granted, the 327/300 fades away beyond 4500 while the 327/365 keeps pulling to the 6500 tachometer redline, but you pay a price in lower fuel economy and busy highway operation due to the shorter required gearing to keep this engine higher in the rev scale at normal driving speeds.

Understanding vintage Corvette engine performance begins with realizing that the advertised *SAE gross* ratings are highly optimistic and little more than advertising copy. Gross torque is somewhat overstated, and power is overstated

by up to 20 percent. SAE gross ratings were based on laboratory dynamometer tests of stripped engines: no front-end accessories other than the coolant pump, large exhaust pipes and no mufflers with an evacuation pump that reduces exhaust back pressure below atmospheric, and test data corrected to *standard sea level conditions*—29.92 inches Hg. barometric pressure, 59 degrees Fahrenheit, dry air. These correction conditions are also commonly referred to as STP (standard temperature and pressure) and are still the defacto standard for field laboratory dynamometer test correction. Fuel flow and spark advance maps did not have to be the same as production, and “flash readings” were accepted as real data. Then it was all sent to the marketing guys for more massaging.

In recognition of the games manufacturers were playing with the numbers during the halcyon days of the horsepower wars, all agreed to adopt a new *SAE net* rating system beginning in 1971. That year both gross and net ratings were advertised, but only net from 1972-on. SAE net ratings require the engine to be in the same configuration as installed in the vehicle, which includes production fuel flow and spark advance maps, all front-end accessories, the engine air-induction system, and the full vehicle exhaust system. In addition, observed data are corrected to 29.23 inches Hg., 77 degrees Fahrenheit, dry air—representative of a warm spring day in Chicago, rather than a mild spring day in Seattle. These same conditions also apply when SAE correction is applied to chassis dynamometer data. This lower air density correction alone reduces torque and power by about *4.5 percent* relative to STP conditions. The remaining losses are from front-end accessories and the vehicle exhaust system.

Various rules of thumb have appeared over the years to convert SAE gross to SAE net, which are usually in the range of 80 percent, but it varies considerably with the specific installation, and the exhaust system is *critical*. Most vintage Corvettes are blessed with very efficient exhaust systems. Zora was clearly thinking ahead when the C2 frame was designed to accommodate 2.5-inch exhaust pipes through the center cross member. Exhaust systems for many vehicles of that era were little more than afterthoughts, which includes many big block muscle cars, and those restrictive exhaust systems ate up a lot of power between the dynamometer cell and vehicle installation.

I have been collecting dynamometer tests of all sorts for many years, and my computed empirical net/gross conversion factor based on these data for 327s with 2.5-inch exhaust pipes is 0.89, and air density correction alone accounts for *40 percent* of the reduction! The 327 LT1 only generates a little over 3 psi backpressure at close to 300 SAE corrected rear wheel horsepower, which is very modest and as good as modern high-performance cars. If the fan clutch does not tighten, its power consumption is small—only one to two horsepower—but up to 15 pound-feet torque and 10-plus horsepower if it fully tightens. The 1962 model’s 2-inch pipes, even with the off road mufflers, are somewhat more restrictive, and Special High Performance big blocks will approach 6 psi due their greater exhaust flow. (Exhaust backpressure increases with the *square* of flow volume.) Many wonder how much improvement the 2.5-inch outlet exhaust manifolds installed on most 327 configurations from ’62-’65 yield over the 2-inch outlet versions used on all

later small blocks. I don't have enough data to nail it down, but I doubt if the difference is more than one to two percent.

## **Cylinder Heads.**

As I've said so many times before, the way to get more power is to *massage* the cylinder heads—work the heads for power and select a camshaft that provides the necessary low-end torque for the type of service. For a high performance road engine with a manual transmission, my target is no less than 80 percent of peak torque at no less than 2000 revs. This is a basic *engine system engineering* requirement, and for automatic transmissions my requirement increases to 90 percent at 2000, which is at or just above torque converter stall speed for most OE automatics. Anything less and the engine will feel soggy and unresponsive in normal driving environments.

Head massaging doesn't mean hogging out the ports to the edge of the cooling jacket passages. Actually, little material needs to be removed. The point is to increase the port *flow coefficient*, not port cross-section area. Engineers use a concept called *isentropic flow* that is an easy way to analyze the flow potential of a duct. It ignores such real world complications as heat transfer and friction, but yields simple formulas to compute idealized flow potential. The ratio of actual flow to computed isentropic flow, the *flow coefficient*, is a measure of the design. As built, OE small block head ports are typically in the 0.4 to 0.5 range. The best purpose-built racing heads exceed 0.70, and with proper rework a set of OE vintage Corvette cylinder heads can achieve about 0.5 on the inlet side and about 0.6 on the exhaust side. The exhaust port has higher flow coefficient potential because it is short and less convoluted than the inlet port, which has to wrap around the pushrod passage, but the exhaust port is more sensitive to the details of hand work and typically shows more flow variation from job to job.

The biggest impediment to OE cylinder head flow is the annular ridge that remains from the first valve pocket roughing operation where the machined area meets the as-cast port. Grinding down this ridge to smoothly blend the machined section to the as-cast section is the most critical operation and is generally known as *pocket porting*. The sides of the combustion chambers should be beveled back to the edge of the cylinder bores, and downstream overhang at all inlet and exhaust port junctions should be ground away. The exhaust pipes should fit snugly in the manifold seats with any overhang corrected with a little hammering.

Valve seats should be enlarged leaving .010-inch valve overhang with .040/.060-inch wide inlet/exhaust seat using 30/60-degrees top/bottom cuts to narrow the seats to these values. Remove the upper unused portion on the valves with a 20-25 degree cut, leaving the same .010-inch overhang. This creates an annular venturi that improves low lift flow. Any significant casting flash in the port interior should be ground down, but *no polishing!* In fact, the final results

should not look pretty. Surface roughness helps break up the laminar boundary layer at the wall to keep fuel off the port walls and in the air stream. That's *head massaging* in a nutshell, and you can see the results in Table 1.

**Table 1** Typical Head Flow and Flow Coefficients  
(CFM at specified valve lift, 28" water test depression)

Head ID Valve sizes	Inlet flow/ Flow coeff. (lift)	Exh. flow/ Flow coeff. (lift)	E/I flow ratio
OE 462 1.94/1.50"	200 / 0.48 (0.45")	135 / 0.50 (0.45")	0.68
McRae 462 1.94/1.50"	230 / 0.55 (0.45")	160 / 0.59 (Note 1)	N/A  (Note 1)
McCagh 997 1.84/1.50"	165 / 0.47 (0.40")	130 / 0.48 (0.40")	0.79
Massaged 461 1.94/1.50"	230 / 0.55 (0.45")	180/0.67  (0.45")	0.78
Massaged 461 2.02/1.60"	230 / 0.53 (0.45")	190/0.62  (0.45")	0.83

Note 1: Additional work was done to the exhaust ports following this test, but the flow was not retested.

**Representative head flow data from the author's files. Note that 1.94/1.50-inch valve heads flow about the same as 2.02/1.60-inch valve heads. The larger valve sizes show more advantage at lower lifts, but are not a major factor in engine performance. Some tests show a reduction in inlet flow above 0.45-inch lift, probably due to some shrouding mechanism. Given real rocker ratios, which are less than the advertised 1.5:1, actual valve lifts on the engine never achieve 0.45-inch with any OE camshaft.**

These procedures have been in publication for over 35 years. *Hot To Hot Rod Small Block Chevys*, the *Chevrolet Power Manuals* going back to the seventies, and various books by David Vizard provide all the details—enough for any restorer to do his own work other than the final seat and valve grinding. It's a time-consuming, labor-intensive process, but well worth the effort if you want to achieve maximum power and extend the useable power range to the valvetrain limiting speed while maintaining original visual appearance and normal driving and idle behavior. Alternatively, a knowledgeable professional may do the work, but given the labor time, it is expensive!

## Matching Valve Timing to Cylinder Head Flow

Valve timing – what is manifested in the camshaft design, like the spark advance map, is nothing more than a *tuning parameter*. The shape and relative position of the torque curve is highly affected by valve timing. Peak torque is primarily a function of displacement and compression ratio. Consider the 250-horsepower and 375-horsepower Fuel-Injection 327 engines. Both carry the same advertised SAE gross torque rating of 350 pound-feet, but the Fuel-Injection engine doesn't achieve peak torque until a lofty 4600 revs versus a mere 2800 for the 250-horsepower engine. The advertised torque ratings of the entire 327 Corvette engine family vary from 344 pound-feet (1962-63 340 horsepower) to 360 pound-feet (300 horsepower and 350 horsepower L79)—a mere 4.7 percent spread—yet the highest rated power is 50 percent greater than the lowest! Granted, the power ratings are highly optimistic, but relative comparisons tend to wash out much of the marketing manipulation.

Optimizing valve timing to achieve maximum torque bandwidth requires matching valve timing to port flow characteristics. In particular, the exhaust/inlet (E/I) flow ratio is a critical parameter. As built by Flint, 461X and later big port heads have relatively low exhaust flow. The E/I flow ratio is typically in the range of 0.65 to 0.70. Ideally, it should be near 0.75, which is suitable for equal duration on both sides.

Some camshaft designs beginning in the mid-sixties recognize the restrictive exhaust port by providing somewhat longer exhaust than inlet duration manifested as a relatively early opening exhaust valve. But the equation changes with cylinder head massaging. Inlet flow typically increases 10-15 percent, but 20-40 percent on the exhaust side. This yields an E/I flow ratio near 0.80, so exhaust duration can actually be *shorter* than inlet duration with a relatively late phased exhaust valve opening point, but I know of no off-the-shelf camshafts that offer these features. Neither OE nor aftermarket camshafts are optimized to the flow characteristics of massaged heads!

## First Design

With the above principles in mind and using average flow values from my library of massaged head flow data, I began working on what I call the *Special 300-Horsepower* camshaft. Starting with the .050-inch lifter rise timing points of the 3896929 camshaft that was used on all base Corvette small blocks from 1967 to 1979, I held the inlet valve opening (IVO) and exhaust valve closing (EVC) points to maintain the low effective overlap, which is required to preserve original idle behavior.

Of the four valve timing points, inlet valve close (IVC) has the greatest effect on top-end power. As IVC is retarded, the torque curve is shifted up the rev scale resulting in more top end power, but less low-end torque. However, a high-

flowing exhaust port allows exhaust valve open (EVO) to be retarded so cylinder pressure can work a little longer, and this partially offsets the low-end torque loss from a late-closing inlet valve. When low-end torque dropped to about 80 percent at 2000, I froze IVC and began varying EVO, found some improvement, then retarded the all timing points until torque again dropped to 80 percent at 2000.

This iterative process occurred on and off over what was a period of months, and the final design indicated about 220 degrees .050-inch lifter rise duration on the inlet side and close to 200 degrees on the exhaust side with the inlet lobe point of maximum lift (POML) phased rather late compared to typical OE and aftermarket camshafts. (Note: Since most OE lobes are asymmetric, the POML is not the same as the lobe centerline, but most aftermarket vendors use “centerline” to represent the POML even if the lobe is asymmetric.)

Clearly, this was a radical design—no vintage OE or aftermarket camshafts are even close! I actually began creating a lobe design program using Excel, where I input a lobe acceleration profile and Excel computed the velocity and lift profiles, but there were a couple of problems with this approach. First, a custom design lobe requires lobe master tooling, which costs at least several hundred dollars. Second, I know what kind of acceleration the OE valvetrain can tolerate based on my dynamic analysis of the OE lobe data, but I wasn’t sure what kind of jerk (the rate of change of acceleration, which is associated with shock loading) the OE valvetrain can tolerate. I do know from my OE lobe data analysis that beginning with new designs in the sixties, jerk was significantly reduced from the previous designs of the fifties. About 15 years ago I asked Ed Iskenderian about acceptable jerk profiles, and he just gave me a wry grin. It’s a matter of experience for those who have designed, analyzed, and tested a lot of lobe designs and valvetrain components.

Then it dawned on me that there are a couple of OE lobes very close to the durations I needed—the L79 lobe for the inlet side and the 3896929 inlet lobe for the exhaust side. Furthermore, use of OE lobes and OE valve-train components, including valve springs, maintains the highly reliable and trouble-free small block valvetrain, especially since these lobes are the later low-jerk designs. I verified that Crane Cams has these lobe masters in their tooling library, so they could grind the camshaft at a reasonable price. The design was frozen (Table 2), and the next phase was to build and test.

**Table 2  
Basic Camshaft Specifications**

<b>Camshaft ID</b>	<b>.050” dur. I/E, deg.</b>	<b>Inl. POML deg.</b>	<b>.050” IVC deg.</b>	<b>Exh. POML deg.</b>	<b>.050” EVC deg.</b>	<b>Lobe Sep. Angle, deg.</b>	<b>Split Overlap deg.</b>	<b>Eff. Overlap sq.-in.- deg. (Note 1)</b>
3896929 (300 HP)	194/202	108 ATC	25 ABC	116 BTC	15 BTC	112	4 BTC	0.9
3863151 (L79)	222/222	110 ATC	41 ABC	118 BTC	7 BTC	114	4 BTC	4.3
Special 300 Horsepower	222/202	115 ATC	46 ABC	123 BTC	22 BTC	119	8 BTC	0.9
McCagh Special	202/194	116 ATC	37 ABC	108 BTC	11 BTC	112	4 ATC	0.9
2001 LS6	204/211	118 ATC	39 ABC	114 BTC	9 BTC	116	4 ATC	1.1
Typical hotrod type	230/230	106 ATC	41 ABC	114 BTC	1 ATC	110	4 BTC	9.1

Note 1: Normalized to 1.94/1.50-inch valve sizes and 1.44:1 rocker arm ratio

**Typical .050-inch lifter rise specifications, points of maximum lift, and lobe separation angle don't tell the entire engineering story. Lower level engineering details such as the velocity, acceleration, and jerk profiles can effect output and valve train durability. The OE lobes have proven durability when used with all OE valve train components, including valve springs. The two custom design vintage small block and 2001 LS6 camshafts inlet lobes are optimized for broadest torque bandwidth and have significantly different phasing compared to the sixties vintage designs. Exhaust lobe duration and phasing for the two custom designs and LS6 are optimized for the specific E/I head port flow ratio. Note the very high effective overlap of the typical hotrod type camshaft. This causes poor idle quality, poor fuel economy, poor low-end torque, and no significant top end power with manifolds and mufflers than the best of the other designs. Avoid this type of camshaft on a road engine!**

I posted a note on the NCRS Technical Discussion Board to ask if anyone contemplating a 327/300 engine restoration was interested in testing this camshaft. I stated that with suitable head work the camshaft would maintain the smooth idle behavior (500 at about 18-19” Hg. manifold vacuum in neutral) with nearly as much low-end torque, but substantially more top-end power and another thousand useable revs. Longtime NCRS member and holder-of-many-

offices Dennis Clark stepped up to the plate. Dennis' 1962 "Beanie Baby," named for the installed Halibrand "Kidney Bean" wheels, has a numbers matching 327/300 engine. It had been previously rebuilt with what sounded (at idle) like an aftermarket camshaft with more than OE overlap. The second volunteer was John McRae. John bought a '67 that had been sitting for 20 years after an altercation with a deer, and all indications were that the engine had never been opened up.

I prepared a technical support paper that discussed some of the finer points in detail. For example, Keith Black KB157 hypereutectic pistons (0.5 cc dome) with proper head gasket selection will increase the actual compression ratio to near 10.5:1, and the late-closing inlet valve can tolerate higher than OE compression ratio. Despite what GM advertised, the typical as-built compression ratio of 327/300s is about 9.7:1. In addition, very careful attention to valve spring (OE 3911068/Sealed Power VS677) installed height will maximize valve train limiting speed to exploit the full expected maximum power range. Both Dennis and John had previously purchased a set of Crower Sportsman connecting rods to ensure high rev durability through a group purchase arranged by Scott Marzahl on The Corvette Forum, so all they needed was the camshaft. (The subsequently released Eagle SIR5700 is now a good choice and less expensive.) My general specification for all other parts is OE or OE equivalent, and virtually everything is available in the Sealed Power and Fel-Pro (Federal Mogul) and Clevite and Victor Reinz (Dana Corp.) brands.

After coordinating with Jerry Clay of Crane Cams to determine what information they needed to grind the camshaft, I e-mailed Dennis and John a request for a quote (RFQ) with all the appropriate specifications and Jerry's contact information. Since the orders were placed in mid-November, near the end of the racing season, Crane's workload was light. Dennis' camshaft was delivered to his Washington State location in about two weeks, and John's cam was delivered to his Vermont address in barely a week! Both cams were clearly Parkerized, and summary dimension checks with a caliper showed they were within specifications. All indications were that these camshafts are high quality components in compliance with Chevrolet drawing specifications with the exception of lobe phasing, which was checked on the engines and found to be within acceptable tolerance of my specifications.

Dennis' project started soon thereafter. His longtime engine builder did the work, so I didn't get much visibility into the details, but Dennis' '62 was on the road within a couple of months. John did his own work except for machining and head flow testing. Initial flow test results (Table 1) showed excellent improvement on the inlet side, but less than expected on the exhaust side. After some consultation, John did a little more grinding in the exhaust ports, but there was no retest.

John is an engineer who owns a company that manufactures geotechnical instrumentation, and he did a superb job documenting his project. At one point when he was working on optimizing the compression ratio, I suggested that he take a couple of tenths (of a cc) out of a couple of chambers to narrow the high/low spread to less than 0.10. (See *The*



*Corvette Restorer*, Fall 2009, for the full story.) When I asked if I was placing too many demands on him, John replied, “No, I love this stuff.” There’s nothing like a couple of engineers building the “perfect engine.” The only trouble is finishing the project as we constantly tweaked in search of engineering perfection.

### **What About Me?**

It wasn’t long before Mike McCagh got wind of Dennis’ “Beanie Build.” Longtime NCRS members Dennis Clark and Mike McCagh have a decades-long friendly rivalry including an infamous race at the 1991 National Convention at Anaheim, California. No way was the “Maryland Missile” going to let “Dipstick” one-up him with a cheater motor. I found out that Mike was working on his own creative interpretation of a very rare ’57 RPO 579B 283/250-horsepower Fuel-Injection engine with Powerglide (see the Fall 2010 *Corvette Restorer* for Mike’s story about the restoration and Duntov Award), and he was interested in the camshaft. The correct 548 block carries a .060-inch overbore and a 350 crankshaft for 339 cubic inches, custom pistons, and 10:1 compression ratio. The correct 997 heads are reworked including 1.84-inch inlet valves, and power is routed through the Powerglide to a 3.70:1 axle. Very interesting! I was beginning to feel like Smokey Yunick.

I told Mike I was concerned about low-end torque with the Special 300-Horsepower camshaft. With Powerglide I want to see 90, not 80, percent of peak torque at 2000, but I had another idea, and if he had head flow data I could come up with a design in short order. Since Mike’s engine restoration was in progress, there was no time to spare, and fortunately, Mike had head flow data (Table 1). Upon reviewing the data, I felt a little more work could improve flow to something closer to OE-machined big port heads, but Mike didn’t want to take any more chances grinding these rare heads, so I said fair enough and went to work.

My idea was to just swap the lobes on the 3896929 camshaft with the same lobe separation angle to provide the same *effective overlap* and index the inlet POML as late as possible commensurate with the 90 percent of peak torque at 2000 requirement. To achieve the same effective overlap (which is computed by Engine Analyzer in square-inch-degrees) the Special 300-Horsepower camshaft required a 119 degrees lobe separation angle (LSA)—quite large for typical vintage engines, but right in line with modern Corvette engines. The only real difference is that modern Corvette engines require more exhaust than inlet duration. For example, like Gen I big port heads, the 2001 LS6 head exhaust port is restrictive with an E/I flow ratio of about 0.70. The good news is that a restrictive exhaust port can be mostly offset by opening the exhaust valve early and closing it a little late. This is evident in the 2001 LS6 valve timing numbers and .050-inch lifter rise durations (which are very similar to the LS2/3 camshafts; in fact some lobes are shared between them) – 204 degrees on the inlet side and 211 on the exhaust side with 116 LSA and a *very late* phased inlet event (Table 2).

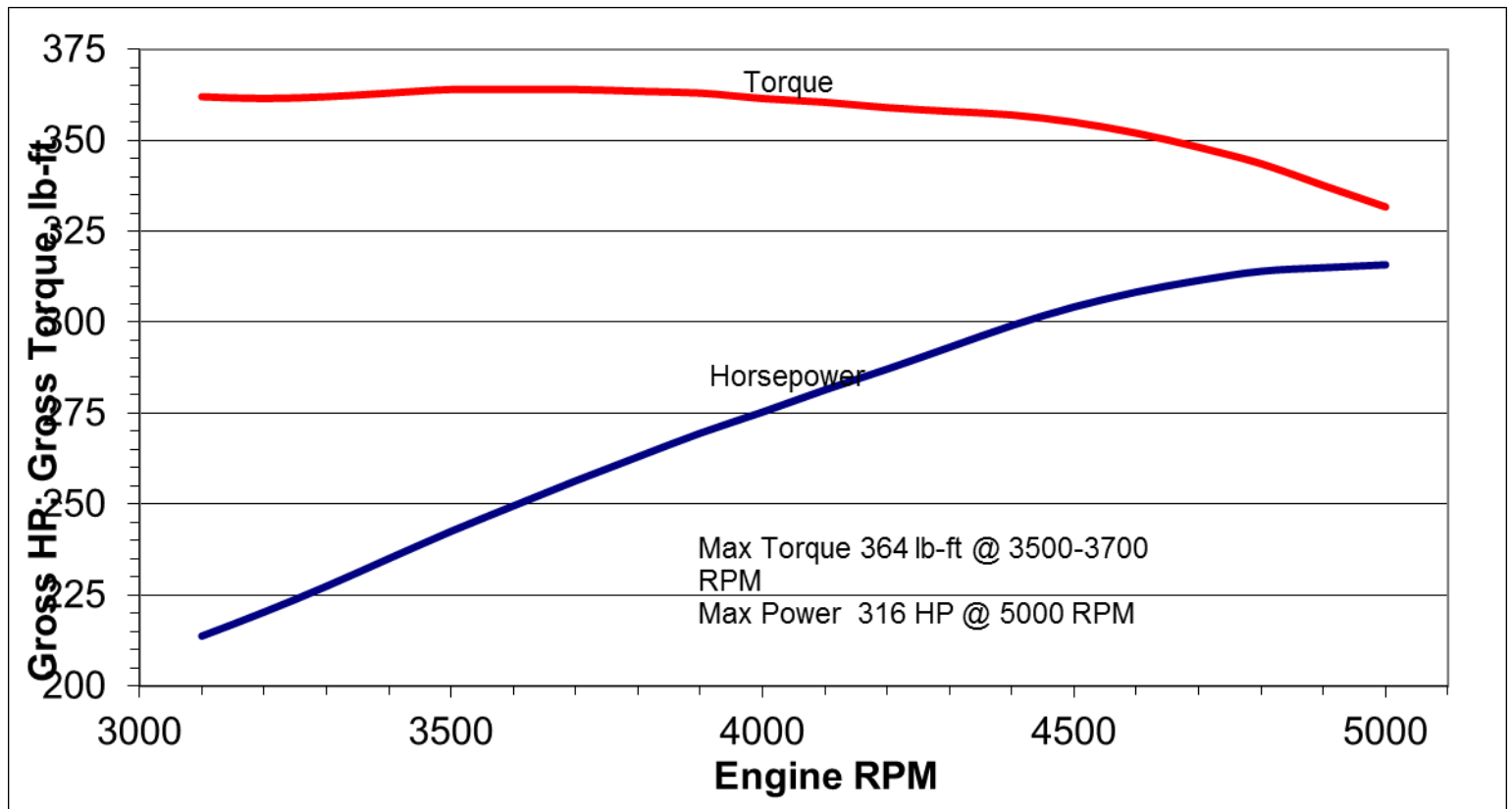
Within a few days I finalized valve timing for Mike's engine configuration (Table 2) to maximize torque bandwidth with 90 percent at 2000 and forwarded a new RFQ to Mike. The camshaft was ground by Crane in short order, and the delivered camshaft was of the same overall quality as the two Special 300-Horsepower camshafts. I named this design the *McCagh Special* camshaft.

## Testing

I wrote a set of test guidelines that included both road and chassis dynamometer testing. The first big test was idle behavior, and the Special 300-Horsepower camshaft in Dennis' '62 was indistinguishable from OE - butter smooth at 500 with about 18" Hg. manifold vacuum. The next phase was optimizing the spark advance map and increasing revolutions in small increments until the engine laid down due to either choked flow or hydraulic lifter pump up. Dennis reported no detonation with 14 degrees initial spark advance (38 total at wide open throttle with the 24 degrees centrifugal), and he said it was still pulling at 6000, which is as high as he wanted to rev it. In fact, he reported that it was the "strongest 327" he had ever experienced, which was an encouraging sign from someone who has owned numerous '62s with Fuel-Injection engines.

Not long after, Mike's engine was assembled and tested on a laboratory dynamometer, and the SAE gross data was very close to Engine Analyzer's predictions. Unfortunately, the tests started at 3000 revs and only went to 5000, so the bottom-end torque was not verified, but extrapolating the torque data indicates that it likely easily achieves the 90 percent at 2000 specification. Note how flat the torque curve is with a respectable 364 pound-feet peak (Figure 1). It doesn't get much better than this and is *perfect* for Powerglide! The 316 peak SAE gross horsepower at 5000 (and still climbing slightly) exceeds the advertised gross rating of the 1961 Duntov cam, 461X head, Fuel-Injection engine, and the useable power curve may extend as high as 6000, but I don't think Mike intends to rev it that high.

**Figure 1 Mike McCagh's 1957 RPO 579C 283/250-Horsepower Fuel-Injection with Powerglide Engine**



SAE gross on a laboratory dynamometer; 3.935-inch bore and 3.48-inch stroke for 339 cubic inches; correct 548 block and 997 heads with 1.84-inch inlet valves; custom pistons, 10:1 compression ratio; McCagh Special camshaft. This engine idles butter smooth and steady at 450 RPM in Drive with the original problem prone 4360 Fuel-Injection unit, passed the NCRS Performance Verification Test in January, and earned the coveted Duntov Award at the 2010 National Convention! Note that this engine achieves the OE gross horsepower rating (250 @ 5000) at only 3600 revs and breaks through 300 gross horsepower at just over 4400. No doubt, this '57 must be fun to drive!

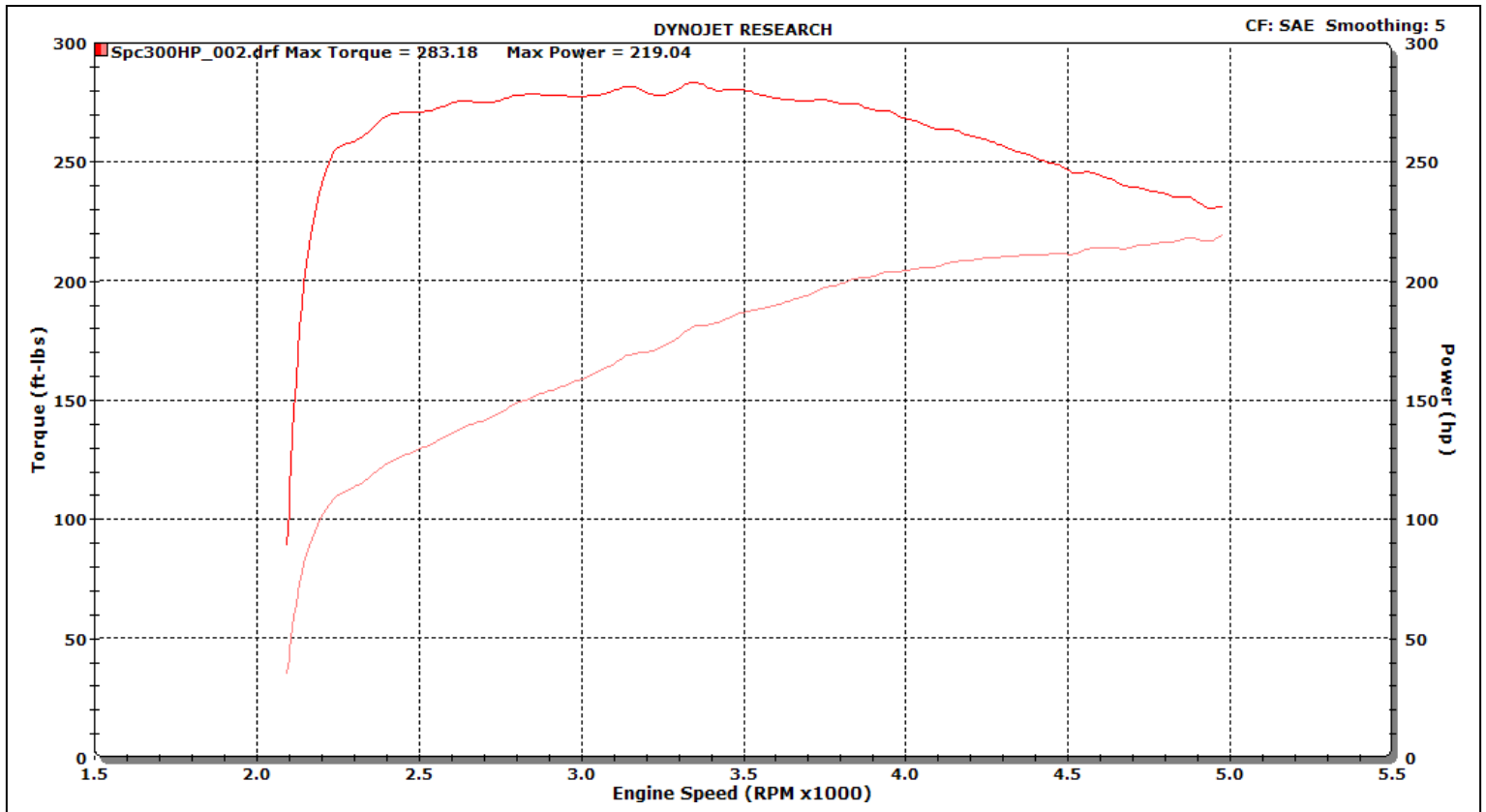
Dennis was next with a chassis dynamometer test, but unfortunately it was done on a Mustang chassis dynamometer, which typically read 5-15 percent lower than the more common Dynojet chassis dynamometers, and, again, the test was only from 3000 to 5000 so the lower end of the 80 percent torque bandwidth and full peak power range were indeterminate. The test was inconclusive.

Meanwhile, John's project was taking more time. Given other demands on his time and our mutual obsession with the smallest details, it was another year before testing could begin. Actual testing was done in a '69 model with a 3.55 axle that was originally a big block, but it has a 2.5-inch replacement exhaust system installed (All '69s for some obscure reason have 2-inch pipes.), so the results would be the same in the '67.

Road testing showed that this 10.4:1 true compression ratio engine could tolerate the most aggressive centrifugal spark advance curve achievable with available spring selection. It showed some lean surge at cruise – probably from a lower average venturi depression due to greater low load reversion from the late closing inlet valve - which was eliminated with one size larger primary jets in the OE Holley; and John revved it to a lofty 6800 before it laid down, which was probably the start of hydraulic lifter pump up. A friend of John's who has owned at least fifty Corvettes went for a ride and remarked, "That's no 327/300!"

Testing on a Dynojet chassis dynamometer, again, failed to capture both the bottom end and extreme top end performance; however, extrapolating the low-end data indicates the 80-percent peak torque at 2000 specification was well exceeded. The test plan did yield some very useful information. Only the best run is shown (Figure 2).

**Figure 2 Dyno Results For John McRae's 1967 327/300-Horsepower Engine**



SAE corrected on a Dynojet chassis dynamometer; .020-inch overbore numbers matching 870 block, massaged original 462 heads, original manifolds and carburetor; 10.4:1 true, measured average compression ratio with 0.08 maximum spread between cylinders; Crower Sportsman connecting rods; Special 300-Horsepower camshaft; centrifugal spark advance curve: 0 @ 900, 11 @ 1500, 25 @ 2500, 28 @ 3500 (maximum) with 10 degrees initial timing. This engine idles nice and smooth at 500 in neutral, as a 327/300 should. As built by Flint, 327/300s typically show low 190s SAE corrected rear wheel horsepower in the 4000-4500 range and are wheezing at 5000. This one pulled to 6800 on the road before it laid down.

The multiple tests led to the following conclusions.

1. Best power was achieved with 38 degrees total wide open throttle (WOT) spark advance—the sum of initial and full centrifugal advance. Backing the initial timing down two degrees for 36 degrees total WOT spark advance showed about one percent less power, particularly in the upper rev range.
2. Completely removing the air cleaner assembly reduced power about one percent across the entire rev range. The element and base act as a nozzle to smoothly accelerate airflow into the carburetor air horn with minimum turbulence, and removing the air cleaner assembly effectively increases carburetor restriction due to increased turbulence at the air horn. The '67 air cleaner has a three-inch tall element, which is virtually no restriction for a small block. The '62-'65 air cleaners have shorter elements that might reduce power, slightly. Without back-to-back testing, it's impossible to quantify, but I doubt if it is more than one to two percent.

3. Using a test to simulate cruise conditions, the cruise air-fuel ratio was determined to be in the range of 14.0-14.5:1, and we know that this is as lean as the engine can tolerate since one size larger primary jets were required to eliminate lean surge early on. This is typical of carburetor/manifold designs. Due to uneven fuel distribution the cruise mixture must be rich enough to prevent misfire on the leanest cylinders. Fuel-Injection engines can tolerate a cruise air-fuel ratio in the 15.5:1 range because the fuel distribution is even. As a result, Fuel-Injection engines with the same gearing as their carbureted Special High Performance siblings will achieve about ten percent better overall fuel economy. Modern EFI engines could also run this lean, but they idle and cruise at stoichiometric air-fuel ratio (14.7:1) because this generates the optimum exhaust gas constituency for maximum oxidation and reduction efficiency in modern three-way catalysts.
4. Pulls in third gear showed about 3.5 percent lower output than fourth gear, which jibes with typical transmission efficiencies. In direct drive when power is transmitted straight through the main shaft, typical front engine/rear drive manual transmissions, as used in vintage Corvettes, achieve efficiencies of 98-99 percent. When power is transmitted through the countershaft, as is the case in the other forward gears, efficiency typically drops to 94-96 percent.

## Conclusions

In past projects, Engine Analyzer 3.0 has proved to be quite accurate in peak power prediction with less rapid actual power roll-off beyond the peak in dyno testing than predicted, but EA 3.0 is somewhat high on predicted peak torque and up to 20 percent low on predicted low end torque. The McCagh Special camshaft's measured 316 SAE gross horsepower was less than one percent below prediction. Predicted torque was about 5 percent higher than the 364 pound-feet measured, which is typical, and though low-end torque was not tested, I have no doubt that it is considerably higher than predicted. Typically, actual measured torque curves are flatter than predicted with a higher average torque throughout the useable rev range, which is a good thing for a road engine.

The Special 300 Horsepower camshaft provides the low-end torque of an OE 327/300 and the top end power and rev range of the L-79, the best of both worlds, but actual peak power fell shorter of prediction than previous projects. Both owners were happy with the seat-of-the-pants feel of their engines, but I was perplexed. The McRae exhaust port flow was less than expected, which may be a factor, but in viewing the valve events (Table 2), I wondered if the relatively early exhaust valve closing of the Special 300 Horsepower camshaft compared to the L-79 and McCagh Special camshafts was an issue.

So I did an isentropic flow analysis of the exhaust stroke. Using Excel to set up the formulas, inputs include valve lift every two crankshaft degrees (from the engineering drawings), valve sizes, and engine revs. (This task is about 10 percent engineering and 90 percent data entry!) The calculation yields cylinder pressure every two degrees, flow velocity and Mach number past the valves, and total isentropic exhaust stroke parasitic pumping power. Given real world flow coefficients, actual pumping power is about three times what is calculated, but the relative comparisons remain valid, and the Special 300-Horsepower camshaft requires about 25 percent more exhaust stroke pumping power at 6000 revs than the L79 camshaft, which given a port flow coefficient of 0.6 adds up to about 15 additional horsepower pumping loss.

The details are buried in the primordial muck of the physics going on inside an internal combustion engine, but I believe most can visualize the process. Late in the power stroke, the exhaust valve begins to open. Since valve velocity is relatively slow compared to piston velocity, valve opening must be initiated well before BDC to ensure that blowdown occurs by BDC so cylinder pressure is at or near exhaust system pressure. As the piston begins its upstroke, very slowly at first, blowdown completes and valve opening area reaches a maximum somewhat before midpoint in the stroke, so even though piston velocity is near peak, the piston easily pushes exhaust gas out the large opening.

But things begin to change as the piston passes halfway. The exhaust valve is now closing, and as the piston approaches TDC, valve opening area rapidly decreases. At high revs cylinder pressure increases late in the exhaust stroke, which increases parasitic pumping power, and it gets worse as EVC is advanced. Further, as the inlet valve begins to open before TDC, high cylinder pressure forces exhaust gas into the inlet port, which dilutes the fresh charge and decreases volumetric efficiency on the next cycle.

Overlap is a primary factor in idle behavior, which is why these camshaft designs had to maintain the same effective overlap as the OE 3896929. However, idle behavior is insensitive, within reason, to overlap phasing, which gives the designer considerable latitude in optimizing valve timing to achieve excellent low-end torque and high top end power with a low overlap camshaft.

*Split overlap* – the point relative to TDC where both valves are open the same distance is one of many ways to compare different camshaft designs. Most vintage OE camshafts' split overlap occurs at about 4 degrees BTC. Looking at the camshaft data in Table 2, you can see that the Special 300-Horsepower camshaft's split overlap is a very early 8 degrees BTC, despite the relatively late phased inlet valve event. Conversely, the McCagh Special camshaft's split overlap is 4 degrees ATC (same as the 2001 LS6 camshaft), *12 degrees later*, so high rev exhaust pumping power and exhaust gas dilution of the fresh charge are less despite the shorter exhaust duration.

Comparing the inlet event specifications in Table 2 shows that the McCagh Special and 2001 LS6 inlet valve duration and indexing are nearly identical. Coincidence? Not really. I didn't make this comparison until the design was finished, and, clearly, one insight gained from this project is that if heads have high flow efficiency, a relatively short duration, but late phased inlet valve event provides the greatest torque bandwidth over the full useable rev range with manifolds and mufflers of low to modest backpressure, which is ideal for a road engine. GM Powertrain figured this out before I did and incorporated this insight into the design of LS-series camshafts, but they have more resources than I do.

Once an ideal inlet event is determined, the exhaust event can be worked out based on the E/I head flow characteristics and may require an event of equal, more, or less duration than the inlet event depending on the E/I head flow ratio.

My overall conclusion is that the McCagh Special camshaft provides the best torque bandwidth from off-idle to design speed (redline). This can also be expressed as the greatest average torque and power over the useable rev range. So the McCagh Special camshaft is my recommendation to anyone interested in rebuilding their engine to Special 300 Horsepower specification, and this can include any 283 with the base engine camshaft, all 327/250/300s, or any base 350 CID engine up to 1974, which was the last year of the full dual exhaust system. Long stroke small blocks will see similar benefits and consideration can be given to retarding the McCagh Special camshaft four degrees from design indexing with manual transmissions.

A couple of more projects are in work, but have not been completed and tested.

Dynos rarely lie!

About the author: Duke Williams is a retired automotive/aerospace engineer. He holds a Master of Science degree in Mechanical Engineering from the University of Wisconsin Engine Research Center and is the original owner of a 1963 340-horsepower Coupe. He resides in Redondo Beach, California and is a member of the Southern California Chapter. He may be contacted via the "send email" function on his profile page at the NCRS Technical Discussion Board on the Web.