Labels on the diagram:
SET SCREW — ADJUSTING NUT "ADJUSTER"
VALVE LOCKS (KEEPERS) — ROCKER ARM
VALVE SPRING RETAINER
VALVE SPRING — ROCKER MOUNTING STUD
VALVE SPRING SEAT
VALVE SEAT (Head) — PUSH—ROD
VALVE TRAIN — LIFTER — CAM
Jim Miller

# Rocker Geometry

BY **JIM MILLER**

ROCKER ARM GEOMETRY seems to raise its head every now and then, and when it does, I rarely ever see it stated accurately. Too often a sound bite of only a small piece of information is taken out of context and then used as the Gospel, totally ignoring the other dynamics that revolve around it. In some cases, something totally erroneous is stated that is not only wrong, but makes no sense for anyone who just stops and thinks about it objectively.

When lecturing at trade shows, schools, engine shops or just getting pinned down on the phone by a knowledgeable engine builder going deeper than most on a technical issue, I have found that I spend about half my time trying to undo various misconceptions about rocker geometry before I ever begin explaining the facts. There has been so much info put out there by reputable companies (and by my reckoning, incorrect), that people are reluctant, by nature, to see something different from the prejudice of what they already know or think they know. If people are used to doing something a certain way, they see everything from that perspective. Usually, my getting through to them involves discrediting what I think

is wrong with what they were doing and then begin to explain what they needed to change. At that point I could break down the simple rules for what geometry really is, and why.

## Background of Rocker Geometry

Rocker geometry (or the lack of it), goes way back to many fathers on both sides of the ocean, to when the Wright Brothers were still studying the theories of lift in an airfoil. But for our purposes here, and to avoid boring the curious who've managed to get this far on this story, I'll come to the point about rocker arms and explain as needed how the "mistakes" got to where they did.

In the old days, rocker arms were all pretty much what we term a "shoe" design; meaning the contact pad with the valve had a large radius scruff surface that depressed upon the valve tip as the rocker moved through its rotation. The term of course comes from the appearance to a shoe's sole, but also to the mechanical motion much like a foot and boot would do, as it pushes off. This pushing off motion, as many will already know, has the effect of the rocker arm stretching itself as it moves through the depressing (lift) cycle. It is actually

lengthening itself as it moves across the valve tip, and you see this by the wide foot print (we call a "witness mark") atop the valve tip.

The use of rocker arms goes back to many things predating engines, but the principles were never required to be so specific on axis point heights and their consequences, as it is for helicopter bell cranks, and racing engines! There was no rocket science to designing these parts a hundred years ago, which ended up on our prehistoric cars and early airplanes. Engineers simply made designs that tried to minimize the degree of how much scuffing was imposed on the valve tip; got it close, and moved on to more important questions. Somewhere along the line, there became a principle to get this in a general ballpark, that someone later coined as the 1/3-2/3 theory (or either of the two). This placed the pivot point of the rocker arm so that it was 2/3 of the way below the valve tip, or the valve tip was 1/3 of the way above the rocker shaft, depending on your point of view. But the answer was the same. This thinking was originally derived from the intention that a near 90 degree arc could be realized when the valve reached its intended full open position. Bear in mind

that valve lifts back then were usually in the quarter inch or so range, on little two and four cylinder engines. So being off a little really had no measurable difference in performance of the engine, and wear and tear was the real yardstick engineering back at the turn of the century was aimed at. Also, the ability to accurately measure wear and tear, horsepower, thermal loss and many other cool things we take for granted on today's computers, wasn't even a possibility back then.

The advent of more valve lift, and thus pushing a budding internal combustion engine technology higher to produce more power was really inspired for leaps and bounds by the advent of aviation, not Henry Ford. Not to take anything away from the automotive crowd's contributions, but only aviation imposed the second requirement that defined "efficiency" – and that was light weight. Making Goliath engines that had more power was a lot easier than making more power from light weight engines that would be flying over somebody's head, somewhere. So the whole "thinking process" for efficiency in engine technology really found its impetus in aviation, because racing back in the early 1900s was still done on the back (or behind) of one-horsepower whose exhaust was more easily stepped in than emitted from a pipe. And as far as my 30 year old memory serves me on the research, aviation was also the first use of a "roller tip" rocker arm, on radial engines as far back as the 1930s, and perhaps before. In fact, to this day, I never cease to be amazed at the foresight and creativity of both aviation and automotive engineers in the 1920s, and '30s, and '40s. Four valve Pent roof combustion chambers, roller cams, fuel injection, nitrous oxide, water injection, two stage superchargers, turbo chargers, and many other cool things we assume were concepts of the last 20 or 30 years, were actually done and done quite well, seventy and eighty years ago.

Aside from the roller tip rockers of aviation long ago, the fundamental rules of rocker arm design were based around the shoe tip, contact pad design still used today. Many of you may know that you can't use a roller tappet on a flat tappet cam, and of course vice verse not only because of hardness of material difference, but because of "geometry." The principles of trailing motion and dynamics between something that is making direct LINEAR contact upon another object that is imposing or receiving a RADIAL (circular) is entirely different than if that contact is occurring with a roller tip making or following the contact. This isn't rocket science either, and you can see how this happens by

drawing a roller tappet in various stages of lift as the cam lobe goes around to push upon it, and see a straight line from its axis to the cam lobe is constantly shifting around as the tappet goes from the close (base circle) position, up along the acceleration ramps, then over the nose. When finally, as it crosses dead center at full LOBE lift, this straight line between its axis and the cam centerline is also in alignment with the tappet bore itself. At all other times, the tappet is actually receiving some level of side thrust in its bore (engine block) from the pushing out forces that the lobe imposes as it chases it up.

Even when engineers were chasing efficiency with aircraft engine and power development, the threshold for seeing measurable loss of engine life, like valve stem or valve guide wear, was not easy unless things were really out of whack. They weren't worried about loss of cam events through small changes in rocker geometry, even though they knew the variables existed. So if they kept to this 1/3-2/3 rule, everything looked good on the valve tip, and the leverage of the rocker arm upon the spring – or more accurately stated – the leverage of the spring on the rocker arm was at near perpendicular relationship with the valve when rates were at their highest. Even though by today's standards for valve springs, spring rates back in the 1920s and '30s were negligible. This "attitude" continued on throughout the decades afterwards. More importantly, and unfortunately, it bled over into the soon to come roller rocker arm market, that got its impetus in the 1950s. The first person who made a working aluminum "roller tip" rocker arm for automotive application belongs to my dear old and departed friend, the late Harland. Some other garage efforts might have been getting tinkered with out in California about the same time, but it is pretty well undisputed that 1958 is the beginning of what we know today as "the aluminum roller rocker." Keep in mind that aviation roller rockers existed twenty or more years before, but they were steel, they were radial engines, and they didn't comport to the automotive needle bearing aluminum body that Harland introduced.

Just like the flat tappet cam and the roller tappet having entirely different geometry because of where the measurement for motion is made, the same rules apply to the shoe tip rocker versus the roller tip rocker arm. But when Harland made his silhouette, he didn't allow for this, and inadvertently moved the axis of the roller roughly .300" of an inch higher than what it should have been. The axis of the roller should have been in the same place as the contact

pad. So when his rocker was placed on the engine, and the roller was positioned for a good "eye-ball" track on the valve, now the push-rod cup was too high. The result, was that it went way up and in toward the stud. In actuality though, most engine builders in the sixties continued to keep using standard length pushrods, and the excessive motion from this mistake was occurring on the valve tip, which was deemed "normal" because the roller rolled! Believe it or not, even to this day, people think that the roller tip is for rolling on the valve. It is NOT. The roller tip is for one reason only, and that is to convert the shifting length of the rocker's arc (that moves across the valve on a shoe design), to a fixed length that moves far less in its effect, because it is always point down in line with the valve's motion, just as a roller tappet of a cam is always aiming its contact tangent line with the axis of the camshaft.

This error stayed, and was copied by many manufacturers and eventually by everyone in some measure or another. It would take several decades before enough trial and error, and even a patent would be studied to make manufacturers rethink this, slowly improving their designs. Ironically, some of the most well known names continue to promote designs they never changed, and even promote the less accurate means of using little tools, that tell an engine builder what pushrod they need, while never even taking into account the valve lift that will be used. Make no mistake; you cannot set rocker geometry without knowing exactly how much the rocker arm is going to move.

It seems logical that since a roller cam can provide all the acceleration any of today's heads need, for any rocker geometry scenario, then why not set the rocker geometry to ONE STANDARD that has the least amount of wasted motion, and will always duplicate the same percentage of cam information, regardless of what cam you use? For understanding this, understanding a little history is always best. This ends a lot of rhetoric.

Whenever pushrods leave their in-line paths to now have their end follow around the rocker shaft by any amount, this is LOST CAM INFORMATION. The cam literally has to turn more degrees to affect the same LIFT at a later point on the crank. Velocity, too, is lost. So you've lost duration, throughout the entire lift cycle (not just overall), and you've lost RATE of acceleration, by slowing the rocker down.

To put this in perspective, let's take a simple even value of cam lift, like .400" to make a point. Rocker geometry is usually thought about as only what is happening at the valve. In our .400"

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cam lift example, and a 1.50:1 ratio rocker arm, this would theoretically yield .600". To have the optimum use of in-line motion being converted into circular motion, you need to divide these values into two equal parts. Engine builders do this at the valve, but the real deal is happening with the CAM. The cam is the source of the information. So for the CAM side of this value, we're only ending up with .200", which is one half of our .400" cam lift. If you fail to place the axis of the tip of the pushrod at the proper length as to divide that cam lift accurately, so the rocker arm is at a 90 degree angle, then you have a pushrod that is going to move in and out more than it needs. The result will be wasted cam information that can require the crank to move several degrees more to effect the same lift of the valve. Those lost degrees were absorbed in the excessive motion the rocker arm had. You will spend hours and hundreds of dollars to get a camshaft that is ground to fourth decimal accuracy, and chosen to give you a specific degrees of duration at .050" tappet lift, and you will change

a cam to gain as little as four or five degrees if you think the engine needs it, but you just threw away more than that because of the PIVOT POINT on your rocker arm didn't establish the correct angles with the pushrod and valve.

## Why did this continue?

Back in the early 1960s, because there was so much inefficiency and experimenting with cylinder heads, cams, induction systems and so on, this valve train flaw went under the radar. Now I was still a punk, barley sixteen in 1969, but if memory serves me right, it wasn't until a real student of engineering and racing stuck his nose into the situation, and started shifting the pivot point around for his own purposes and seeing distinct changes. He had some odd name car in a new class of drag racing, called Pro Stock; I think it was "Grumpy's Toy." Bill Jenkins was one of the real pioneers for many things in not just fixes to problems, but also a more scientific Yunick-like approach to analyzing. He didn't follow other people; other people went out of their way to follow

him. It was a short list of real pioneers to both cylinder head and valve train development back then, and Bill was on the short-short list. But prior to Bill and a few others like him of that era, rocker geometry was totally ignored beyond the vague generalities of the 1/3 rule. But technology in the cams and heads was soon catching up. Right about this time, in 1969/70, Chrysler approached Crane Cams for a new camshaft for the factory backed Hemi teams of Sox & Martin, Herb McCandless and the "Motown Missile" (later Mopar Missile). That development was the beginning of the .700"-plus valve lift boundary being broken.

The late sixties and early seventies were really exciting times for factory muscle cars, and the stepping stones of technology that has brought us to where we are today. It all began back in this limited, golden era. And the fundamentals established then, cut in stone, have not changed to this day either. They've only gotten repackaged, renamed and resold, even though other boundaries in valve lift, cylinder heads
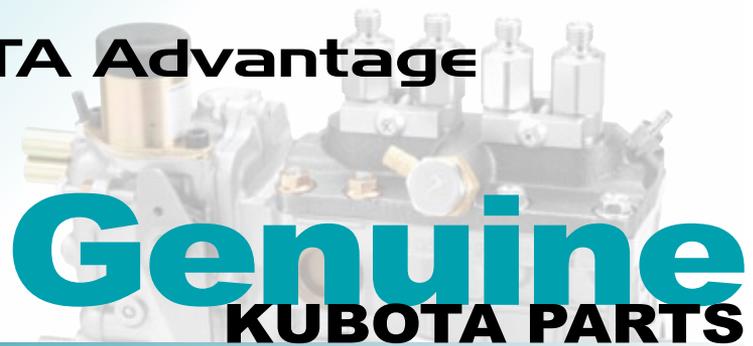
and so forth have been elevated. The principles for cam technology and specifically rocker arm geometry that would soon come along in 1980, but spawned in 1973, have not changed to this day.

## Definition

What is rocker geometry? Rocker geometry is "angles of motion." It is not some linear reference point on the tip of the valve, that trying to adjust the wear pattern will guarantee it being correct. What is correct? Correct, is "efficiency." It is having the least amount of wasted motion being used to do the greatest amount of work (that is designed to be done by the cam). This last point is important, because the rocker arm can be used to add to the cam, besides what it usually does by error, which is take away from the cam. But I will get into each of these below. I just wanted a simple "mission statement" that defines what geometry is and is not, so that the following hopefully makes sense.

## Importance

WHY is rocker geometry so important? When you change the pivot point of where the rocker arm is, in relation to the valve tip, it CHANGES THE CAM. It doesn't matter whose rockers you use, it doesn't matter what style rocker you have, it doesn't even matter what application your engine is; whenever you change location of the rocker pivot point in relation to the tip of the valve, you are changing cams. You are changing all three parameters simultaneously: LIFT, DURATION and VELOCITY (rate of acceleration).

The degree you change these depends on how much you move the pivot point. And one or two of these three parameters may be affected more than the other. But if you don't LOCK your geometry in to the SAME THING all the time, which has the least amount of wasted motion, then you are aiming at a moving target with every cam change. Whatever results you get from one cam to another is tainted by the diluted effects of wasted motion in the rocker arm.

Rocker arms are a "radial" device being ordered to do a "linear" thing. They rotate on an axis, moving in a circle. But what they have to impart is a straight line command. They get their order from the camshaft, in the form of IN-LINE information that they then have to ROTATE around an axis and then MULTIPLY it by some ratio, and finally TRANSFER this result back to another IN-LINE component of greater movement. This movement has THREE values: LIFT, DURATION (of lift), and VELOCITY (acceleration of lift). If the rocker arm does ANYTHING ELSE besides this, then it is NOT efficient, and SOME of this information is being lost.

Let me make a point about something on this. Your camshaft is ground to ten-thousandths of an inch precision. It is computer designed to millionths of an inch, you (or your cam manufacturer) selected it for a division of duration values where you considered two or three degrees important; any more was too big, and any less was too small. Hopefully by now, you realize that moving this rocker pivot point will change this at the valve.

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You just don't know how much. Well, it is MORE than the two or three degrees you think is important. In some cases it can exceed TEN degrees, and is often five or six degrees. As if this isn't enough reason, consider this: It is that value of loss throughout most of the lift cycle, not just total – where you're only inclined to measure it from, and where your cam card is limited to. That is what's missing. Engine builders only check at FULL LIFT. They check rocker ratio and total valve lift, and that's that. But when your rocker geometry is off, you've lost those degrees of duration throughout most of the entire cam profile. Which means rate of acceleration is lost, but you may only see a small change in lost valve lift, thinking the difference is just flex in the rocker ratio.

## Two Geometries
Rocker Geometry is the correct DESIGN and INSTALLATION of the rocker arm so that its relationship to both sides of up-and-down motion is fully realized by BOTH. This is of course, the pushrod, and the valve (respectively).

The rocker arm is a RADIAL tool, asked to do a LINEAR job. It pivots around an axis in a reciprocating radial (circular) motion, and has all the dynamics that anything revolving around an axis will have. But on either end of the rocker arm's connecting points are two other instruments each live and breathe by the laws of linear (in-line) motion.

Now all this may sound like "rocker arms 101" and we may all know this, but few people I have found over 35 years, seem to understand how sensitive, precise and important this observation is. I think this true because most treat both the design of the rocker arm and its installation with casual regard.

To have the "most efficient" design and use of the rocker arm requires TWO things: (1) The rocker arm must be designed to mirror the inherent angles of each engine's pushrod to valve geometry. (2) Rocker geometry must have an accurate location of its rotational axis with the valve tip's height. The first of these is called DESIGN geometry; and the second is called INSTALLED geometry.

Every engine has an inherent acute angle (I call the "attack angle") which the pushrod leans either into or away from the valve centerline. The small block Chevy for instance is 19 degrees positive (leans into). This comes from the engine block having a 4 degree angle of its tappet bores with the piston cylinder centerline. The head is already a 23 degree valve angle (actually it is a

67 degree, because it references from the deck), so you simply subtract the 4 degrees that the tappet and pushrod "aim" toward an already inclined 23 degree valve, and you end up with simple math: 19 degrees. Every engine is unique. The SB Ford is 20 degrees for this same value. This is what rocker geometry needs to be designed to, otherwise your efforts of installing the rocker arm accurately will be limited to just one side of the rocker or the other.

## What Rocker Geometry IS NOT
Rocker geometry is NOT where the roller (or contact point) is at on the top of the valve. Forget that. Rocker geometry, especially is NOT the idea that you want to place the roller or wear pattern (shoe tip rocker) in the "middle of the valve." Forget that, too.

The valve tip is everything. It is ground zero. This is where all leverage, and the full stroke of valve lift begins, this is our reference point. There are many ways to measure rocker geometry, but there is ONLY ONE way to SET IT. Now, you can set rocker geometry in the closed position, or you can set it in the MID-LIFT position (half open), or you can do like most people have done, and simply roll the engine over a couple dozen times watching the valve open and close to see your witness mark (foot print) atop the valve tip and play hit and miss with chasing a "minimal wear pattern."

The problem with this latter point is that this is a symptom, it is not geometry. Granted, when you have the rocker geometry set properly, you will have the least amount of wear pattern, but to try to set geometry through moving the rocker and trying to see how small you can get it, is better than nothing – not quite good enough. You can easily be off by .005" to .010" (or a great deal more) on even seeing this actual width, let alone measuring it. And being off .010" on the horizontal plane of what you are really trying to measure, which is the vertical plane (valve lift), will multiply out quite easily to .030" or .050" or .080" or more in your error of where the trunnion is to the valve tip! Those kinds of errors will cost you several degrees of crank rotation to open the valve a like amount.

What about using a tool, or dial indicator designed to measure this in and out motion, resting on the spring retainer? Well, this is a better way of the same thing, but it is still measuring a horizontal plane for a vertical plane result. Error can be off several times more than measuring directly in the vertical plane, or parallel to the valve

motion itself. When using these tools, just like a dial indicator on the top of a piston, as soon as the piston reaches perfect TDC, you will have two or three degrees of crank movement before you see the dial indicator move. There is a float time there, and so too is the effect by using a tool on the roller tip of a rocker to measure in and out motion. It floats enough to allow the valve lift to be off by .005" to .010" or more. But if you can set it dead nuts within .002" to .003" without having to buy such a tool, why wouldn't you do it in a more precise way?

## Alternative Geometries
Before getting too deep into philosophies, history and facts, let me restate the key point of what rocker geometry is, then I will mention the comparative arguments people (and companies) have made against this. MID-LIFT geometry is rocker geometry that has the ultimate "efficiency," in that it is doing the greatest amount of work with the least amount of effort. It has the least amount of wasted motion in the pushrod and valve, commonly referred to as the "in-and-out" motion. It affects the maximum response through linkage of whatever the cam's instructions are. If you don't have geometry set precisely, your consequences range from simply losing a little cam information at the valve, to excessive side loads in various directions, on various parts that will at the very least rob you of power. In more extreme cases, wearing out parts or outright catastrophic part failure may occur.

There are arguments to using different geometry than mid-lift. I simply don't agree with them because they violate the principles of efficiency. One of these theories is to adjust the rocker arm's height so it reaches a 90 degree relationship when the valve is about 2/3 open, not half open. The logic being, that the spring loads on the rocker body are less. Another reason I've heard is that it accelerates the valve in the "mid-range" better, thus making more power. Other variations of this approach shift the rocker arm's pivot point higher on the valve tip to create this 90 degree effect sooner in accelerating the opening of the valve at a lower point of lift, thus increasing what is termed "area-under-the-curve."

Both of these are a way to add different cam information to the valve by using the rocker arm's geometry. The only reason you would use the rocker arm for creating a "second dynamic" of valve acceleration, is if the cam was unable to give you the acceleration

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you needed. Now, in some cases this limitation exists. They would be flat tappet cams, of mechanical or hydraulic operation, and an engine where cylinder heads had the flow potential, and/or cubic inches had the demand which required a crazy acceleration to mid-lift flow values off the seat. In other words, the engine was so big, and the heads were so big but for rules or some other illogical reason, the cam they HAD TO USE was a flat tappet profile that had limited "rate of acceleration" by its limited tappet diameter and base circle constraints. I won't get into cam technology and limits, but that is the first reason I can think of for using the rocker arm as its own cam tuner. In some Stock classes where the original cam must be used, a creative (and well funded) engine builder can play games with rocker geometry to change valve lift rates, but these are very limited differences, usually not worth the trouble, and most of all, in ALL these examples, there is going to be detriments that outweigh benefits.

In the first place, for those who have an engine of large flowing heads, and big cubic inches, or heads for very high rpm's, they will have the benefit of using a roller cam. So the issue of how fast you can open the valve is not even a consideration, because by nature of roller tappet geometry, any value of acceleration up to and through suicidal parts destruction can be implemented on the cam profile. And for those classes where a flat tappet cam is required, the cubic inches and head limitations of most rules I've seen over the years, fall within airflow and rpm limits that a flat tappet cam fit just fine. Too many times, cam companies talk customers into roller profiles that are not needed, and in fact don't make as much power as a well chosen flat tappet would, because it takes more power to operate the roller. Using rocker geometry as a second cam shaft is not a good idea. The velocity of the rocker arm increases where its motion line reaches a 90 degree angle, and trying to pick a particular segment of the valve lift that you want to impose that thinking over what the cam manufacturer has done, is bad news. But there's another point to consider on this issue.

The rocker arm is a symmetrical device to whatever geometry it is set at. In other words, whatever acceleration it exhibits on the opening side of the valve gets reversed on the closing side. Simply, if you set geometry with a HIGH pivot point, so it increases its velocity quickly off the valve seat, then slows to full lift... Guess what? It's going to accelerate back to the close position when it leaves

full lift. Because it will always mirror whatever its settings are.

To see the difference all you have to do is take an old fashioned needle pointer torque wrench and turn the engine over without spark plugs and measure the drag. Then set the geometry to MID-LIFT, and see the difference. I've had people do this and fall out of their seats. They see as much as 45 foot pounds or more and LESS torque to turn the engine over. Usually it is 15 or 20 foot pounds but it depends on how much spring pressure you have, how complex the rocker geometry is (Hemi versus SB Chevy, for instance), and how wrong their geometry really was. Either way, it quickly shows you the definition of efficiency. If they MAP their valve acceleration, throughout the entire lift curve, then I need say no more. The best way is to DIVIDE the arcs EQUALLY and standardize this for all cam and cylinder head testing. Change your cam as you need.

## Setting Geometry

As mentioned, the rocker geometry has TWO considerations: The accuracy of how it is installed, which I have always referred to as the "installed geometry." And second, how accurately it is designed, which I have always referred to as the "design geometry." This second item relates to where the adjusting screw or cup is placed in the body, and at what angle.

As an engine builder, you can move the rocker arm up and down to the valve tip in setting that side of the rocker's effect, but you can't do too much about the pushrod side, and that is where the information comes from. That is up to the rocker arm manufacturer.

Of the two sides, however, the lesser of evils to shoot for is setting the VALVE side (or installed geometry) as closely as possible, because (a) this is where the motion is constrained by the valve guide, (b) this is the side that has the greatest motion, (c) this is the side that has the valve spring, where harmonics are generated and amplified, (d) and this is the side where all of the foregoing multiply into a measurable resistance value that generates heat, robs power and creates additional friction. The pushrod is, by comparison, free floating with the rocker body's movement in and out as well as up and down, and it is moving less (the cam's lift). But make no mistake about it, when the pushrod side is not performing to mid-lift geometry, it is losing information. The upside though, is that whatever is left, is getting through to the valve. When the geometry is off on

BOTH sides of the rocker, because you didn't install it accurately, you lose twice! Only 90% may go into the rocker from the cam, and of that, only 90% comes out, or 81% goes to the valve. I'm of course rounding things off for example, but the principle is the same.

I should add one other point here. Everything is "net." So if you have those cute little "checker springs" laying around, find some other use for them, because outside of holding a valve in a head for display where someone can use their finger pressure to push the valve open, they have little use. You need the REAL running springs for any geometry setup. The same goes for checking flex or piston to valve clearance or anything else critical. Checking springs ADD about .040" (or more) NET valve lift to your engine. Or, another way to say it is you will LOSE .040" or more NET valve lift when you put the heads together with the running springs, compared to whatever you measured using the checking springs. This is true across the board, flat tappet cams, roller cams, aluminum rockers, steel rockers – it makes no appreciable difference.

Later in this article, I will offer installation and assembly tips but for now, here is the easiest of accurate ways to set INSTALLED rocker geometry:

The closed valve position is the easiest and the best. The cam must be in the closed position, on its base circle. Heads are assembled to the engine, with no pushrods in place. You must know what your "net" valve lift is supposed to be (we can get nitty-gritty later). You will subtract any valve lash so you have an accurate "net" lift.

For stud rockers, put it in place with an adjustable pushrod. You don't need the poly locks; just let it set loosely on the stud. Knowing your net valve lift, DIVIDE it in half, and write it down on a piece of paper. Then, lengthen or shorten the adjustable pushrod to raise and lower the back of the rocker until you get the center of the trunnion exactly HALF of your net valve lift BELOW the center of the ROLLER TIP. If for example, you have .600" net valve lift, then this would be .300". Keep in mind that I refer to "center" of the trunnion and roller pin. It is their axis that is what you are measuring. Some are easy to see and some are not. For those with flat machined surfaces, take a scribe, measure and mark these centers as best you can. But the main trick is that you want to be sure you measure this from a precise 90 degree reference to the valve centerline. To accomplish this, you are best served to use the top of the valve spring retainer.

Simply lay a short machinist ruler (or something comparable) atop the valve spring retainer, and pass it along the side of the rocker arm.

When you have the height installed accurately, the trunnion will be exactly HALF of your net valve lift, below the roller tip centerline when the valve is CLOSED. As it opens, and moves to exactly mid-lift, the axis of the roller tip will have dropped down to be straight across from the trunnion and an imaginary line that runs between them (I call the motion line) will form a precise right angle with the valve centerline. The roller is at its farthest point across the valve when this happens. When the valve continues to open the second half of its lift, to full lift, the roller will have moved exactly an equal amount BELOW the trunnion as it was above the trunnion when it was closed. And the roller will be at its closest inside point on top of the valve. You will also have the ultimate least amount of roll across the valve.

For shaft mount rockers, it's a little different, but the principle is the same. With shaft rockers you must use shims, or have a stand that has a surplus of metal that you are machining exactly what you need away. But you can take a measurement of the stand height without using a rocker arm. Just bolt the stand down to the head with a couple of bolts, lay the shaft in the stand's bed-way, and use a machinist square along the side of the valve (or spring) and shoot the long end along the top of the shaft, so there is a gap beneath it and the top of the valve. Everything is about finding the centerlines, and being creative about doing this, while being accurate at the same time and measuring at an accurate right angle with the valve.

It makes no difference where the wear pattern is at on the top of the valve, when you have correct mid-lift geometry, and providing the pattern is "on top of the valve." (Running off the edge of the valve is not good.)

## Graphing the Cam at the Valve

To see what is really happening at the valve, you need to check your rocker's motion by measuring it at the valve. The best way is to essentially treat this like you're degreeing your cam, but you're measuring motion at the valve. Only instead of just picking up points of lift to compare to the crank, as you would with the .050" tappet lift measurements on a cam card, you will be creating a graph all the way through the entire cycle of valve lift, opening and closing. If you are fortunate to have a CAM PRO or CAM DOCTOR, or something similar, life is good. If not, you can do the old fashioned way. You will need graph paper that can be found at art stores, engineering supplies and many science or school supply providers. You need a dial indicator that you'll mount directly above the valve spring retainer, nearly fully compressed so you follow the valve's stroke fully – and you want to be sure the indicator stem is lined up parallel with the valve. As with setting your cam, you need to have a degree wheel in place on the crank and zeroed accurately to TDC of the piston.

With the above in place and ready, you have TWO choices to how you measure this; which are merely opposite perspectives. You can choose an even

# ROCKER GEOMETRY

BY **JIM MILLER**

number of crank degrees you move through to measure valve lift, or you can choose an even number of valve lift to measure degrees of crank rotation. It doesn't really matter which you use, because it is the comparison against other like tests that is important, and both need to be the same. You don't have to be too crazy about fine increments here, just choose valve lift jumps of maybe .020" and write down the crank movement; or choose 5 or 10 degree crank measurements and write down the valve lift.

Personally, I like the second method of using a fixed crank stepping, and then noting the valve lift. This goes directly to the points I make about "area-under-the-curve" that you are looking for. For those new to this term, area-under-the-curve refers to the VALVE LIFT CURVE as charted across a graph of time (meaning crank degrees), and what most engine builders agree, is that lifting the valve as quickly as possible and as soon as possible, while setting it down quickly after it has hung open for as broad a period of time as needed, but without being too fast to damage the valve train from excessive "bounce" is what everyone wants. So, when you want to see the gains and losses in this area from inaccurate rocker geometry, you're really looking for wasted time when the crank is moving more than it needs to lift the valve the same amount. So, if you standardize your testing to the same valve lift measurements, the gains and losses in the crank are readily seen.

Once you've charted one rocker geometry setup, perhaps the one you've been using, now make the changes with pushrod length and/or stand height (for shaft systems) to meet with what I hope I have informed about earlier in this article. You will often see that PEAK valve lift is very close to the same, but much less at other points in the curve. That is the lost information. The degree of this will depend on many factors that take another story to itemize. But the bottom line is; you will appreciate how important it is keeping the same geometry, for making sure your cam changes show results that are directly accredited to them and only them. Otherwise, your information is tainted.

## Shoe Tip Rocker Geometry

As with aluminum rocker arms, there are different design geometry shoe tip rocker arms, but the priority for adjusting the valve tip side is still there for the same reasons. However, you don't easily have the same accuracy as you do with picking up precise center points on the roller

and trunnion of a needle bearing roller tip rocker arm. Even finding the axis is not easy. Setting it by the same rules is best simply for standardizing one cam to another. Only your reference is the actual contact point of the pad itself. When it is at mid-lift, you will have a 90 degree relationship between the TIP of the valve and the center of rotation for the rocker arm. But finding that center is tricky, because these are usually ball fulcrum rockers, and they are surrounded by the stamped metal that has no clear axis to it. One solution is to put a stud upside down in a vice and rotate it carefully while observing the fixed point on its side that most closely represents where the center is, then making a little felt tip pen mark. This would then be set exactly half of your net valve lift below the valve tip in the closed position. It's not as accurate as fixed points to set calipers against, but it will get you very close if you have patience and a sharp eye; and with shoe tip rockers, the amount of error you might be off will have no measurable effect in cam efficiency as it would with needle bearing roller tip rockers.

## Twisted Rockers

Unfortunately, engine builders are led into a false security by stud mount rockers sold for heads that shouldn't use them. These are aftermarket heads with pushrod offsets that require an offset pushrod cup, or adjusting screw. Shaft systems usually have this adequately fixed, but when heads are sold with studs in them, that clearly need to be removed for a shaft system, this is the false sense of security one gets. The two rockers shown here is exactly what you DON'T want to have (Figure 4).

When you have this much rotation, the roller tip does NOT lay flat on the valve tip, and as it opens the valve its energy is making a cross sword slash across the valve tip that rounds off the top of the valve tips, side loads the guide on the X axis (length of head), and shifts side loads to the bearings in a way that often push or break one prematurely. It's bad news, costs horsepower and breaks parts.

## Ratio & Geometry

If the rocker geometry is off, then so too is the ratio. There's some good news though: Don't worry about it, because very few of the rocker arm makers did. The history of rocker design didn't have much accuracy involved. There was no standard, because there was no need for such accuracy in the old days. Rockers always were, and to a great degree, still are, designed in the closed valve

position. But as long as you stay focused on what you need to do, you won't try juggling things around outside of the real priorities, based on a false idea of what YOUR rocker ratio ought to be. It can be all over the place from .05 less, up to maybe close to what it is supposed to be, simply because many manufacturers began in the closed position, and then started moving specs around from a hit and miss until it got close. Once it did, they left it there. Over the years, more consistent manufacturers with the least amount of broken parts have been the model other newer companies would copy. But the mistakes get copied, too. You have to always check things for yourself, forget the advertising. If you do, then you can't blame anyone but yourself.

## Installation, Measurement and Adjustment Tips

### Figure 1

Here's the typical stud mounted needle bearing roller tip rocker shown in its CLOSED position. Dimensions are not shown, because they are relative to your installation, which is relative to your NET valve lift (after lash, if used). The important dimension you want to find and set, to be HALF your net valve lift, is the ROCKER HEIGHT, shown to the upper right, and illustrating the cumulative value between the ROLLER axis and TRUNNION axis. The other references are shown with regard to their heights above and below (respectively) the valve spring RETAINER. All measurements for where the trunnion is sitting, is referenced to the top of the retainer, marked here by TRUNNION REFERENCE. Raising and lowering your rocker's tail by adjusting the Pushrod as needed, will set this. Although shown in this illustration, the adjuster doesn't even need to be here, as it will only get in the way. Let the rocker sit loosely on the study, with its adjustable pushrod and set this trunnion reference as needed to get the trunnion exactly HALF of your NET valve LIFT.

If using a HYDRAULIC TAPPET, be sure that it is fully extended during this check. You can prime the motor to do this. After getting your pushrod length, ADD .020" to allow for hydraulic tappet compression during actual engine operation. Order EXACTLY what you need for pushrod length, rounding off to the nearest ten-hundredth of an inch (two decimals).

ROLLER
HEIGHT

TIP
HEIGHT

ROCKER
HEIGHT

TRUNNION
REFERENCE

TRUNNION
AXIS
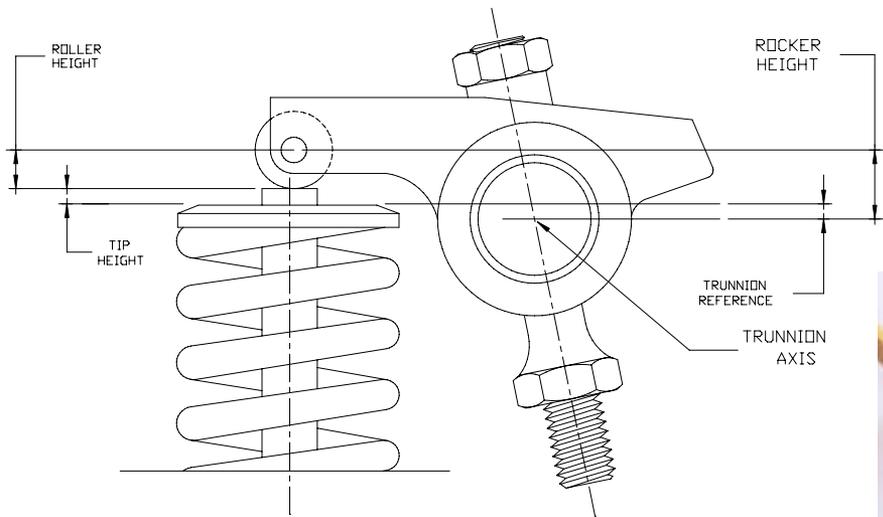
CONVENTIONAL
ROLLER ROCKER ARM
AT CLOSED VALVE POSITION

1



3



HALF NET LIFT

90°0'

CLOSED VALVE
POSITION

VALVE CENTER

2



4
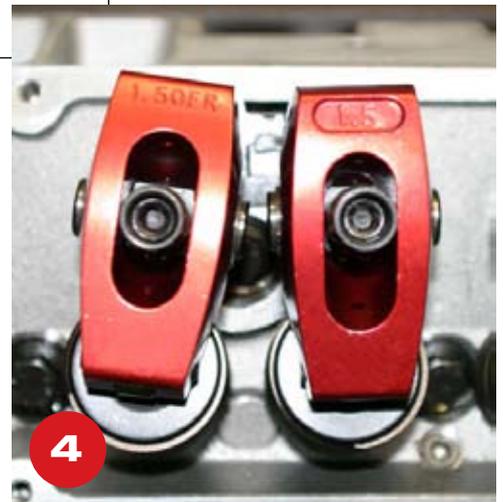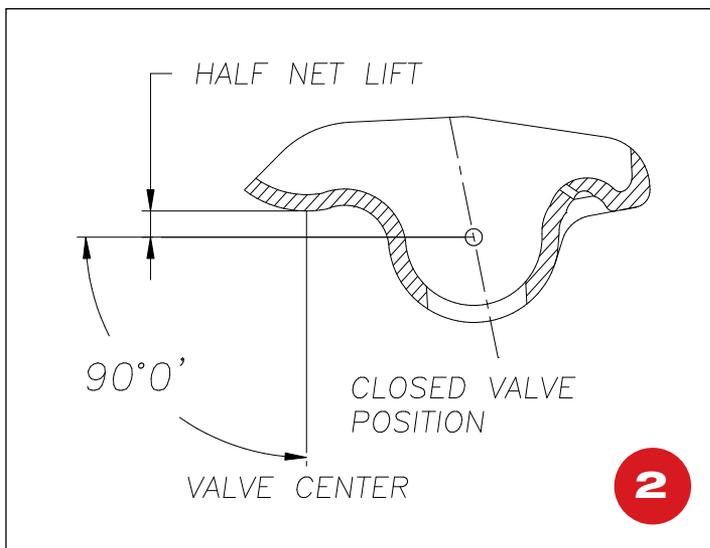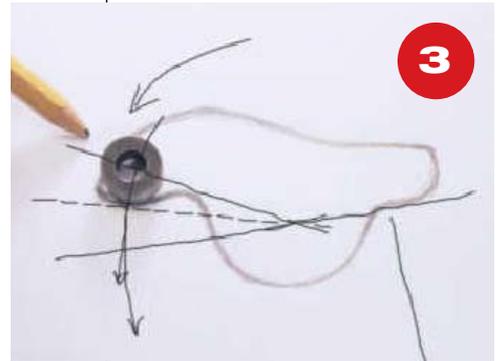
### Figure 2

This is the references to pay attention to for setting a SHOE tip rocker arm for your NET valve lift. As mentioned, finding the centerline of the fulcrum's rotation is required first, since this cutaway only shows where the axis would be. This is not easily seen on the outside of a stamped or cast rocker body so, you need to simulate this rotation and make a mark to reference to.

### Figure 3

As crude as this may look, this is an actual recreation of the drawing illustrated to author by the late Harland Sharp, explaining his original layout of a rocker silhouette on paper before scribing where the roller would be, and using an

actual roller on his outline to calculate this. Setting the roller's diameter in direct position where the scuffing surface was at, instead of the roller's centerline is what is wrong here. It was the beginning of a duplicated error that would last over a quarter century.

### Figure 4

As explained in text, this is the real example of what you don't want to do. But it is very typical, and the problem is propagated more by the head companies luring engine builders into a false sense of "acceptability" to such an installation, by selling their heads with studs in them that have this stretched over placement, when they've had to move the pushrod for wider ports. The better alternative would

be to leave the STUDS on the same centerline as the VALVES, and force stud rocker manufacturers to put their offset for the pushrod solely in the rocker arm, with an offset cup. This is what Ford did on their N-Head, and it is the best way. Otherwise, you need a stand (shaft) system. Here's the bottom line: You can never have an inline valve array cylinder head, like a SB Chevy, Ford, Mopar, etc., and NOT have the rotating axis of the rocker's trunnion be IN LINE with the CAM. Any twist at all, is COMPOUND geometry, and will make it impossible for the roller to lay flat on the valve, or follow the correct straight down path on the Y-axis.

# ROCKER GEOMETRY

BY **JIM MILLER**

*The Goal*

MID—LIFT

.521

.281

.240

STAND

**5**

.3500

STAND

**6**

.3250

.6500

STAND

VALVE

CLOSED VALVE POSITION

**7**
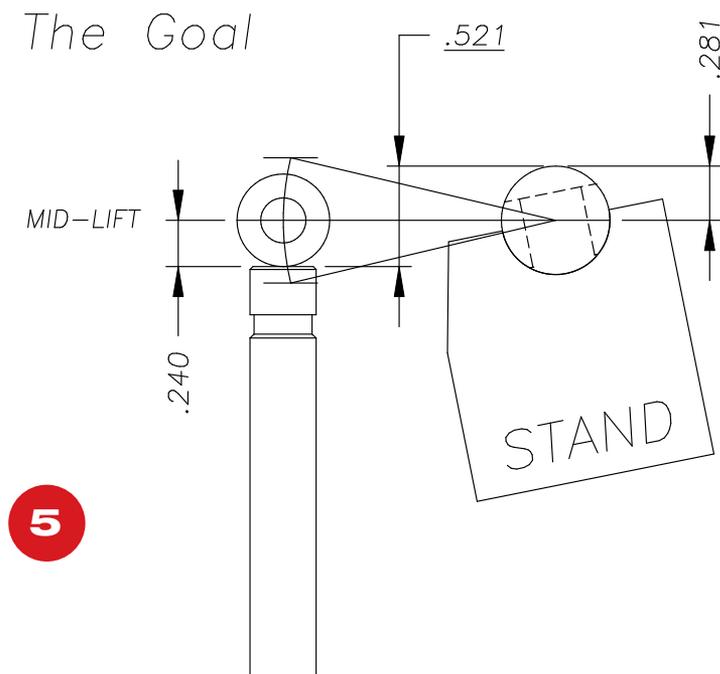
**Figure 5**
This represents the STAND (SHAFT) rocker system, with only the most important things to be considered; namely, the STAND, the SHAFT and the ROLLER TIP.

The valve is shown for angle and location, and is shown here in the MID-LIFT point of motion, as noted to the side. THIS IS THE GOAL. The roller and shaft are to be horizontally in line with each other as measured to a perpendicular right angle with the valve.

**Figure 6**
This shows a SQUARE (crosshatched) being used to lie against the valve and atop the shaft of your stand system. The stand is bolted to the head, and a shaft is laid in position to now make this check while the valve is in the CLOSED position. The cool thing about this is it can be done on a work bench. No springs, no anything, just the parts shown. If you are doing this with an assembled head, you can run the square along the side of the valve springs, providing they're uniform diameter. Otherwise, you may have to use the valve spring retainer technique from our stud mount example.

**Figure 7**
Here's the stand (shaft) mount system shown in the close position, and our example here has a NET valve lift of .650", or a MID-LIFT value of .325".
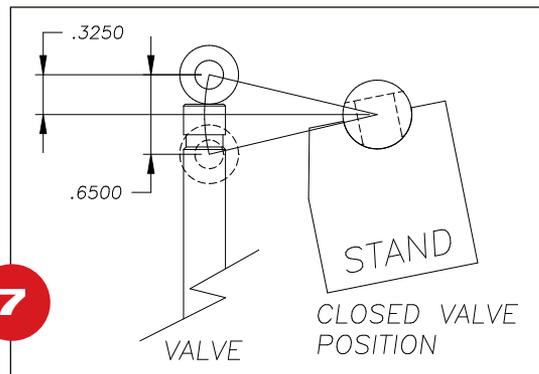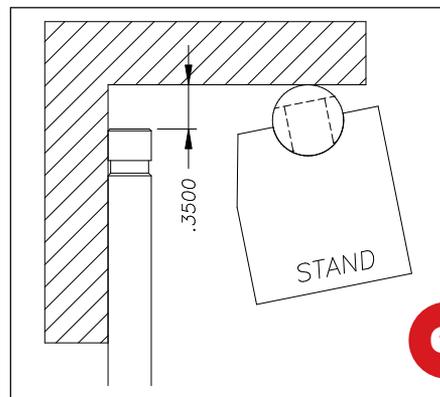
You MUST know this for your engine. It is impossible to set correct mid-lift rocker geometry without knowing your net valve lift. The ROLLERS are shown here in their two critical states, the dashed version representing where the roller will be when it has opened the valve to FULL lift. But the valve is only shown in the closed position, as is the solid roller atop it.

First, take HALF of your ROLLER diameter; and HALF of your SHAFT diameter, and ADD them together, you will come up with a "standard." In this example, that standard would be .521". This comes from a roller diameter of .480" and a shaft diameter of .562". Why half? Because this finds our CENTERLINE for each. It is always the centerline that you are setting with rocker geometry.

Second, write down the height of the SHAFT'S TOP above the valve tip.

Third, you need to write down your MID-LIFT value (half net valve lift).

**Here's the TRICK:**
With these three things written down:

1. Subtract HALF net valve lift from your STANDARD (.325" from .521" in this case, for a sum of .196").

2. Subtract the sum of the above (.196" in our example) from your rocker's height (for our example this would be .350" minus .196" = .154". This is how much the SHAFT needs to be

lowered to bring the centerline of the shaft, half of the net valve lift below the center of the roller). Usually, with shimming, you might have figures that make you ADD shims to get the correct value.

As you might notice, we are using the outside diameters of the shaft and roller to calculate this standard from, because these are easily measured with common tools. But it is their centerlines that are so important.■

Jim Miller has been involved in the racing industry for more than 30 years. Jim's roots stem from amateur racing, starting at 16 but by age 17, he was the youngest Ford trained line mechanic authorized to do warranty work on all of Ford's factory muscle cars. At the age of 21 Jim was invited to take over Chief Mechanic duties for Dyno Don Nicholson's Pro Stock Maverick, which Jim respectfully would decline – passing the opportunity on to Jon Kaase. Jim holds a number of patents for valve train design and is the proprietor of MID-LIFT Precision Geometry. Jim can be reached at 1777 Blount Rd., 501 Pompano Beach, FL 33069. Phone 954-978-7001.