THE BIG TWIN HIGH-PERFORMANCE GUIDE

INCLUDES:

- PERFORMANCE THEORIES
- ENGINE COMBINATIONS
- STROKER MOTORS
- BIG CARBS
- HI-FLO EXHAUSTS
- POWER CAMS & VALVETRAIN
- HIGH COMPRESSION
- HOT IGNITIONS
- TOP TUNING TIPS
- HOT STREET & RACING INFO

GET THE MOST OUT OF YOUR BIG TWIN FROM STREET TUNING TO ALL-OUT DRAG RACING

Winning Combinations

Little things don't mean a lot — they mean everything

❖

Practice does not make perfect — Perfect practice makes perfect

❖

Losers want to win on Saturday, but not the other six days a week

❖

Winners make it happen — Losers let it happen

❖

The harder you work, the luckier you become

❖

Effort + Concentration + Skill + Dedication = Success

❖

Knowledge is Power

❖

Knowledge doesn't become power until it is used

❖

Knowledge = Power = Performance

❖

You can't beat the competition unless you know more

❖

The more you know, the faster you go

❖

It's not how much you learn, but how much you remember
and this book is your "bible" to remembering

❖

Invest in your own knowledge — read this book

Contents

Preface

In 1964 I bought my first Harley—a Sportster for $1400. At the time, the name Harley-Davidson was synonymous with high-performance and the Sportster was considered the Corvette of the motorcycle world. Also, Harley's race team was having great success with the Sportster on the fast TT tracks that were then prevalent around the country. The competition was primarily from Triumph Bonnevilles, BSAs and other British twins.

The Sportster came equipped with a very strong, highly developed engine that had roots dating back to the early 1950s and the side-valve V-Twin engine typically referred to as the Model K. The engine was an excellent choice for those who wanted to increase horsepower because it responded extremely well to modifications.

My initial quest for more performance led me to increasing the size of the Sportster's engine from 55 cubic inches to 65, along with adding hot cams, a bored-out Linkert carburetor and ported heads. At that time, finding any performance tips or information was difficult at best. Some racers would share their knowledge, while others would not. A performance oriented document could have saved me countless hours of confusion, many costly phone calls and the effort to reinvent the wheel.

Since the 1960s, the high-performance market for Harley-Davidson motorcycles has grown tremendously and this has created a big demand for performance parts. Consequently, we are faced with choosing from a large and often confusing variety of performance parts and components. Sorting through the vast array of sophisticated parts is a more difficult task today than it was years ago. Although some performance information is available, it is frequently superficial in nature and often lacks an in-depth and complete perspective of key performance considerations. Additionally, it is usually scattered around and hard to locate. As a result, I decided to take on the long process of writing this manual with hope that it would eliminate much of the confusion and pitfalls facing today's performance enthusiast and racer, while providing essential performance information in one handy guide.

The Big Twin High-Performance Guide is designed to provide performance and racing enthusiasts with the most comprehensive information ever presented in one manual dedicated to Big Twin performance. It takes a vast amount of vague and confusing data and turns it into information that will help you or your mechanic build an engine that will not only make more power, but also have a better chance of reaching its real potential. It strips away the cloak of mystery shrouding the secrets of the fast and famous. Discussed in detail are different engine combinations and why certain combinations work better than others. Included are chapters covering performance considerations, engine combinations, carburetion, cam and valvetrain factors, high compression, high-flowing exhaust systems and hot ignitions.

Also included are many performance engine building principles and concepts along with performance tuning procedures that will benefit not only Big Twin enthusiasts, but also automotive enthusiasts, vocational school students and avid do-it-yourself engine builders and racers.

Hopefully, ***The Big Twin High-Performance Guide*** will help stimulate your creativity, enhance your dedication and promote your will to experiment by providing you with a base of knowledge required to differentiate yourself from the competition.

Whether you want to learn Harley performance engine modifications and tuning from the ground up or simply want to sharpen your general performance knowledge, ***The Big Twin High-Performance Guide*** is your encyclopedia for more performance. ❖

How To Use This Book

The ***Big Twin High-Performance Guide*** covers the Harley-Davidson Big Twin Evolution and Shovelhead models. The Evolution motor entered production in 1984 and is currently still in production, while the Shovelhead motor was produced from 1966 through 1984.

This comprehensive guide is designed for either the serious racer or novice performance enthusiast. Its wealth of information is also beneficial to automotive enthusiasts, vocational school students and avid do-it-yourself engine builders.

The guide provides the information you need to maximize the performance of your Big Twin. However, before attempting any operation involving engine disassembly or part replacement, it is suggested that first you consider your capabilities and skills, besides the tools and equipment available to you. If you intend to perform any performance modifications, it is highly recommended that first you buy a factory authorized service manual and parts catalog to acquaint yourself with basic maintenance procedures and major repair methods. This book is not intended to replace the factory service manual; instead, it is meant to provide supplemental information. Remember to follow all safety recommendations in your service manual.

Keep in mind that some procedures require special equipment and precision tools. Many of these tools can be purchased from dealers, aftermarket performance companies and some automotive specialty shops. Refer to the Performance Directory in the Appendices for a listing of parts and tools suppliers.

If you're in doubt about performing any performance work, remember there are numerous highly qualified dealerships and aftermarket performance shops that specialize in performance modifications.

This guide is divided into eight chapters with each chapter devoted to a major performance topic. Also included in the Appendices are camshaft specifications, ancestry of the V-Twin engine, racing organizations, high-performance parts suppliers along with new performance products and updated information for this edition. ❖

Chapter 1

Introduction

Insights

When the local small talk turns to motorcycles these days, Harley-Davidson inevitably is a major topic of discussion.

It's not simply another motorcycle. Instead, it is an experience of sounds and sensations that generate the great interest among Harley enthusiasts in the marquee's performance. The Harley V-Twin engine design has evolved through over 90 years of development. These engines are typically low-revving and provide lots of low-end power that's ideally suited to slow, easy cruising. By today's standards, these engines produce about half as much power per cubic inch as do contemporary, state-of-the-art motorcycle engines. So they are not really fast. Nevertheless, Harley-Davidson is riding the crest of popularity and sales are booming. Harley's low tech design and evolutionary developments offer owners great resale value that results in an excellent investment.

Today's Harley owners come from many walks of life. Some are hard-core, tattooed bikers while others are Rolex-donned business men or celebrities. Others are purebred racers to whom speed is most important while still others are touring riders with fully dressed machines enjoying the sights and occasionally needing to pass an eighteen-wheeler. Regardless of their lifestyle, over time the vast majority of owners develop their own ideas what a Harley should be and eventually opt for more power and performance. And Harley's low tech, evolutionary design lends itself to performance oriented projects.

Many riders walk into an engine shop and ask for the perfect engine. They say, "I'd like lots of power down low, a bunch in the midrange, and a strong top-end." However, building an engine is a compromise. The components selected determine an engines "personality" or powerband. Some modifications, such as greater displacement and higher compression, help performance throughout the RPM range, while other enhancements invoke compromises. Generally, for a given engine size, if it is built primarily for top-end power some bottom end performance will be sacrificed. But the bigger the engine, the less compromise that is necessary. Having the knowledge to select the right parts and install them properly can help you minimize the compromises.

In general, there are five routes to more power. An engine can be built larger with a bigger bore, longer stroke, or both. It can be made to turn faster. It can be modified to fill the cylinders with more air and fuel. It can be optimized to burn more fuel, thereby increasing combustion efficiency. Finally, friction losses can be reduced. Each method offers its own advantages and each one involves various tradeoffs.

The first question the performance enthusiast must answer when deciding to modify an engine is: what is the engine's major purpose and how will it be used? For example, is it going to be used for all out racing, touring, hot street, Bonneville, etc? The answer will determine which methods are employed to gain more power.

With the vast array of performance parts available, it's easy to dump big dollars into engine modifications. Therefore, it makes sense to know what options are available and what to expect, regardless of whether you are doing the work yourself or paying someone else to do it. Also, many of today's engines are built by catalog part numbers and then assembled by way of instruction sheets. And this is okay for many performance enthusiasts; however, serious racers need every last fraction of power. They need to have a "performance edge." Consequently, they need to know not only how and why things work, but also how to modify a part so it will perform at its best.

This book is designed to help you better understand the available performance options along with providing the insight not only to do things right, but also to do the right things. The book contains the essence of what I have learned during my 35 years of motorcycling, of which the last 29 have been devoted entirely to building, riding, and racing Harleys. Hopefully, it will be of interest and benefit to you. ❖

Chapter 2

Power Theories

Myths & Truths

The key to performance is knowing what is right, what is wrong and how to make it better.

If you copy the latest trick design or speed secret from someone else without having the knowledge to understand fully how to use it and make it work, you can end up going backward instead of forward. Performance depends on the collection of many small elements that collectively add up to a competitive edge. To win today, a powerful engine alone is not enough. Understanding why things work, having a specific plan, keeping things simple and building the engine to last to the finish is also necessary. Building a good engine takes a lot of patience, perseverance and dedication to doing things the correct way. Additionally, attentions to small details, mechanical skill and hard work are also required.

There currently exists a large variety of quality, highly developed performance parts for the Harley-Davidson Big Twin. As a result, there is a deluge of potent, stock displacement and big inch stroker engines running on the streets and drag strips. So, you may wonder, what factors allow a racer to differentiate himself from the competition? Why is one 98 cubic inch stroker engine consistently faster than other 98 inchers? There are many answers to the above questions, but the basis of each answer is centered on one common point — and that point is *knowledge*.

Many of today's engines are built by catalog part numbers and then assembled according to instruction sheets. And this is acceptable for many enthusiasts. However, if you really want to go fast and keep a step ahead of the next guy, you need to know why things work, how to select the right combination of parts and how to modify parts to perform at their best. As an example, if you want to improve your engine's performance, but don't need to be the fastest guy on the block then there are many combinations of carbs, cams, exhaust systems and cylinder heads that will give a respectable power increase. Yet, if you're competing in a heads-up class, every last fraction of horsepower is vital to winning and you need the best combination of parts — not just any parts. Although, the best combination is dependent on many factors, with the proper knowledge and some experimentation, you can assemble an excellent combination for your application.

Racers who are frequent winners don't win just because they copy the same combination everyone else is running. Although their combination may be similar to their competitors, they prove out the combination through trial and error testing and experimentation. For example, some racers order two or three different cams that theoretically appear correct for their engine displacement, head flow characteristics, gearing, bike weight, etc. Each grind may vary in intake or exhaust duration or lobe separation angle. Then, after testing each cam, the best one is selected. This time consuming, cut-and-try experimentation often is the reason one racer consistently runs a fraction of a second faster than all the others.

With the enormous interest in high-performance Harleys, the brightest racing future lies ahead for the self-taught, cut-and-try experimenter — particularly one well versed in electronics, chemistry, petrochemistry, physics, metallurgy, aerodynamics, and chassis-dynamics. In racing, your effectiveness depends on the collection of many techniques that add up to give you a competitive edge. Understanding certain principles is the genesis of that competitive edge. And knowledge, dedication, perseverance, preparation, experimentation, attention to detail and hard work, when applied to performance engine building, provide the opportunity for that competitive edge. There is a cliché that says "Little things mean a lot." Don't believe it. Little things mean everything. It's the little things that add up to that extra 1/10 of a second or going 30,000 miles between engine rebuilds instead of 5000.

THE OBJECTIVE

Before you build an engine you must first decide the purpose of the engine and its rpm range. For example, do you want to increase the touring performance of a heavy "dresser," build a top-fuel drag bike, or are you interested in a hot street bike or bracket racer? Your answer will help determine the engine's size and the components.

Currently, top-fuel drag engines range between 114 and 120 cubic inches with some nearing 150 cubic inches. Bracket racing and hot street engines can be any size you can afford to build, although most generally end up between 80 and 98 cubic inches with some as large as 103 to 114 cubic inches. Stock class engines are just as the name implies, unmodified 74 or 80 cubic inches (the 80 is actually 81.7) for Big Twins.

Evolution Big Twin engines are limited to about 5250 rpm by the stock ignition module. Although many factors, such as stroke length, cam design and cylinder head airflow contribute to an engine's rpm range, most mild performance engines are revved between 5800 and 6500 rpm, while more radical engines are turned between 6500 and 8000 rpm. Regardless of the engine's purpose and rpm, it must be durable enough to finish a race and dependable for worry free street riding. This requires proper assembly procedures, proper break-in, correct maintenance and using some discretion when riding.

WINNING COMBINATIONS

The Big Twin is composed of many complex, interdependent components and for each part to be effective it must be compatible with numerous other components. Component selection must be based on how the pieces will work together. This means that each engine configuration must be carefully matched with complementary combinations to produce the best horsepower and torque curve for a particular application.

However, no combination is perfect because there are too many tradeoffs. Consequently, there is no such thing as a free lunch. If you build the engine *primarily* for top-end power, you'll give up power at the lower end. This is because an engine will only give maximum performance over a short 1500 or 2000 rpm range.

The key to reliably going fast is to select a good combination of parts that all work in harmony with each other — each one building upon the contributions of the others. For example, a cam should be matched to the head's airflow capability, engine displacement, bore-to-stroke relationship, rpm range and desired torque curve. Any change to the combination requires retesting to confirm the impact of the change. Remember, for every engine there is one component that creates a threshold point that limits an increase in performance. Once the restricting component is replaced, the next restriction in the chain becomes the limiting factor. In summary, run a combination of parts that work together because an engine is a system, not a list of parts.

THE PUMP

At first glance the internal combustion engine seems like a rather simple device. A mixture of fuel and air is drawn into a closed chamber where it is ignited and burned to produce energy. However, as we spend more time analyzing the engine, it becomes obvious that it is quite complex and offers an endless number of possibilities for power and efficiency improvements.

In its simplest form, the internal combustion engine is nothing more than a pump. To be more accurate it can be called a heat pump because heat is power. Regardless of the type of engine you build, it will never live up to its potential unless it is an excellent sealing pump. What we want to do is make a "happy motor." By this it is meant that the engine doesn't need to have the most radical cam or biggest carb, but it does have to have good airflow and excellent sealing characteristics. Excellent pump seal and the right combination of parts are the two most important characteristics of a good engine.

In fact, there is no way to tune an engine correctly unless it has an excellent seal. A commercially purchased leak-down tester can be used to determine the percentage of leak-down for an individual cylinder. The process involves pumping air at about 100 pounds per square inch into a cylinder and measuring the amount of leakage. Two percent leak-down or less is excellent. Three to seven percent is considered maximum for a race engine. Many factory built engines fall into the seven percent area. Anything above seven percent is considered out of the ball park for serious racing because power is going to be down and it's impossible to read the spark plugs accurately. Ideally, we are looking for a maximum of two percent leakage after three or four passes down the drag strip.

To build a good pump, you need to start with round, straight cylinder bores. When head bolts or studs are torqued, they twist and distort the cylinder. To get a cylinder to seal properly, you need to use a torque plate during the boring and honing process and it is important to use the exact type head gasket, bolts or studs and the correct torque when setting everything up for machining.

To determine whether the torque plate is duplicating the same loads on the cylinder that the cylinder head imposes, visually check each bore after initial run-in. Shaded areas indicate uneven ring to cylinder wall wear. Compression ring seal at the top 1-3/4 inch of ring travel is the most important area. Beyond that, oil control is the major consideration. Keep in mind that a compression leak-down of less than two percent, when coupled with high oil ring leakage, also is a bad combination due to potential detonation and the inability to read plugs correctly.

Another important point is to ensure that the cylinder heads and crankcase decks are as flat as possible. Finally, a top notch cylinder head job with good valve guides, solid seats and concentrically ground valves and seats completes the pump.

ENGINE CYCLES

Now that some critical elements for a good sealing pump have been discussed, lets review the major cycles of the four-stroke engine or pump. Specifically the cycles are the intake, compression, power and exhaust. All four cycles thrive on the efficient use of air and fuel and the key word is efficient. By efficient it is meant that the air and fuel can be examined not only for *quantity*, but also *quality*.

During the intake cycle the piston moves down the cylinder and the greatest possible amount of fresh air/fuel mixture must be drawn into the cylinder.

Next, the piston moves up the cylinder and compresses the mixture. During this cycle the intake charge not only must be compressed to increase cylinder pressure, but it also must be thoroughly mixed for the highest possible amount to burn. Additionally, it must be ignited at exactly the right moment with a big fat spark from the ignition system.

The third cycle is the power stroke. Now the burning fuel creates a pressure increase on top of the piston that forces the piston downward in the cylinder. If designed correctly, the engine will optimize cylinder pressure, maximize the burning fuel's chemical energy and convert the mixture into the greatest amount of mechanical energy to turn the crankshaft.

Finally, during the exhaust stroke the piston moves back up the cylinder and purges the remaining burned mixture from the cylinder. For maximum performance, the engine must maximize the amount of burned mixture expended from the cylinder while minimizing the effort required to purge it.

Notice that the description of the four engine cycles not only mentions that the maximum quantity of mixture must be drawn in or purged out of the cylinder, but it also emphasizes the thoroughness of the air/fuel mixing, exact timing of the ignition, optimizing cylinder pressure, maximizing energy use and the completeness of expending exhaust. These points refer to the *quality* of the processes that take place during the engine cycles and are key for maximizing the power output of any given engine combination.

TORQUE vs. HORSEPOWER

A completely stock 80 cubic inch Big Twin engine produces about 57 horsepower at 5200 rpm and 70 foot pounds of torque at 3000 rpm, when measured at the rear wheel. Before discussing methods to increase these numbers, let's take a closer look at what they mean.

Figure 2.1 reflects the horsepower and torque curve of an average stock Big Twin engine. To understand the relationship of torque and horsepower and their significance, it is first necessary to understand how they differ. Torque is the force that moves or accelerates the bike. It is also expressed as the amount of work

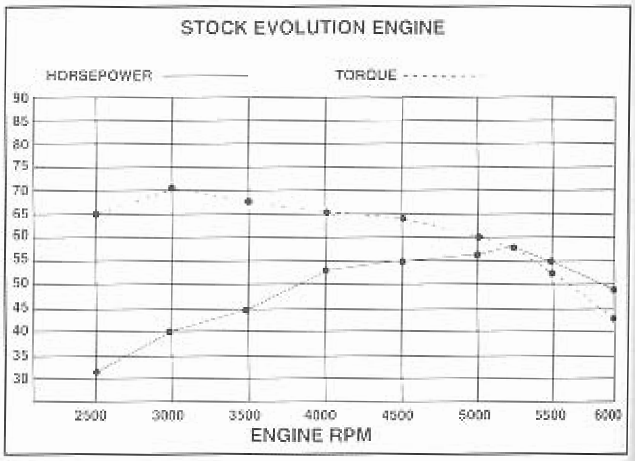


Figure 2.1 *The stock Evolution engine achieves maximum torque near 3000 rpm because cylinder fill is highest at this point. The major reason horsepower drops off after 5250 rpm is because the rev limiter kicks in. Take note that horsepower and torque always cross at 5252 rpm.*

an engine can produce and is the equivalent of how much force is pushing down on the piston and how hard the crankshaft is being twisted. Horsepower, on the other hand, is the speed at which an engine performs work. The faster an engine performs a given amount of work, the more horsepower developed.

An engine is sometimes referred to as a "torque engine," meaning that it was built predominantly with torque in mind. In reality, there is no such thing because any engine that has torque, also has horsepower. Figure 2.1 shows that the Big Twin produces 60 pounds-feet of torque at 5000 rpm. In fact, any engine that has 60 pounds-feet of torque at 5000 rpm, also has 57 horsepower at that engine speed. Torque is present at all engine speeds and more torque at a given rpm always yields more horsepower at that rpm.

Torque and horsepower are often confused, although they are closely related in the equation for measuring horsepower. Currently, the only efficient method that exists for measuring horsepower is by measuring actual engine torque. By definition, horsepower is equal to force multiplied by distance, divided by time. With the proper constant in place, a simple equation is used to calculate horsepower:

$$hp = \frac{torque \times rpm}{5252}$$

The constant 5252 originates from the following formula used to determine horsepower: 33,000 pounds-feet divided by 2 times *pi* (6.283) equals 5252.

Take note that torque and horsepower are always equal at 5252 rpm. This results in torque and horsepower always crossing at 5252 rpm when drawn on a line graph. One pound-foot of torque below 5252 rpm produces less than one horsepower, which means that torque below 5252 rpm will always be higher than horsepower. Conversely, one pound-foot of torque above 5252 rpm makes more than one horsepower. Consequently, torque above 5252 rpm is always less than horsepower.

TORQUE PEAK

The reason the Big Twin's torque falls off in the higher rpm range is that the cylinder's volumetric efficiency (VE) drops and torque is directly proportional to VE. This is because VE represents the percentage of cylinder fill. Torque peak is the point at which the engine achieves maximum cylinder filling. When an engine's VE (and therefore torque) declines faster than its rpm rises, the engine has gone beyond its torque peak.

Airflow generally increases with rpm until it reaches the maximum flow capacity of the valves, ports, manifold, carburetor or whichever is the major restriction. Once maximum flow capacity is reached, it remains constant despite increases in engine rpm. At this point the engine becomes "flow limited," therefore volumetric efficiency and torque go down. If you increase the engine's cylinder filling capability, the torque peak moves up the rpm band along with horsepower. The greater the cylinder is filled, the greater the force exerted on the piston by the expanding gases and the greater the torque.

The point at which the engine runs out of air is determined by the efficiencies of the intake and exhaust system and the cam. However, combustion chamber design, piston dome shape, spark plug location, and the evenness of temperature throughout the combustion chamber determine the actual power potential of the engine. If the engine has a good combustion chamber design and sufficient intake and exhaust capability, power will continue to increase as the engine turns higher rpm's. The amount of torque may not increase in value, but it will move to a higher point in the rpm range. In reality, the real key to making power for a given engine displacement is the engine's ability to breathe and the total design of the combustion chamber.

A highly tuned engine's torque peak is usually about 75 percent of redline rpm. The Big Twin's torque peak is about 57 percent of its 5250 rpm redline. Figure 2.1 shows that torque at 6000 rpm is about 61 percent of what it is at 3000 rpm. This indicates volumetric efficiency is down due to an inefficient intake system, poor combustion chamber design and a restrictive exhaust.

If you don't exactly follow the above formulas and details, don't worry because they're not that important. What is important, however, is the relationship of torque to horsepower. Keep in mind that at a specified engine rpm, horsepower is directly proportional to torque. As torque increases at a given rpm, horsepower increases a corresponding amount. If torque remains constant, horsepower will either increase or decrease in direct relationship to a rpm change. By spinning the engine faster, more horsepower is developed for a given amount of torque. For a given operating range, the only way to increase horsepower in that range is to increase torque. To produce a certain amount of horsepower at a low rpm requires greater torque than it does at a higher rpm. Assuming an engine is capable of breathing sufficiently, the most effective method to increase torque and thereby horsepower is to increase its displacement. Understanding these relationships will help when selecting an engine combination for a particular application.

When we think about power, we're inclined to judge engines exclusively by peak power without considering the operating range where the power occurs. Instead, engines should be compared by the amount of power they produce in the intended operating range. Big Twin drag engines typically operate between 4500 and 7500 rpm, while a street driven "dresser" engine usually runs between 2000 and 4000 rpm. The drag engine is built for maximum torque at high rpm because this produces the most potential horsepower, although it invariably reduces power in the lower rpm ranges. Then the engine is geared and ridden in a narrow, high rpm range to take advantage of the torque curve. The "dresser" needs a different engine combination, that produces a different torque curve to address its lower operating range requirements.

In summary, to increase torque for a given engine displacement, you can increase compression, get better flowing heads, make the combustion chamber more efficient, change the camshaft timing, and tune the induction and exhaust systems. Increasing torque, and shifting its location to the proper operating range is crucial to building a winning engine combination, regardless of whether the intended use is drag racing, hot street or touring.

HORSEPOWER VARIABLES

In reality, there are three variables that control horsepower: displacement, rpm and brake mean effective pressure or BMEP. First, a large displacement engine breathes in greater amounts of air/fuel mixture, consequently, it produces more power. Second, revving the engine faster through higher rpm allows an engine to perform its power-producing cycle more frequently, which produces more power. The last variable, BMEP, is more complex than the first two because it involves the following: the amount of air/fuel mixture filling the cylinder; the amount of mixture that is burned; and the amount of power that is lost to internal friction and heat. In simple terms, BMEP means average effective combustion pressure.

These three power variables lead to five methods for increasing engine power. The first method is to *increase the engine's displacement* with a longer stroke, bigger bore or both. Second is to *rev the engine faster* to take advantage of more power strokes in a given amount of time. Third is to *fill the cylinders* with more air/fuel mixture. The fourth method is to *enhance combustion efficiency* by burning the greatest possible amount of the air/fuel mixture. The last method is to *minimize internal friction* losses by using the correct parts and proper assembly techniques.

LARGER DISPLACEMENT

Displacement is king when it comes to making serious horsepower, especially if you're dealing with a heavy bike. All things being equal, a good big engine will always beat a good small engine. Up to a point, you will benefit by building the largest engine you can afford.

In general, a properly built high-performance Big Twin engine will put out between 1-1/4 and 1-1/2 horsepower per cubic inch. A little math will confirm that cubic inches quickly add up to cubic horsepower. However, keep in mind that there is no guarantee horsepower will increase at the same rate as displacement. For horsepower to increase linear to displacement, the engine components must be compatible and proper assembly techniques must be used.

Given two engines that make the same horsepower, the larger-displacement engine will generally make more torque and this is highly desirable for street and drag engines. Also, street riders who lug their engine at low rpm or those with heavy bikes can especially benefit from high volumetric efficiency in the low rpm range because it increases torque. And this is easier to achieve in a big engine, even if you slightly miss the best combination.

Increasing the engine's displacement allows it to breathe in more mixture during each cycle for greater torque and horsepower. This results in the most effective method for increasing torque and power. The Big Twin engine offers great potential for increased displacement, while still retaining excellent reliability.

A large displacement engine when combined with stock size intake ports will increase the velocity of the incoming air/fuel mixture. This decreases the pressure in the intake tract below that of a stock engine and causes a big engine to run out of breath sooner in the rpm range than a stock engine. And the larger the displacement, the more pronounced the effect is. This means that increasing displacement without increasing cylinder head breathing capacity, results in an engine that produces higher torque, but always at a lower rpm. This phenomenon typically gives the illusion of a more powerful engine during normal riding conditions.

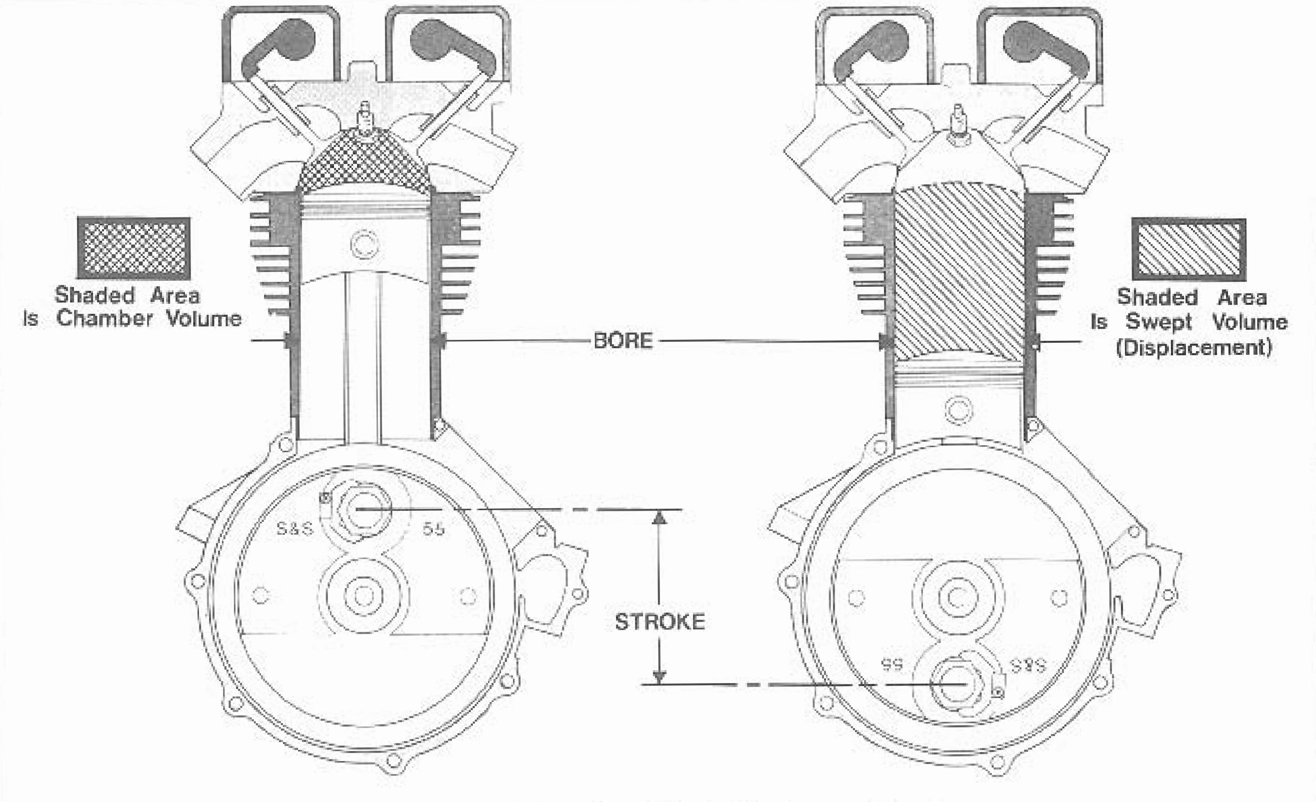
However, as stated earlier, horsepower is a function of torque times rpm. So, without increased breathing capacity it is possible to reach a point where horsepower is decreasing and can drop to a point below a stock displacement engine. An engine must have rpm to make horsepower because 100 foot pounds of torque times zero rpm equals zero horsepower. Since high torque over a maximum rpm range is the most desirable attribute of an engine, major increases in engine displacement should always be combined with increased breathing capacity.

There are three methods for increasing displacement: increase stroke, increase bore, or increase both. The size of a cylinder's bore is about equal to its piston's diameter, while stroke length is the distance the piston moves up or down in the cylinder. A stock displacement Big Twin engine has an under-square configuration. This is because its 3-1/2 inch bore is smaller than its 4-1/4 inch stroke length. Under-square engines generally run at lower rpm and generate maximum torque at lower rpm than over square engines. On the other hand, over-square engines have a larger bore than stroke and typically generate maximum torque at a higher rpm level.

LONGER STROKE

The most effective way to increase torque and thus power, is to increase an engine's displacement. Boring and stroking both increase torque, however, stroking a Big Twin engine is most effective. Dollar for dollar and cubic inch for cubic inch, the most noticeable performance improvement with a Big Twin comes from an increase in stroke. Over the last thirty years, racers proved that Harley V-Twin engines really respond to stroking. To understand why, let's consider bore-to-stroke configurations.

Increasing the engine's stroke lengthens the crankshaft's arm. This is accomplished by moving the position of the connecting rod crankpin farther from the rotational centerline of the crankshaft. This change not only increases engine displacement, but also increases the mechanical leverage the rod has on the crankshaft. As a result, torque increases at the crank throughout the rpm range since torque equals



$$\text{Compression Ratio} = \frac{\text{Swept Volume (or Cylinder Displacement)} + \text{Chamber Volume}}{\text{Chamber Volume}}$$

Figure 2.2 *The distance the crankpin is positioned from the flywheel centerline determines the engine's stroke. Stroke is equal to twice this distance. For example, an 80 cubic inch Big Twin engine has a stroke of 4-1/4 inch. This results in a crankpin location 2-1/8 inches from the flywheel centerline. Illustration courtesy of S&S Cycle.*

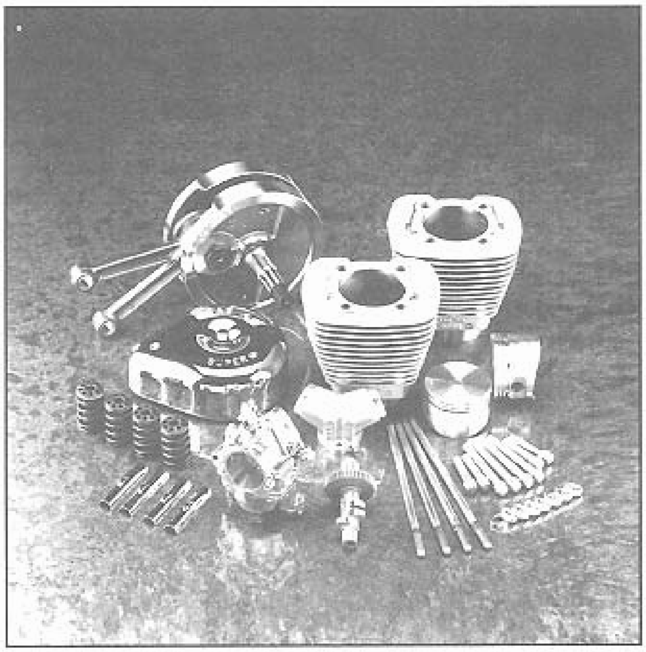
force times the length of the arm on which it pushes.

As with anything, there is no such thing as a free lunch. Although, we have elaborated on the benefits of enlarging displacement by increasing stroke, there is also a downside. This is in the form of increased internal friction and cylinder wear generated by the greater distance the piston travels during each cycle and the increased angularity of the connecting rod. The following formula is used to determine average piston speed in feet per minute.

$$\text{Piston Feet / Min.} = \frac{stroke \times 2 \times rpm}{12}$$

Notice that the formula calculates average piston speed, not maximum speed. Maximum speed is actually much higher and is arrived at approximately 75 crank degrees after either top-dead-center (TDC) or bottom-dead-center (BDC). However, average speed works fine for our comparison.

The stock Big Twin engine has a stroke of 4-1/4 inch and it redlines at about 5250 rpm. Inserting these constants into the above formula results in an average piston speed of 3683 feet



Long stroke flywheels and stroker pistons provide a simple and direct route to increased displacement. Large bore cylinders can be installed along with the stroker flywheels or added later for an even larger increase in displacement. Photo courtesy of S&S Cycle.

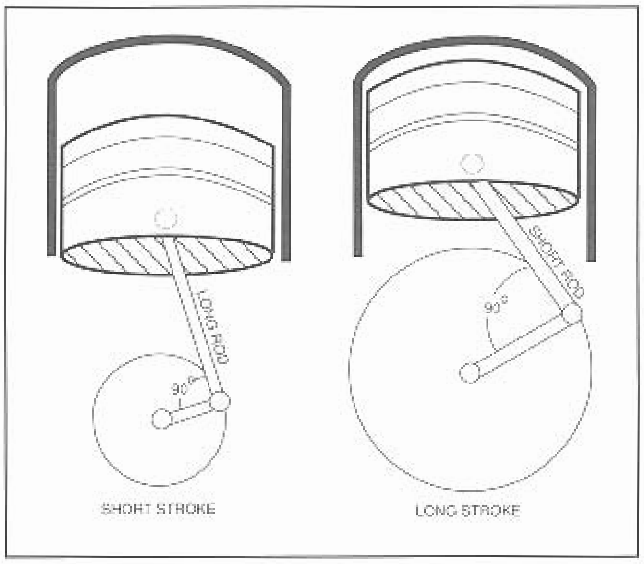
per minute. Now, if we increase the stroke to 4-3/4 inch (which is a very common stroke) and still redline at 5200 rpm, we see that piston speed increases to 4117 feet per minute. However, neither of these piston speeds is considered exceedingly fast. In the real world, a 4-3/4 inch stroke engine would typically be revved to about 6000 rpm on the street and 6500 rpm or more on the race track. Using 6000 rpm, we arrive at an average piston speed of 4750 feet per minute. Take note that many five inch stroke engines are revved to 6500 rpm for short periods of time, resulting in 5400 feet per minute piston speed.

As piston speed increases, internal friction increases by the square of piston speed. Consequently, frictional horsepower losses increase. Increased piston speed also wears out rings and cylinder bores sooner. Additionally, long strokes increase the horizontal inertia effects on the connecting rod and increase side-loading of the piston due to increased rod angularity. Using a longer connecting rod helps reduce rod angularity.

As such, reliability and longevity can be compromised, unless some discretion is exercised regarding rpm. For a street engine, it is wise to limit piston speed to about 4600 feet per minute most of the time, with only occasional bursts higher. This should be acceptable for street engines, since increasing stroke length generally shifts an engine's torque peak to a lower rpm and proper gearing can be used to take advantage of it.

Although piston speeds can reach high levels with long arm strokers, using discretion is the key to long life and reliability. You should be aware that some riders have recorded 50,000 miles on a large stroker engine without ever having it apart, while others have destroyed a small stroke engine in less than 5000 miles. I personally have put more than 15,000 street miles on some of my stroker engines and could have gone much farther. The mileage included riding double through 7500 foot high mountains and 110 degree deserts besides all out drag strip runs.

It's exhilarating to run through the gears to maximum rpm from every stop with a big stroker. However, it wears out an engine unnecessarily quick. As such, consider the following rule



Maximum pressured is transferred to the piston when the connecting rod is positioned 90 degrees to the flywheels centerline. A combination of short rod and long stroke produces the greatest rod angularity. The longer the stroke, or the shorter the rod, the higher the piston will be in the cylinder when the rod/flywheel angularity is 90 degrees. A longer rod moves the piston lower in the cylinder when the rod/flywheel angularity is at 90 degrees. Engine wear is affected by rod angularity and camshaft timing is influenced by the piston's position in the cylinder at a given flywheel angle.

regarding engine rpm: Never rev a Big Twin engine beyond 5200 rpm or a Sportster engine beyond 6000 rpm unless you want to make a statement to someone. In the end, the longevity of a stroker engine is just a matter of how well the engine is built, broken-in, maintained and the discretion used when it is ridden.

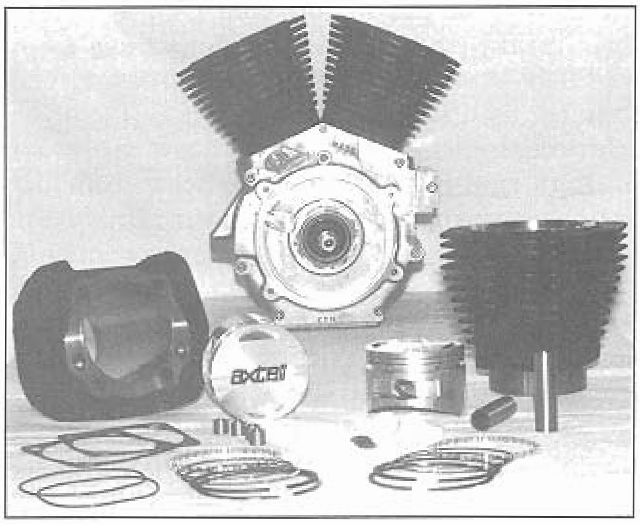
BIGGER BORE

In the fifties and early sixties most increases in displacement for a Big Twin were achieved through longer strokes rather than bigger bores because long stroke flywheels were easier and less costly to make. Since then, big bore cylinders have grown in popularity because of greater availability and the introduction of complete installation kits.

For a given increase in displacement, a big bore engine will produce peak power levels at a higher rpm, but usually provides less torque at low rpm than a stroker engine. For a given engine size, if horsepower is the major consideration, it is better to go with a large bore and the ability to turn higher rpm. With a larger bore diameter, the piston's surface area is larger and the combustion pressure acting against a greater piston area will allow the engine to produce more power at higher rpm.

Big bore cylinders have the potential for more squish area in the combustion chamber. Squish can increase mixture swirl and homogenization, which leads to improved combustion efficiency and reduced potential for detonation. However, remember that very large bore diameters can have greater potential for detonation simply because the flame front has to travel farther across the piston's top. This is because for a given amount of fuel turbulence, the rate of burn remains relatively constant regardless of engine rpm or bore size. Detonation can be minimized or eliminated by increasing the turbulence in the combustion chamber and by reducing the distance the flame needs to travel. Although, there are several other factors that influence good combustion, it should be noted that, in general, the bigger the cylinder bore or the higher the engine rpm, the more difficult it is to achieve complete and equalized burning of the air/fuel mixture across the entire combustion chamber.

Although a big bore engine is usually run at higher rpm than a stroker engine, its piston speed is lower and the rod's side-loading force is less, so the engine has the potential to last longer.



Axtell's 3-13/16" big bore kit increases the Big Twin engine to 97 cubic inches while retaining the stock stroke. Shown is Axtell's Shortblock Kit, which includes a complete bottom end assembly. Photo courtesy of Axtell Sales.

In the end, the question of stroke versus bore is really one of torque versus horsepower. Torque is needed to accelerate the bike from a standing stop, while horsepower is needed to provide maximum speed through a drag strip's timing lights or Bonneville's long straight-away. For maximum performance, the best way to increase displacement is to combine a bigger bore with a longer stroke. This provides the best of both worlds, is cost effective and easily accomplished with one of the many available kits.

HIGHER RPM

A stock Big Twin engine is designed to produce maximum torque and horsepower at rpm levels that fall well below design limitations. This guarantees reasonable engine life and an acceptable period between engine rebuilds. Currently, a Big Twin engine produces maximum torque at about 3000 rpm and maximum horsepower near 5200 rpm.

A rev limiter built into the stock ignition module limits maximum horsepower and rpm by retarding the ignition timing starting at about 5250 rpm. Revving the engine beyond this level will not produce more horsepower unless the intake and exhaust system are modified to allow the engine to breathe better and burn a greater percentage of fuel at high rpm. Remember that engines with increased displacement, such as strokers, will run out of air and be down in horsepower in the higher rpm ranges unless their intake and exhaust systems have been modified for better breathing. Increasing the engine's ability to breathe is a key factor to achieving higher rpm.

High rpm engines require efficient combustion chambers because there is considerably less time to burn the fuel mixture. Characteristics of good chambers include: high mixture turbulence, confinement of the fuel mixture to a small area, minimal flow restriction around the valves, and short unrestricted flame travel.

Rev limits are determined not only by an engine's ability to breathe and its bore to stroke relationship, but also by the speeds at which the engine's parts can no longer support the forces created by the weight and acceleration of the reciprocating parts. An engine's internal stresses increase with the square of the rpm. If the Big-Twin's rev limit is increased 20 percent, from 5250 rpm to 6300 rpm, the increase brings 44 percent greater stress (120 percent times 120 percent equals 144 percent). As a result, modifications that increase the engine's rpm range must also include the proper combination of parts to support the additional stresses. Stiffer valvetrain components, stronger connecting rods, harder crankpins, tougher pistons, better oil pumps and proper assembly procedures are only a few examples of what it takes to support high rpm.

BETTER CYLINDER FILL

Regardless of displacement, an engine must be able to flow a high volume of air/fuel mixture to make serious horsepower. Every internal combustion engine is restricted in the power it can produce largely by the amount of air it can flow through it. So, making your engine breathe better is the first step to making more power.

There are three major methods to improve airflow and cylinder filling: a more efficient induction system, different camshaft, or a less restrictive exhaust system.

Cylinder Heads

The internal combustion engine was described as an air pump and its most crucial component is the cylinder head. The cylinder head is the gateway to success or failure of any engine and proper cylinder head modifications are the key to unlocking the engine's maximum power potential.

Horsepower and torque are quickly gained or lost in cylinder head preparation. For this reason, knowledgeable engine builders spend a major amount of their efforts on cylinder head development to squeeze a few more horsepower out of the engine.

The objective of the cylinder head is to pass as much air/fuel mixture as possible from the mouth of the carburetor or injector, through the intake port and into the cylinders of the engine. Then it must efficiently pass the burned gases out the exhaust port. To be efficient, the intake port must do its job with the least amount of pressure drop (that means retain maximum amount of positive pressure) between the carburetor and valve head. In the case of the exhaust valve the objective is to maintain the maximum amount of negative pressure.

Everyone knows that a large diameter hole will pass more air than a small diameter one. So, the first thought of the uninitiated is to make the ports bigger without thought to why they need to be bigger — if indeed they need to be bigger. Although, the ability of a port to flow the maximum volume of air is mostly a function of its size, some Big Twin ports are already large enough or even too large.

When considering only size and not shape, most ports are optimized for efficiency when they are about 80 percent the diameter of the valve head. Therefore, increased flow cannot be achieved by simply making big holes for the air to flow through because that flow will come at the expense of air velocity. The higher the air velocity, the higher the amount of kinetic energy it contains. As the kinetic energy increases, the air's ram effect becomes greater. Consequently, cylinder fill is increased.

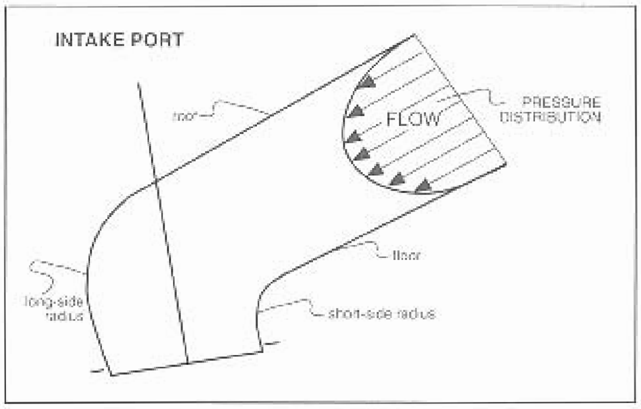
In general, enlarging a port lowers velocity at any given rpm. As a result, ram effect that is affected by velocity is also reduced. This lowers cylinder fill and reduces torque. Ram energy effect increases or decreases with the square of the speed through the port. A port that is small, efficient and flows the same amount of air as a larger inefficient port will generally produce a superior power curve. Experienced porters generally keep ports as small as possible while using optimum port shape to keep flow high.

It should be noted that port size and port velocity are relative values that are dependent upon the objective of the engine. As stated earlier, the engine's objective must be outlined before the right combination of parts can be selected. Different objectives allow for a different combination of parts. For example, what is considered an acceptable port size for a pro stock drag bike will be entirely too large for a mildly modified, heavy street bike. Since the drag bike probably revs between 5500 and 7500 rpm, larger ports can be used since high rpm compensates for sluggish flow velocity at the top end and high velocity at low rpm is not a prime consideration.

This doesn't mean that big ports are useless. The port size is related to the efficiency of the valve. The more efficient the valve is at passing air into the cylinder, the more port area it will accept. In general, port size is determined not only by the size of the valve, but also by the effectiveness of the valve itself.

Intake Port

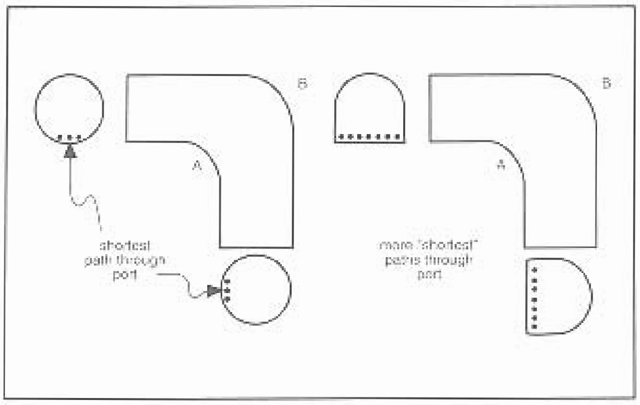
The intake port's responsibility is the combined movement of air and fuel. Up to this point we only have discussed the *quantity* of airflow, however, we also must be concerned with the *quality* of the airflow. Fuel droplets that separate from the airstream will puddle in the port and lead to inefficient combustion and lost power. The air/fuel mixture must be "conditioned" to maintain a well mixed (atomized) air and fuel so that it is in its most combustible form. This point is crucial to making an efficient, powerful engine.



Pressure and velocity distribution are near symmetrical at the mouth of a port. But as the flow changes direction at the radius, velocity and pressure become less uniform. Velocity becomes higher and pressure lower on the long-side radius. Conversely, velocity is low and pressure is high on the short-side radius. Increasing the size of the port equally around its circumference will reduce velocity. However, removing material only in selected areas can increase velocity in high pressure areas, leading to increased flow.

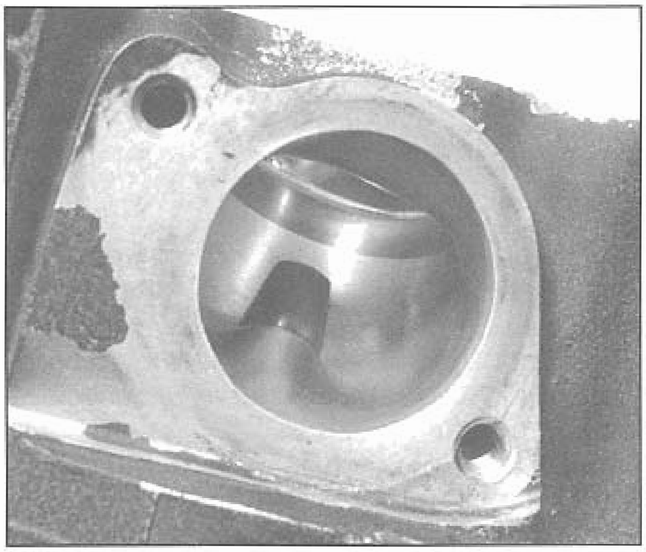
Pressure distribution within the port tends to change and lose its uniformity with directional changes in flow. This is typically happens at the port's short-side radius (the sharp turn on the floor of the port). Keeping all areas of an intake port equally pressured and as flow-active as possible discourages air/fuel separation. Velocity probes can be used in conjunction with a flow bench to measure pressure uniformity.

The flow path within a port usually takes the shortest route to the combustion chamber and this is along the short-side radius. The air/fuel mixture moves slower along the short-side than



Flow seeks out the shortest path at location A. Pressure is high and velocity is low at this area. By flattening the area of the shortest path into a D-shape, flow volume is increased because pressure is more evenly distributed throughout the port's cross-section.

the long-side. Pressure goes up on the short-side and causes the fluid to tumble because on the long-side the velocity is up and the pressure is down. So, the idea is to increase the area on the short-side to pick up velocity and drop the pressure closer to that of the long-side. If the short-side radius is flattened so the port takes on a D-shape, flow can be increased without increasing cross-sectional area that would reduce velocity. With proper contouring and size, a port will maintain high flow rates in combination with pressure distribution that is as uniform as possible. This will reduce air/fuel separation and promote sustained atomization.



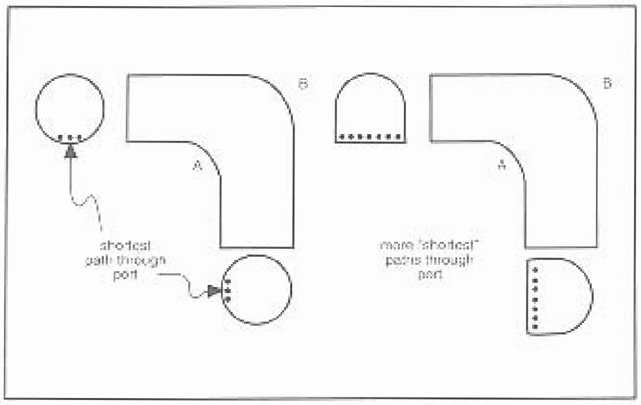
Notice the flattened D-shaped bottom of the intake port. Enlarging a port only where airflow is restricted increases flow while keeping velocity high. The port also can be biased to help direct the mixture into the cylinder. Photo courtesy of Carl's Speed Shop.

A port's surface texture is extremely important to good combustion. The highest flow rate is at or near the center of the port, with little or no activity at the port walls. This is the same for a river of flowing water. As the flow passes through the port, it wets the outer walls and forms what is called a "boundary layer" that has some degree of thickness. Its thickness is partially determined by surface finish. During normal flow, some fuel droplets separate from the air and puddle on the port walls or boundary layer. Large droplets burn slower than smaller droplets so there is a greater possibility for the fuel to pass through the combustion chamber unburned, resulting in lost power.

A polished intake port allows fuel to creep in a puddled form down its walls, which causes a high percentage of fuel to pass through the combustion chamber unburned. Roughening the port walls is a technique that can help the puddled fuel back into the airstream. Although, this technique has the effect of shrinking the port due to increased drag, the positives out weigh the negatives. Atomization and mixture density will increase, power will go up and the chance of detonation will decrease. Roughening port walls does not show any benefits on the flow bench, but will show improvements at the race track.

The typical procedure for creating intake port wall roughness is by using 60-grit cartridge rolls so that scratch marks are produced perpendicular to the direction of the airflow. If the surface finish is too rough, however, it excessively increases the thickness of the boundary layer and flow is lost because the port now appears smaller.

In general, boundary-layer thickness increases as flow rates increase or as the flow path bends or turns. At typically port velocities, the boundary-layer is between 15 and 30 thousands of an inch thick. In certain cases, the boundary-layer can actually separate from the port walls, producing turbulent eddies or whirlpools. This reduces flow and knocks fuel out of suspension. In most cases, this can be prevented or minimized by contouring cross sectional shapes through bends as previously mentioned.



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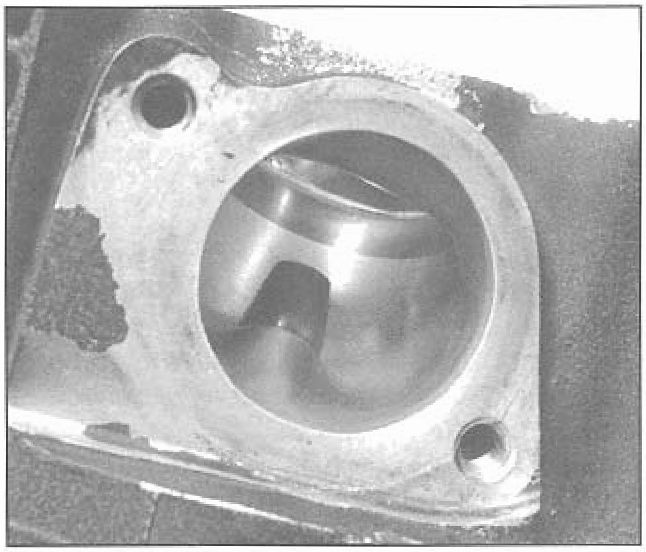

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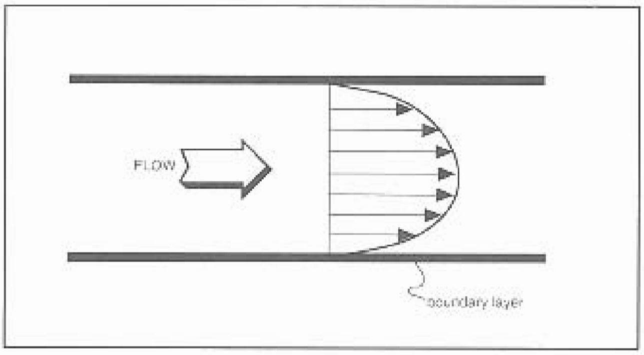
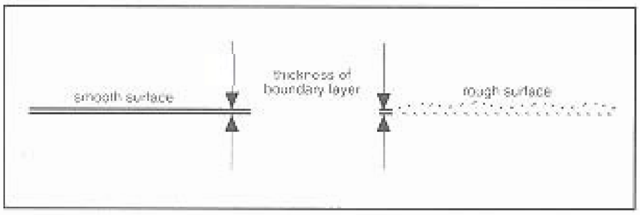


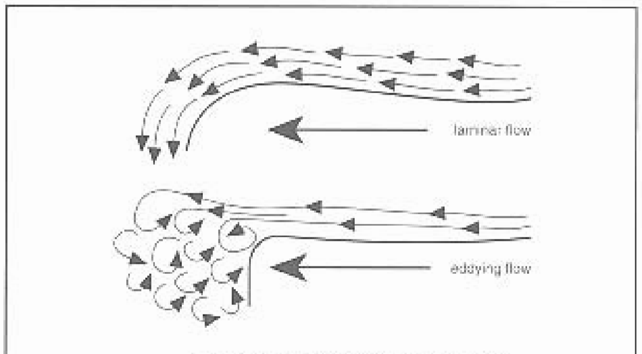
A boundary layer is created between the moving air/fuel charge and the surface of the intake tract. Fuel tends to puddle and air is stagnant in this area. This results in an inactive area whose thickness is determined by the surface finish and other factors.



A smooth surface finish produces a thin boundary layer, while a coarse surface increases boundary layer thickness. Head porting specialists have different schools of thought, but it is generally accepted that a rougher surface promotes fuel suspension in the airstream. A smooth surface helps dry-flow exhaust conditions.

Sharp edges that protrude into the port passage significantly affect port flow. These edges, such as the sharp-edged factory undercuts below stock valve seats, induce boundary-layer eddies. The eddies or disturbances can reduce effective flow and encourage air/fuel separation. The same is true of a miss-matched intake manifold or port opening.

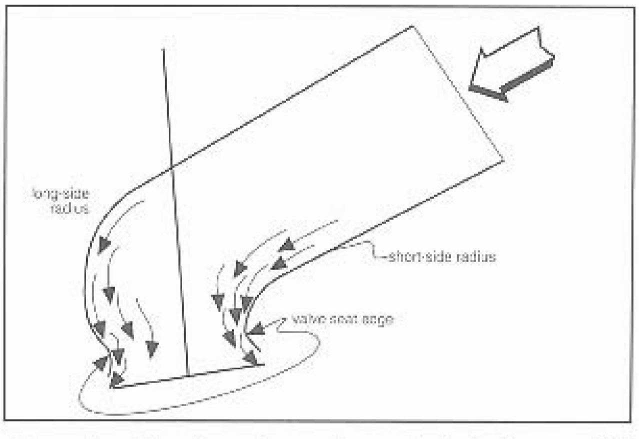
The major "bottleneck" in the port is the valve, so the closer the valve is to the modifications the more important the changes become. Because air moves at its greatest speed past the valve seat, this area of head modification is very critical. Modifications made within one inch above or below the valve seat can make or break the performance given by the head. Airflow gained by modifications in this area are more important than similar increases gained at the carburetor. The valve bowl and seat areas are the most critical areas of the port and yield the most horsepower for every dollar spent. A good multi-angle valve job, proper blending of the seat to the bowl and combustion chamber, along with correct contouring of the short-side radius are the most important modifications to the port.



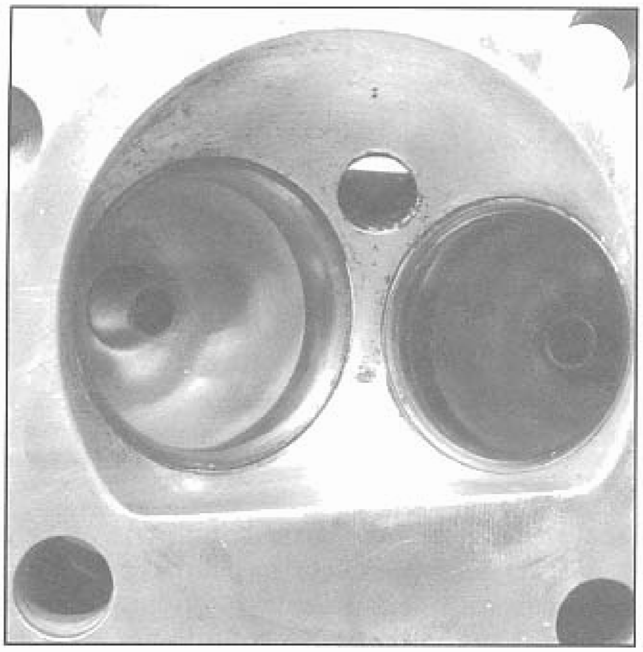
SMOOTH AND TURBULENT FLOW

Eddying, which is caused by turbulence reduces kinetic energy and decrease airflow. Radical directional changes and sharp edges in a port create unwanted eddies.

At any given moment, the point at which the most resistance to flow occurs may transfer from one point in the induction tract to another. At low and medium valve lifts, most restriction attributed to limited flow area is located between the seat in the head and edge of the valve. As valve lift increases, limitations at this point are reduced. At high valve lifts, flow restrictions are more attributed to port shape than valve seat area. As the valve moves up and down, it spends more time nearer the seat than it does at full lift. At low lift levels, flow efficiencies of the seat have significant importance because this area determines the amount of air that will flow through at a given moment. Keep in mind that a valve reaches half lift position twice during each cycle, but full lift position only once.



The underside of a valve seat creates turbulence, which reduces airflow. The seat must be radiused into the port's bowl area. Shovel cylinder heads are particularly prone to over hanging seats.



The intake port illustrates a venturi formed between the port's bowl area and the valve seat. Note the elimination of all sharp edges under the valve seat and how the seat is radiused into the bowl. The flat side of the combustion chamber must not shroud airflow around the valves and the chamber roof should be blended into the valve seat area. Blending the chamber reduces compression, but more importantly it increases airflow. Photo courtesy of Carl's Speed Shop.

Correct intake port size is directly related to the engine's displacement and operating range. A port is normally enlarged in areas where airflow is the greatest. Enlarging a port in a low flow area reduces air speed even more in that area and enhances fuel dropout from the airstream. As a result, a port that is too large is worse than one too small.

Generally speaking, the straighter a port is designed, the better it is. The Big Twin's engine dictates a port design that bends a lot. This means that as the air/fuel mixture passes through the carburetor, it still has a tortuous path into the cylinder. To increase the port's flow capability, a change involving port direction and port elevation is often required rather than a change in port size. For example, sometimes it can be helpful to angle a port's centerline to bias the flow in a more favorable direction before entry into the chamber. Both the Evolution and Shovelhead can gain flow by raising their ports to straighten out the flow path. For example, some aftermarket Evolution heads have the intake port raised about 3/8 inch. Also, Shovelheads can benefit from internal welding, particularly the exhaust port.

Exhaust Port

For the most part, most of the concepts and principles applied to intake port will also be valid in the exhaust port. However, there are some differences because intake and exhaust flows have different forces affecting them.

Airflow is quite simple until you try to change its direction. In the case of intake ports, we have air and fuel coming in. With exhaust ports, there are hot gases going out under quite a bit of pressure. Exhaust gases are traveling about 300 feet per second at about 1300 to 1500 degrees Fahrenheit. Hotter gas means lower pressure and greater velocity. It also means a less efficient combustion chamber because more heat is escaping out the exhaust. Heat radiated away from the engine is lost power. Keeping the exhaust temperature inside the system for as long as possible is important.

The flow relationship between the intake and exhaust ports must be considered during cylinder head modifications. When measuring airflow on a head with nothing bolted to it, the exhaust port ideally should flow between 80 and 90 percent of the intake flow for a race application. By comparison, a stock Evolution Big Twin exhaust port flows between 70 and 75 percent of the intake port. When the intake manifold, carb and exhaust system are bolted on the head, the flow should be equal on both sides. For example, if an Evolution Big Twin intake port flows 155 cubic foot per minute (cfm), it probably ends up flowing 145 cfm after the carb and manifold are attached. On the exhaust side, the head probably flows about 140 cfm. With the header in place, the flow ends up at about 145 cfm. As a result, both the intake and exhaust sides are equal when the engine is running.

As the exhaust valve opens, the high pressure in the cylinder starts to move toward the relatively lower pressure in the exhaust system. Additionally, with the added scavenging of the header, the cylinder can be exhausted with a rather small port. Opening up the sides of the port where the guide protrudes into the port allows the escaping gases to split smoothly around the guide.

Also, the shape of the exhaust valve's underside has major importance because it offers a significant potential for flow increase. The exhaust valve has a great deal of combustion pressure pushing on it when it opens, so it is best not to use a valve any larger than is necessary.

Just as with the intake port, exhaust port velocity is also important. A slow moving exhaust port can reduce the range of the engine's power band and hurt power at low rpm. If the exhaust port has a large velocity differential between its top and bottom, slow moving exhaust gases on the bottom of the port can back flow (reversion) into the chamber and reduce power.

Since the exhaust port is not dealing with a wet-flow environment, there is no concern about air/fuel separation. Therefore, exhaust port wall texture takes on a different condition than the intake's. As a result, the exhaust port surface should be polished as smooth as possible so that it produces a small boundary-layer and the least possible friction. Also, as stated earlier, the internal combustion engine is a heat pump, and heat is power. Heat radiated away from the engine is lost power. Polishing the exhaust port not only reduces friction, but also helps retain more heat, which increases the velocity of the exiting exhaust gases. Polishing also helps to minimize carbon buildup in the port.

Carburetion

The carburetor is one of the most misunderstood elements of the engine, but since it supplies fuel it is the basic source of horsepower. Because the carburetor is readily accessible and has many tunable elements, individuals are frequently adjusting, modifying or replacing it on the Big Twin. When modifying a stock displacement Big Twin, the carburetor is usually one of the first four components that is replaced — the other three being the exhaust system, cam and ignition module.

The carb starts out just like a straight piece of pipe. As air flows through the pipe, it must draw fuel into the airstream from the fuel reservoir or float bowl. The element that allows fuel to be drawn into the airstream is called a venturi. A venturi is an obstruction in the pipe that starts out as a sharp narrowing of the pipe passage and then gradually widens to the original size of the pipe. Air speeds up as it moves through the venturi, just as it does over the top of an airplane wing. The fast-moving air creates a pressure drop that causes fuel to be drawn out of the float bowl, up the spray tube and into the airstrip. The stronger the pressure drop, the more accurately the carburetor will react to small changes in airflow.

The venturi allows the carburetor to meter fuel *almost* proportionally to airflow. Therefore, fuel flow roughly triples as airflow triples. A uniform air/fuel ratio throughout a wide rpm range is paramount to a wide torque band. Not only must a carburetor meter fuel, it must also assist the fuel to break up into very small droplets, which is called atomization. The greater the fuel is atomized, the greater the possibility is for complete combustion and the greater the potential for increased power.

For a given carburetor and engine rpm, the intake tract airspeed is governed by: engine displacement, throttle position and the venturi's diameter. A smaller venturi creates faster airspeeds and a stronger booster signal. All things being equal, the stronger the booster signal, the better performance will be at low rpm. A high

A carburetor with high airflow takes priority over air filtration on a drag bike. Note the large fuel line and deflector shield on the end of the velocity stack. Photo courtesy of Larry Smith/ Handcrafted American Racing Motorcycles.

flowing, large venturi carburetor with a weaker booster signal works best on large displacement or high rpm engines. If used on a small, low rpm engine, horsepower and throttle response will suffer. Conversely, using a small venturi carburetor on a large displacement engine will provide great throttle response, but the engine will be down in horsepower in the higher rpm ranges.

Most carburetors available today for the Big Twin are the fixed venturi butterfly type. This type carburetor uses a butterfly and shaft mechanism to regulate the airflow through the venturi. These carburetors usually have three adjustments to regulate the air/fuel mixture throughout the rpm range. They also typically have a large venturi to offset flow losses resulting from turbulence generated as the airstream hits the butterfly and shaft.

Another carburetor design is the variable venturi constant velocity type. It includes a slide that regulates the size of the venturi and a butterfly and shaft that regulates the airflow. The engine's demand for air regulates the position of the slide, which in turn regulates the size of the venturi. At low speed the venturi is small for maximum booster signal and throttle response, while at high speed it is large to satisfy the engine's greater demand for air. This type of carburetor provides the combined benefits of a small and large fixed venturi carb and usually offers excellent throttle response.

A third type of carburetor is a smoothbore radial flat slide. It has a very clean surface through its bore, hence the name smoothbore. Instead of using a butterfly and shaft, it uses a flat slide that is raised by the throttle cable. This arrangement gives it the benefits of a variable venturi carburetor without the negative effects of a butterfly and shaft type. Its thin flat slide design results in a greater pressure drop over the main fuel system, which results in quicker fuel delivery as the slide is opened. This provides superior throttle response and tractability at low rpm while providing excellent peak power.

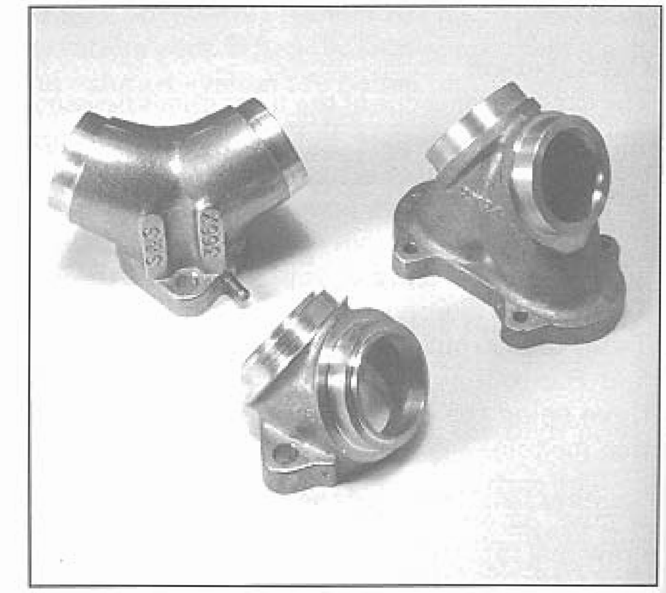
The ideal high-performance carburetor has high flow capability, maintains high air velocity with a minimum amount of turbulence, includes sufficient adjustments for precise fuel metering over the entire rpm range and allows for easy jetting changes. A carburetor for street use should also allow for easy installation and not inhibit the rider's leg position.

Intake Manifold

The intake manifold essentially is an extension of the cylinder head's port. The Big Twin's intake manifold is designed as a short, stubby "Y" shape. As air leaves the carburetor, the airstream must turn 45 degrees in the manifold and then travel down the port. In stock form, the manifold is restrictive. This is especially true with the Shovelhead and early Evolution manifolds.

Air passing through a turbulent carburetor has a difficult time negotiating the 45 degree bend and moving into the port because the manifold's short-side radius is too small. To correct the problem, it is necessary to increase the radius at the 45 degree turn to give a less abrupt change and help straighten the flow. Also, the Shovelhead, early Evolution and some aftermarket manifolds have a sharp ridge that divides the two sides of the manifold. This ridge also causes a lot of turbulence and needs to be radiused.

Some stock manifolds do not have enough material to increase the short-side radius sufficiently, so an aluminum extension needs to be



S&S intake manifolds: Top left, Evolution; bottom, Shovelhead; top right, manifold for S&S Two-Throat carburetor. Notice the sharp directional change the intake charge must make with all Big Twin manifolds. Photo courtesy of S&S Cycle.

welded to its carburetor end. Performance manifolds are now available that eliminate the need to modify the stock unit. These manifolds come radiused and ported and include a larger plenum area for enhanced cylinder filling capability.

Due to a common manifold feeding both cylinders of the 45-degree V-Twin, the lengths of time between the pulses in the intake runner vary extensively. The front intake valve opens and then closes, which is followed in rapid sequence by the opening and closing of the rear intake valve. Then there is a long pause until the sequence again starts with the front intake valve. As a result, the time between the front cylinder filling and the rear cylinder filling is less than the time between the rear cylinder filling and the front filling. The odd 315 degrees to 405 degrees firing pattern results in the rear cylinder benefiting from the air/fuel flow that has already been established in the intake runner during the filling of the front cylinder.

The longer time between the rear intake and the front intake allows the air/fuel mixture to slow down more and lose more inertia than between the front intake and the rear intake. Since air is lighter than fuel, air/fuel separation occurs and this causes the front cylinder to end up with a lean air/fuel charge. For this reason, a racing spark plug check frequently indicates the front cylinder is running leaner and hotter than the rear. Machining the intake manifold (on its carburetor mounting surface from one to three degrees slanted toward the front cylinder) sometimes corrects the problem.

Some performance manifolds incorporate a large plenum chamber designed to add volume to the manifold. With the odd 315 degrees to 405 degrees firing pattern of the Big Twin, performance is sometimes improved by using a large plenum area. In general, a manifold with a large plenum area helps midrange and top-end performance, but sometimes reduces low rpm power due to diminished fuel metering caused by a reduced intake "signal" at the carburetor venturi. The size of the plenum chamber can affect throttle response, torque, horsepower and the rpm at which the torque and horsepower are produced.

Intake Tract Tuning

Acoustical wave tuning, which is commonly referred to as "ram tuning," can be used to increase cylinder filling. The ram tuning technique was originally pioneered by Chrysler corporation on their Ramcharger engines and is used today by many engine builders.

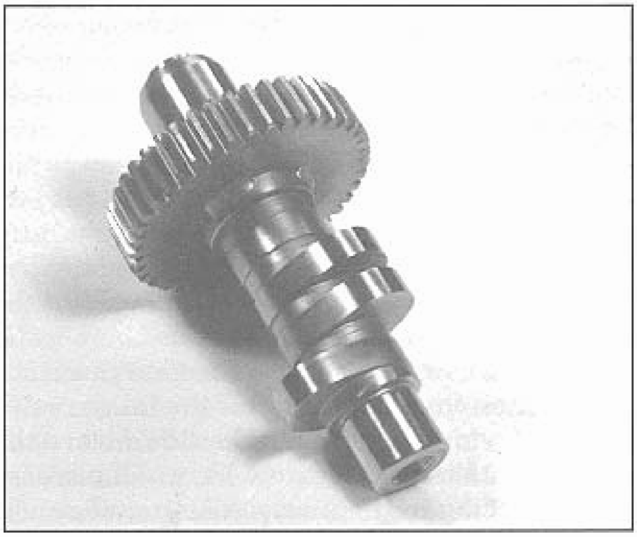
Strong energy pulses are created in the intake tract by the opening and closing of the valve. As the intake valve opens, the air/fuel mixture starts flowing into the cylinder at high speed. A short time after the completion of the intake stroke, the intake valve closes and the fast moving air bounces off the valve, creating an energy pulse. The energy pulse travels down the intake port, through the carburetor and to the atmosphere at approximately 1100 feet per second. When the energy pulse exits the carburetor and hits the atmosphere, a negative (low) pressure pulse is created and reflected back up the intake tract toward the combustion chamber. If the negative energy pulse is timed to reach the combustion chamber just after the intake valve opens, it can help start intake flow before the piston's intake stroke starts. This will increase cylinder filling and improve volumetric efficiency.

If room permits, the intake tract can be adjusted in length to optimize the time at which the negative wave reaches the combustion chamber. Its length can be adjusted to increase performance over a short 1500 to 2000 rpm range, although performance outside the range will be compromised. Increasing the tract's length helps low rpm power, while shortening it helps at high rpm. An Evolution's intake tract (when measured from the valve seat through the middle of the tract to the edge of the carb at the air cleaner mounting) is approximately 9-1/2 inches long. An engine tuned for street use in the 2500 to 3500 rpm range needs a tract length between 24 and 32-inches. A large displacement stroker engine tuned for the 4500 to 6500 rpm range requires between a 14 and 18-inch intake system length. Obviously, it's impractical to accommodate a street engine's requirements, so in certain applications intake tuning is limited.

Camshafts

If the cylinder head is considered the heart and soul of the internal combustion engine, then the camshaft is its personality. The camshaft is the primary component used to tailor the torque and horsepower curve of an engine.

Camshaft design is not an exact science. The best cam for an engine is determined by a number of interrelated factors that are difficult to measure. As a result, determining the best cam is accomplished more often by trial-and-error methods than any other. The only sure way to determine the cam's capability is to test it.



The Big Twin engine uses one camshaft with four lobes. The cam is driven by the pinion gear, which is located on the flywheel's right side main shaft.

Every engine design has inherently different overall camshaft requirements. One reason for this is that a camshaft needs to be matched to the specific flow characteristics of the cylinder head. Cam lift and duration values are affected by these characteristics. Specific engine combinations also determine camshaft design as does engine displacement, rod-to-stroke ratio and rpm range.

The poppet valve is the basic type of valve used in practically all four-cycle internal-combustion engines. Raising and lowering poppet valves is an extremely slow and inefficient method to control the operation of intake and exhaust operations. The ideal situation would include the ability to instantly open and close the ports and to advance or retard the timing of these events. This capability would optimize the torque and horsepower throughout the engine's operating range. Unfortunately, the camshaft and valvetrain are unable to instantly open or close valves. As a result, overcoming these restrictions is best dealt with by first understanding the different valve timing events.

Intake, compression, power and exhaust are the four cycles of the four-stroke engine and the concept for filling a cylinder during these cycles is rather simple. First, open the intake valve as the piston nears top-dead-center (TDC). Then, as the descending piston creates a vacuum, the air/fuel mixture from the intake tract will fill the cylinder. When the piston reaches bottom-dead-center (BDC) close the valve so compression can take place. Although simple in concept, conditions are actually more complex under actual conditions. As the piston descends, the air/fuel mixture lags behind and takes a moment to get moving. Once moving, it has stored up kinetic energy to help fill the cylinder. By keeping the intake valve open a short time after BDC the kinetic energy can be used to ram fill the cylinder a little more, even though the piston is now rising. The greater cylinder filling creates higher combustion pressure, thereby creating greater torque and power.

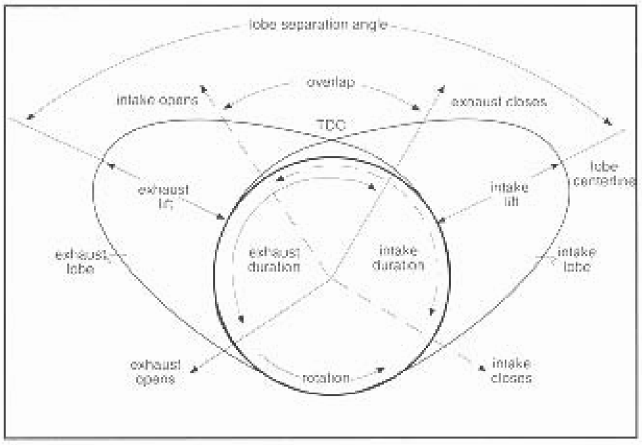
The faster the engine spins, the faster the piston moves. This increases the intake charge velocity and the charge's kinetic energy or ram effect. The greater the ram effect, the longer the intake valve can be left open after BDC and still derive benefits from the ram effect. Keeping the intake valve open even longer after BDC improves cylinder filling at high rpm, However, at lower speeds, intake velocity is too low for ram effect and the rising piston steadily builds pressure in the cylinder and overpowers the induction inertia. This forces the intake flow to reverse and move backward into the manifold. The result is reduced cylinder fill, torque and power at low rpm. Delayed closing of the intake valve moves the point of peak torque higher, while reducing torque at lower speeds.

Of all the valve opening and closing events, intake closing is considered the most important because it signals the changeover from the induction cycle to the compression cycle. The second most important event is exhaust opening. Together, these two events determine the time the cylinder is sealed (with both valves closed) and cylinder pressure can push against the piston. In part, opening and closing cam timing events are directly related to the piston's acceleration rate, which in turn is related to engine's stroke and connecting rod length. Ideally, the intake valve should reach at least half of its total lift by the time the piston reaches its maximum acceleration rate away from TDC. This is the point where the piston generates its strongest signal to the intake tract.

A cam is designed to lift a valve starting at a specified crankshaft position, lifts it to a specified height above its seat and then closes it at another timing position. Cam specifications provide opening and closing points for the intake and exhaust valves, total duration (total time valve is open) and total lift (distance valve is lifted off the seat). Some cam specifications provide two timing values for duration. The first, called advertised duration, includes the time required for the lifter to move past the gradual lobe clearance ramps that take the slack out the valve train before the valve is opened. At this time, the lifter is moving very slowly and checking cam timing at this point does not give realistic values. The second duration value is taken at either 0.020-inch or 0.053-inch lifter (tappet) lift and provides a more accurate reading because lifter movement can be measured more precisely at these points.

Tom Sifton, the founder of Sifton cams, started measuring cam timing events at 0.053-inch tappet lift and now most aftermarket cams are measured at this point. The lower the lift at which the opening and closing events are measured, the longer the duration will be. Be aware that the automotive industry's de facto standard for measuring cams is at 0.050-inch tappet lift.

In many cases, cam specifications also include two additional values: overlap and lobe separation angle (LSA). Overlap is the period when the piston is near TDC at the end of the exhaust stroke and the beginning of the intake stroke. At this point both valves are opened at the same time — the exhaust valve is just about to close while the intake is just starting to opening. Having both valves open at the same time uses the moving mass of exhaust flow to scavenge the cylinder of residual combustion gases and initiate intake flow, even before the piston has started to move down on its intake stroke. The early induction flow improves cylinder filling and increases horsepower at high rpm, but reduces power at low rpm when ram effect is low. Short duration cams designed for low rpm use generally have short overlap periods while long duration cams used for high rpm racing typically have longer overlap periods.



Lobe separation angle is the distance measured in cam degrees between the point of maximum lift of the intake lobe and maximum lift of the exhaust lobe for the same cylinder. A cam's LSA has a direct bearing on the amount of overlap for a given duration. Camshafts having the same amount of lift and duration can be ground with different lobe separation angles that results in different amounts of overlap.

Increasing a cam's lift produces additional power, but without narrowing the engine's power band as much as a longer duration cam would. Since most high lift cams lift the valve faster, stronger valve springs are often required. However, too much spring pressure can quickly wear out valves and valve guides, besides generating higher levels of horsepower robbing internal friction. Also, a heavy valvetrain or too little spring pressure allows valve float to take place and this leads to reduced power and possible major engine damage.

As duration increases, the engine's power band is moved higher in the rpm range. As a general rule, for each ten degree change in duration (measured at 0.053-inch tappet lift), the power band moves up or down about 500 rpm. More lift can increase torque and horsepower, but the speed at which the valve lifts is a factor. Although two different cams may have the same lift, duration and timing specifications, they may provide totally different performance. One reason for this is that the specifications do not show the rate of lift. To get a truer comparison, you need to plot a cam's profile to determine its lift curve. The faster the lift, the more area there will be under the lift curve, therefore the more power. You may have heard of the term "cheater cam." Racers running in some classes are required to use cams with stock specifications. "Cheater cams" retain all the stock specifications except one — the rate of lift.

Many factors must be considered when selecting a cam profile. These factors include basic considerations such as intended engine use, engine displacement and rpm range. Additionally, the cam should be properly matched to: cylinder head flow, compression ratio, exhaust system, carburetor size, stroke length, rod length and the weight of the bike.

Keep in mind that no cam can have maximum effectiveness for greater than about a 1500 rpm range. When in doubt, it is always better to under cam an engine than over cam it. In general, for good low rpm torque choose a shorter duration cam with as high a lift as practical, consistent with head flow characteristics and valve spring pressures. For a light weight, high rpm bike, where maximum performance is the objective, a long duration and high lift profile cam is the norm.

Also, a four-valve head design uncovers flow area faster than a two-valve design. This is because two valves have more perimeter for a given area than a single valve and low-lift flow is greater when there is more perimeter. As a result, four-valve heads require less duration and lift for a given combination than a two-valve design. Furthermore, since four-valve heads have smaller and lighter valves, cam acceleration can be quicker, which results in the valves reaching maximum lift sooner.

Don't forget that since the cam is affected by many variables, the only sure way for determining the best profile is through trial-and-error track testing.

Exhaust System

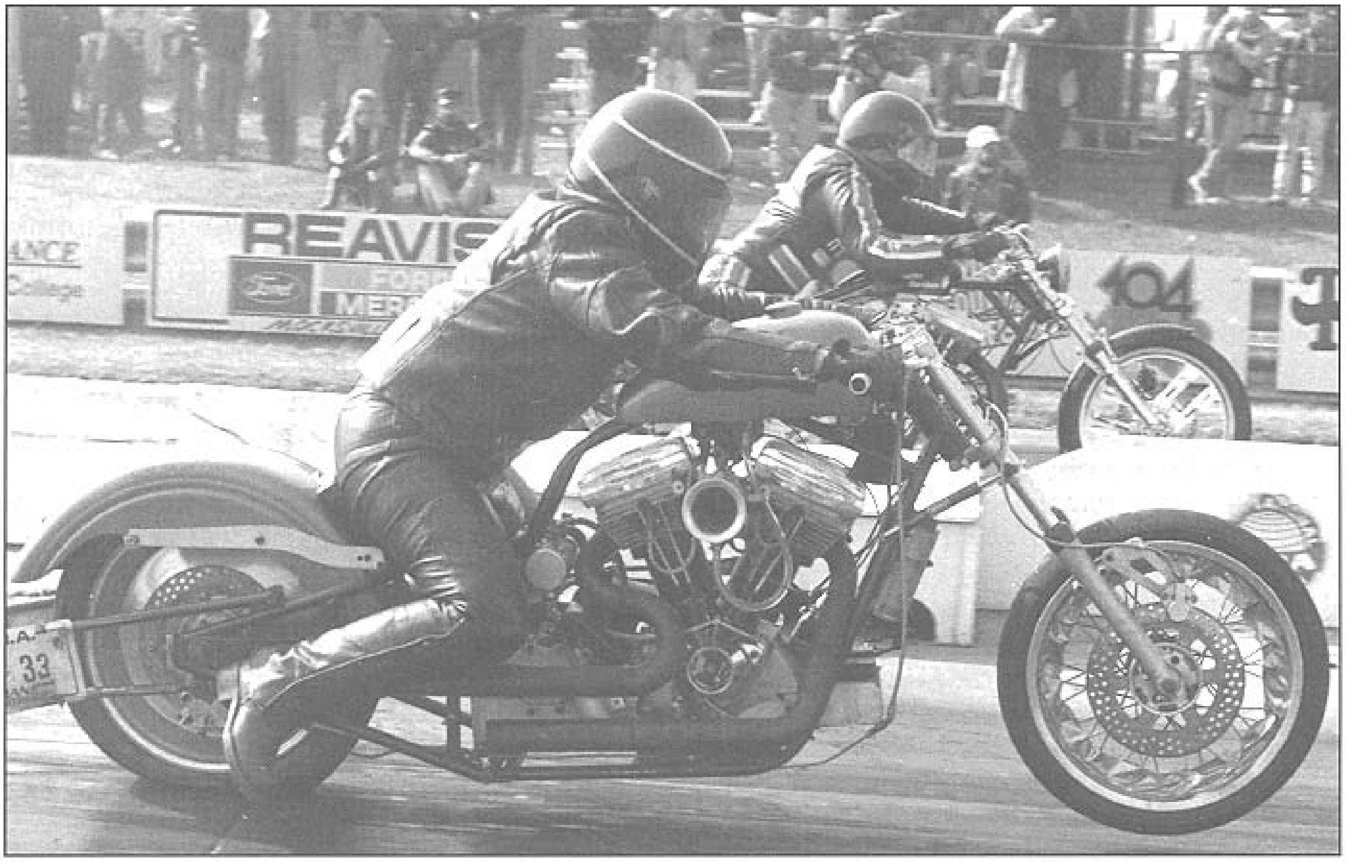
Previously it was explained how an engine is like an air pump and how to cram more air/fuel mixture into the pump. As it turns out, moving the burned gases out of the pump is just as important to making big power as is filling the pump.

The function of the engine's exhaust system is to clear the engine of as much combustion residue as possible. The less efficiently this is accomplished, the greater the air/fuel mixture contamination. Consequently, both power and fuel efficiency are reduced. An efficient exhaust system is a fundamental ingredient for any high horsepower engine. The diameter and length of the pipes, along with the overall design of the system, all contribute to how efficiently the exhaust system scavenges. Up to a point, the higher the scavenging capability, the higher the power.

During normal operating conditions, when the exhaust valve opens the piston has to pump the exhaust gases out into approximately 15 pounds per square inch of atmosphere. A well-designed exhaust system employs two concepts during operation: inertial scavenging and wave scavenging. The use of these concepts generates "free" horsepower by overcoming the atmospheric pressure in the exhaust tract.

When the exhaust valve opens, hot exhaust gases escape out of the combustion chamber, move past the exhaust valve and flow down the exhaust pipe at about 300 feet per second. The moving gases include inertia, which is the ability of a moving object to resist any change in its motion by another force. As the escaping gases move down the pipe, a low pressure area is created behind the gases by the inertia. This helps pull more air/fuel mixture from the intake port into the combustion chamber during the overlap period when the intake and exhaust valves are both open. It also helps scavenge combustion residue out of the chamber. This effect is known as inertia scavenging and results in higher volumetric efficiency and more power.

The second exhaust scavenging concept uses acoustical tuning to help purge exhaust gases. Acoustical tuning or wave scavenging uses a pulse of low pressure that is generated by the high pressure gases exiting the exhaust pipe to help pull additional air/fuel mixture into the cylinder during valve overlap. When the exhaust valve first opens, the "burst" of escaping gases creates a positive energy pulse. The energy pulse



Straight pipes offer maximum exhaust scavenging, which is needed for lots of horsepower and low elapsed times in drag racing events. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

travels through the exhaust pipe to the atmosphere at the speed of sound. Based on the temperatures and pressures present in the pipe, the speed is approximately 1700 feet per second. When the energy pulse exits the pipe and hits the atmosphere, a negative (low) pressure pulse is created and reflected back up the pipe toward the combustion chamber.

If the negative pressure pulse is timed to reach the chamber when the piston is near TDC (during overlap when the intake and exhaust valves are both opened), it can not only help scavenge the remaining exhaust gases from the cylinder, but also start intake flow before the piston's downward stroke begins. Adjusting the exhaust header length to optimize the moment the negative wave reaches the combustion chamber effectively increases the valve timing and maximizes power. Unfortunately, wave scavenging only helps during a narrow part of the rpm band. If the wave is optimized for high engine-speed, then power and torque will be down at low speeds.

Exhaust gases travel about 300 feet per second and pressure waves between 1700 and 2100 feet per second. It is important to know that pipe diameter changes gas velocity and pressure, while pipe length changes pulsations. Smaller diameter pipes work more efficiently at low rpm due to higher gas inertia, but they reduce power at high rpm due to volume restrictions. Larger diameter pipes benefit top-end performance, but hurt power at lower rpm because of reduced inertia. Short length pipes provide maximum horsepower at high rpm while long pipes yield maximum performance at lower rpm. If the pipe is too large in diameter and too long, the exhaust cools and affects wave pulsing. Also, the shape and the number of bends in the pipe can increase unwanted pressure.

Heat retention in the exhaust system is also very important. After heat has performed its function in the combustion chamber, it travels out the exhaust port to the pipe. As the gases cool, they lose velocity and their scavenging effect is reduced. Insulating the exhaust system helps keep the gases at the highest possible temperature, resulting in greater velocity, greater pressure drop and higher efficiency.

Cam specifications generally determine the rpm of maximum torque and horsepower. The combination of cam and intake system also determine the timing and magnitude of pressure pulses. The stronger the negative pulses, the more fuel that will flow in the intake system. Therefore, pipe diameters and lengths must always be tuned to the camshaft.

As an example, to establish optimum pipe diameter and length, the engine's rpm range must first be determined. Let's assume a drag engine leaves the starting line at 5000 rpm, but drops down to 4500 rpm when the tire hooks up. It is up-shifted to the next gear at 6500 rpm and then drops down to 4500 rpm after the up-shift. For maximum performance, this engine should have its pipe length optimized at about 5500 rpm because this is half way between the low point of 4500 rpm and the high point of 6500 rpm.

It can be difficult to figure out the perfect exhaust system, since so many variables come into play. The best racers never stop trial-and-error testing. As a rule, the higher the engine's rpm range, the larger the pipe's diameter and the shorter the length should be. Conversely, the lower the rpm range, the smaller the pipe diameter and the longer the length should be. In recent years, some top racers have experienced excellent performance by coupling a large diameter pipe with a long length. For maximum performance, all pipes should be exactly the same length, flow the same amount of air, and run at the same temperature. In the end, all exhaust systems are proven through trial-and-error track testing.

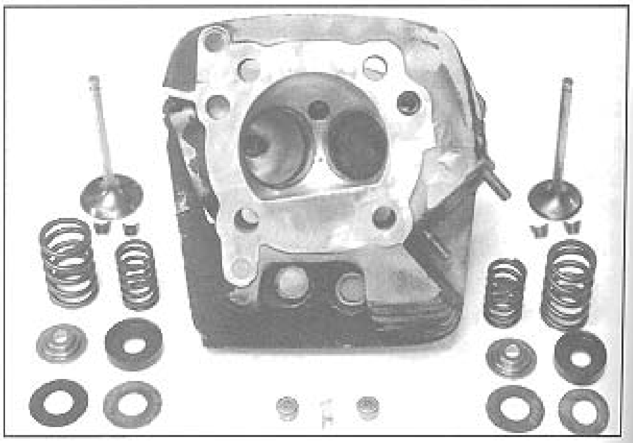
COMBUSTION EFFICIENCY

The fundamental factors for better cylinder filling and scavenging were previously discussed. However, they are only part of the puzzle for extracting maximum horsepower from an engine. The more heat produced from the air/fuel mixture, the more pressure that is produced on the piston and the greater the horsepower extracted from the engine. Generating more heat can be accomplished not only through higher cylinder filling, but also by burning a greater percentage of the air/fuel mixture. So, not only is mixture *quantity* important, but so is *quality*.

The key to improving mixture quality and burning a greater percentage of the intake charge is proper design of the ports, combustion chamber and piston dome. Many important aspects of ports already were discussed, so lets concentrate on the chamber.

Combustion Chambers

Good combustion chamber design can optimize combustion and thermal efficiencies. This results in the maximum extraction of heat energy from the air/fuel mixture. The combustion area consists of the roof (combustion chamber), floor (piston dome) and walls (cylinder surface). The shapes of the roof and the floor, along with the relationship between them are critical to encouraging full atomization of the air/fuel mixture and smooth flame travel. Poor combustion area design promotes air/fuel separation, wide variations in the air/fuel ratio during combustion and less than maximum cylinder filling. This results in lost power and the potential for detonation.



Stock D-shape Evolution combustion chamber has a 90 degree squish ledge on one side and the spark plug on the other. This chamber is much smaller and more turbulent than the Shovelhead chamber, resulting in faster and more complete combustion. This head was reworked by Carl's speed shop. Photo courtesy of Carl's Speed Shop.

The ideal combustion area includes a relatively flat, compact space for rapid combustion and provides for short, unobstructed flame travel. It also allows for high compression without restricting airflow or flame travel and it encourages turbulence for maximum air/fuel atomiza-

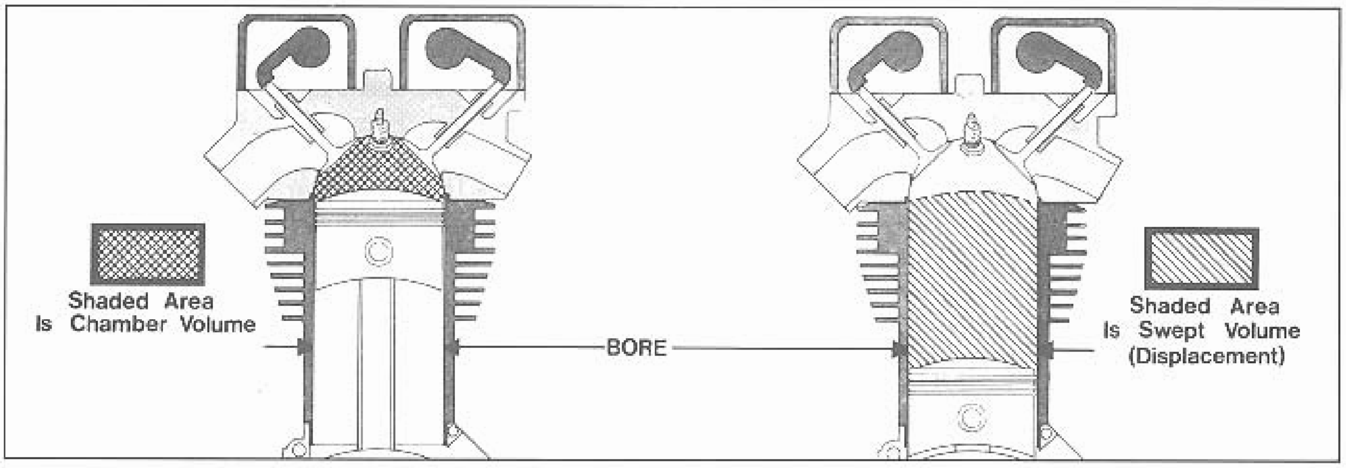


Figure 2.3 *A hemispherical chamber like the Shovelhead's results in a deep chamber that requires a large dome piston for high compression. It also requires the combustion flame to travel a long distance. A high dome piston interferes with flame propagation and robs heat from the combustion process. Illustration courtesy of S&S Cycle.*

tion and homogenization. Obtaining all of these characteristics in a combustion chamber is difficult to accomplish, especially without restricting airflow near the valves. As a result, most chamber designs are a compromise between achieving good airflow, high turbulence, short and unrestricted flame travel, and high compression.

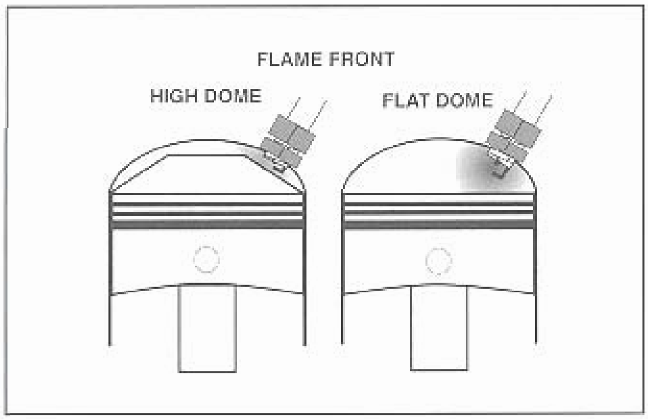
A flat or shallow chamber exposes less surface area to the 4500 degree flame it contains, so it absorbs less heat. A shorter flame front requires less time to consume the air/fuel mixture and exposes less heat to the cylinder walls since less ignition advance is required. Consequently, less heat energy is absorbed by the engine components and more is expended to drive the piston downward. Lowering the combustion chamber roof also allows the chamber floor (read: piston dome) to be lowered. Lowering the piston dome reduces the dome area and exposes less surface area to heat. This allows the cylinder pressure to be concentrated upon a smaller area. It also reduces heat loss to the various engine components and allows greater pressure to drive the piston down. Additionally, it permits more rapid and uniform combustion and potentially greater cylinder filling at high rpm's.

The ideal chamber is easier to achieve with an over-square engine design — one with a larger bore than stroke. It is very difficult for two-valve head designs like the Evolution's and particularly the Shovelhead's to achieve the above characteristics. To get two large valves into a chamber for a given cylinder size requires the valves to set at a very steep angle to each other. This angle is referred to as the valve's included angle. The larger the valves for a given bore diameter, the greater the included angle must be and consequently the deeper the combustion chamber design.

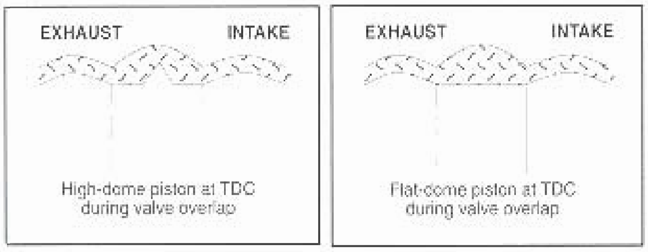
Shovelheads and iron Sportsters have deep combustion chambers with two valves set at a 90-degree angle to each other (45-degree included angle). These deep, hemispherical chambers were patterned after pre-World II aircraft engines because they allow the use of large valve sizes for a given bore diameter.

A deep combustion chamber requires a high dome piston to achieve a high compression ratio, but a high dome can severely interfere with flame travel. The interference slows the combustion process and requires more spark advance for a complete intake charge burn. But initiating the combustion process very early with a large amount of ignition advance exposes more cylinder wall area to the combustion heat. This results in reduced thermal efficiency and lost power. Furthermore, a high piston dome adds more dome surface area that absorbs more heat than one with a flatter dome. This reduces the energy in the combustion gases that would normally be used to increase the Brake Mean Effective Pressure (BMEP) on the piston. BMEP is an engineering term that refers to the power output of a given displacement engine. The resulting loss in thermal efficiency reduces power and increases cylinder head, piston and oil temperatures.

A high piston dome that is designed for a high mechanical compression ratio also interferes with cylinder filling during the valve overlap period at high rpm. The reduced cylinder fill results in less BMEP; therefore horsepower and torque are lost. However, the same piston may give good performance at lower rpm because of its high mechanical compression and its ability to provide adequate cylinder filling without needing to rely on the valve overlap period to enhance filling.



A high piston dome can interfere with flame propagation and slow the combustion process. This can lead to less efficient combustion and the engine may require more ignition advance.



A high dome piston can interfere with the intake flow during valve overlap and reduce volumetric efficiency. It also can increase heating of the intake charge, resulting in less efficient cylinder filling.

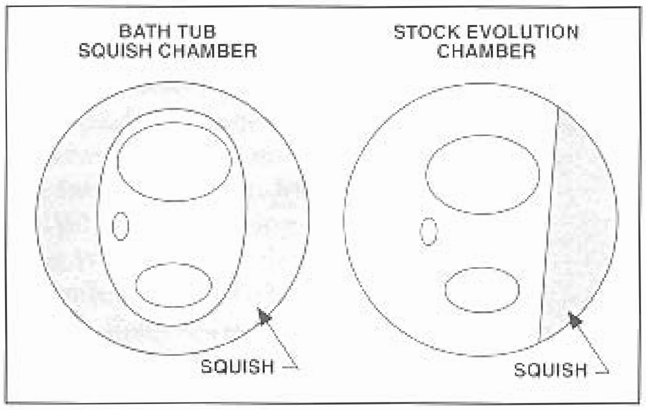
Unlike the Shovelhead and iron Sportster, the Evolution engine has a relatively shallow chamber because its valves are set at only 58 degrees to each other or a 29 degree included angle. Without a doubt, the Evolution chamber is not as flat or compact as some of the foreign "multis," but it is a big improvement over its predecessors. This chamber incorporates more desirable characteristics such as a shallow roof, flat-topped piston, good squish band and shorter flame travel.

Many engine builders spend most of their time chasing big horsepower numbers by building large displacement engines — it's called the "bigger is better theory." Given all other things are equal, a good big engine will beat a good small engine. However, many large engines never reach their maximum potential power output. Somewhere along the line it is forgotten that for a given engine displacement, the efficiency of the combustion chamber is one of the two major factors that govern the engine's power — the other factor is the engine's ability to breathe. Using an efficient combustion chamber design that includes a good piston dome shape, good spark plug location and an effective squish band will provide enormous power gains. Additionally, evening the temperature throughout the chamber to within about 50 degrees F. from the intake to the exhaust side can also help power output. When building your engine, remember that an efficient combustion chamber design can be the difference between a winning big engine and just any big engine.

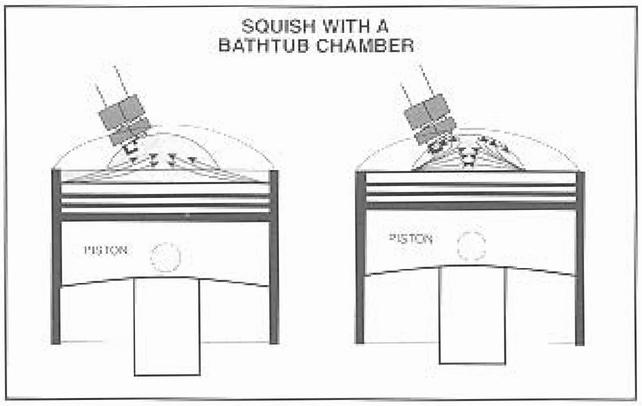
Squish Band

The squish band (sometimes referred to more appropriately as quench area) is one of the most common methods used to enhance air/fuel turbulence before combustion. Areas where parts of the cylinder head and piston dome come within proximity at TDC are referred to as squish bands. As the piston nears TDC, mixture trapped between the head's squish area and piston squirts out at high velocity. This increases turbulence, which promotes better air/fuel atomization and reduces the potential for detonation. Most turbulence is generated during the intake stroke as the intake charge is rushing through the narrow valve opening and at the end of the compression stroke where the flow is forced into the combustion chamber from the squish band area.

The Evolution squish band is located in the head directly opposite the spark plug side. It is shaped as a 90 degree ledge and is positioned above the piston to create the thin squish zone. For the Evolution engine, the ideal squish zone is between 0.025 and 0.040-inch. Anything larger, diminishes the squish effect. The Evolution chamber design can be improved by filling in the chamber's sides through welding and then recontouring its shape similar to a bathtub.



A bathtub chamber increases squish area for better turbulence and unshrouds the flow area around the valve heads. For a street engine, piston deck height should be .030 to .040-inch below the cylinder head squish ledge.



Chamber turbulence is generated as the air/fuel mixture is compressed between the head's squish band and rising piston. The compact chamber and flat dome piston provide a short and unobstructed path for flame propagation.

This improves squish, reduces potential detonation and increases torque. It also can increase airflow by reducing valve shrouding.

The deep chambered Shovelheads and iron Sportsters create very little air/fuel turbulence because they have little or no squish area. Unfortunately, this is one of the inherent characteristics of a hemispherical chamber design. The lack of turbulence reduces the amount of fuel mixture burned and reduces horsepower.

When building an engine, especially the Shovelhead, it's critical to do whatever is necessary to create as much squish area as possible. When doing so, be sure to minimize the nooks and crannies where mixture is excessively cooled and goes unburned. The cylinder spigot sticking up into the Shovelhead's chamber can create dead areas that reduce the percentage of fuel burned.

Flame Propagation

An efficient combustion chamber minimizes the distance between the spark plug and the farthest part of the combustion area. A centrally located spark plug is the best design for a single plug head, but two-valve heads usually do not have enough room for a spark plug to fit between the valves, so the plug resides on one side of the chamber. This requires a long flame travel that needs lots of ignition spark advance and increases the potential for detonation. This situation is sometimes made worse when the piston dome is raised because the dome can interfere with mixture turbulence and flame travel.

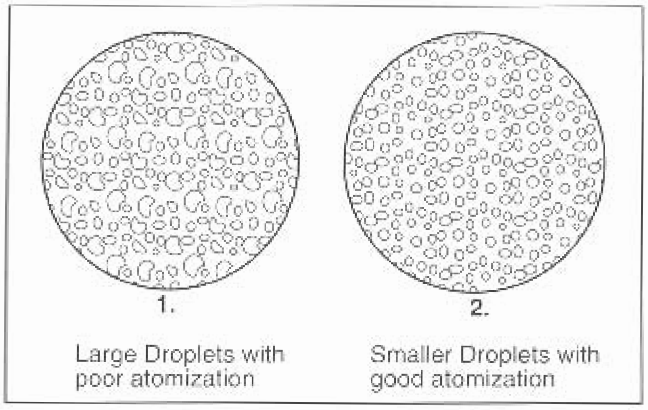
Adding a second spark plug to the opposite side of the chamber minimizes these problems because it reduces the distance the flame needs to travel and requires approximately five to eight degrees less ignition advance. Dual plug heads generally allow the use of higher compression ratios with a given gasoline octane rating, or they allow the use of lower gasoline octane with a given compression ratio. Shovelheads and iron Sportsters can greatly benefit from dual plug heads. Big Twin Evolution heads, with their compact chamber and better mixture turbulence have less need for dual plugs, yet they still can benefit from the second plug.

There are a number of evolutionary differences between Evolution and pre-Evolution engines. However, any major performance difference between the two can be directly attributed to the Evolution's more efficient combustion chamber design.

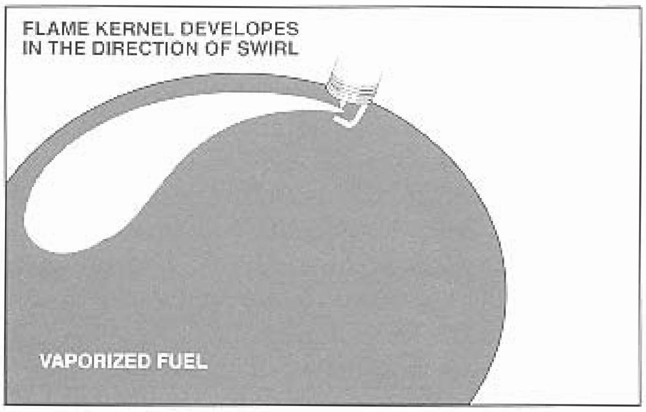
Combustion Process

Briefly, we have discussed how liquid fuel will not burn and that some amount of oxygen needs to be added for combustion. As a result, it becomes necessary to atomize the mixture into a vapor or "fog" like substance so the fuel particles are dispersed into the smallest sizes possible. This provides a more rapid and complete burn because small droplets burn faster than larger ones.

The actual combustion process begins at the outer portions of the droplet and progresses inward. The larger the droplet, the longer the time required to burn it. When there is little turbulence, the flame front occurs in a smooth,

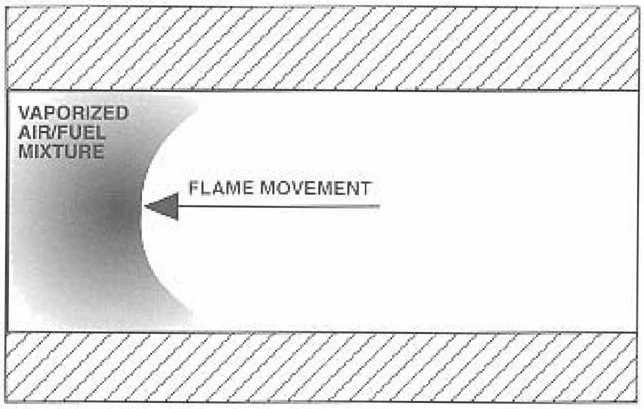


Large fuel droplets take longer to burn than small droplets and they burn more completely. Increased combustion chamber turbulence improves fuel atomization and reduces the size of fuel particles.

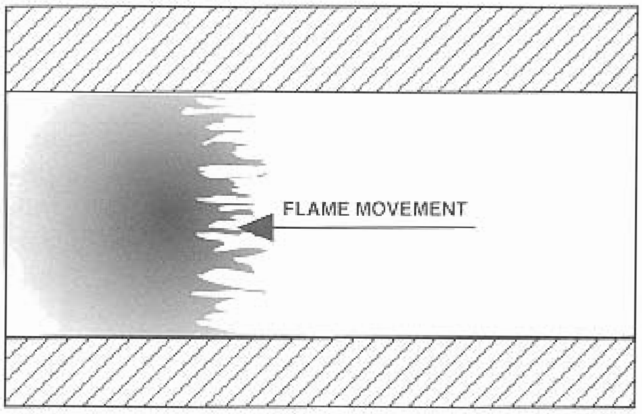


The turbulent fuel charge is swirling through the spark plug gap as the plug fires. This creates a long, radiused flame kernel. High turbulence develops a larger flame kernel, which improves combustion of the fuel charge.

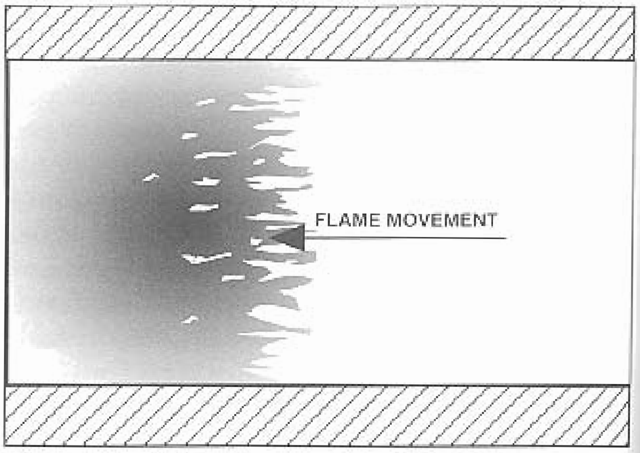
laminar fashion. In general, atomization of the fuel mixture will increase as turbulence in the cylinder increases. High atomization, which is partly generated by a high level of turbulence, causes small parts of the flame front to separate from the main flame kernel and penetrate the fuel charge. This results in a shaggy flame front that has more surface area and consumes the fuel charge more completely and quickly. When high fuel atomization is combined with proper ignition timing, the result is more combustion near TDC and greater cylinder pressure and more torque. Remember, the efficiency of fuel atomization is a crucial factor in burning the maximum amount of fuel in the shortest time.



Low combustion chamber turbulence allows a smooth laminar flame front to develop. This exposes less air/fuel mixture to the flame and slows the combustion process.



High turbulence creates a jagged and bushy flame front, which exposes more air/fuel mixture to the flame and increases combustion.



With a bushy flame, small sections of the flame break away and penetrate into the body of the air/fuel mixture, thereby enhancing combustion.

As discussed earlier, combustion chamber induced turbulence is a common method used to enhanced atomization. Poor chamber design may provide fuel atomization, but it also can cause the mixture to break into "clumps" where one area of the chamber tends to be rich, while another area is lean. As a result, flame travel is erratic, which makes it difficult to optimize ignition spark advance and run with high compression ratios. High dome pistons generally contribute toward this condition.

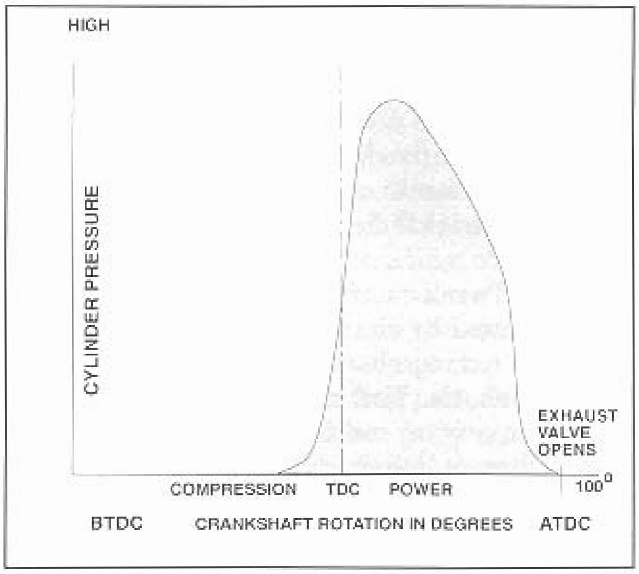
When considering combustion chamber design, remember to view the combustion space as a combination of the chamber (roof) and the piston (floor). Also, keep in mind that the chamber roof, floor and the space between them not only contribute to mixture turbulence, but also determine how evenly the mixture is dispersed.

Brake Specific Fuel Consumption (BSFC)

Engines with good fuel vaporization and even mixture dispersion generate more power and require less fuel because less fuel is dumped out the exhaust and burned in the pipe. Also, their exhaust temperatures are generally lower since there is better thermal efficiency in the combustion chamber and the engine is making more power per each pound of fuel. Engines with efficient fuel vaporization typically have a lower value for brake specific fuel consumption.

Brake specific fuel consumption (BSFC) is a measure of how efficiently an engine is converting fuel to heat or power. The lower the BSFC number, the more efficient the engine is. A stock Big Twin's BSFC typically ranges between .60 and .75-pound per horsepower per hour throughout its rpm curve. This means that it takes between .60 and .75-pound of fuel to make one horsepower continuously for one hour. The more even the BSFC is throughout the rpm curve, the better. Due to port design and combustion chamber shape, some engines have a lower BSFC than others. Very efficient engines generally have a BSFC ranging between .45 and .49. A good running Big Twin engine has a BSFC between .50 and .55-pound per horsepower.

A more homogenous air/fuel mixture burns faster and produces more even peak cylinder pressure than a poorly blended mixture. This broadens the pressure curve in the cylinder, reduces detonation causing pressure "spikes" and allows for higher compression ratios. A broader pressure curve extends peak pressure at maximum piston velocity, which is where the greatest mechanical advantage is realized. Remember, effective cylinder pressure (BMEP) is what pushes on the piston to create torque and thereby horsepower.



Greater turbulence increases atomization and speeds combustion. This allows less ignition advance to be used, while still maintaining maximum cylinder pressure at the optimum time after TDC. Reduced ignition advance means less combustion pressure is pushing on the piston during the compression stroke and more work is used to generate power.

Another point to consider refers to the term "combustion area" (chamber roof, floor and cylinder walls) which was defined earlier. The combustion area includes the entire surface area over which the combustion flame is spread during the burning of an air/fuel charge. It is important to keep the total surface of the combustion area as small as possible because the greater the surface area exposed to combustion, the lower the combustion temperature and the lower the BMEP.

Compression Ratio

Almost every performance enhancement involves raising cylinder pressure on the compression stroke or releasing it on the exhaust stroke. In general, increasing cylinder pressure generates a bigger bang that results in higher torque and horsepower output. Raising an engine's mechanical compression ratio is one method for building more cylinder pressure and it is of vital importance to the engine's power and torque output.

Higher compression primarily increases low and midrange torque, enhances throttle response and increases gas mileage. Raising compression can increase torque because the cylinder's brake mean effective pressure (BMEP) is increased. However, peak cylinder pressure is limited by detonation. Take note that an engine's detonation level is not fixed, but can move either up or down with different quality of fuel, ignition timing, volumetric efficiency and atmospheric conditions.

A Big-Twin's mechanical compression ratio can be altered by changing any combination of piston design, combustion chamber volume or cylinder length. The mechanical compression ratio of an engine can be mathematically computed when cylinder volume and combustion chamber volume are known. Although mechanical compression ratio is most commonly discussed, it is actually dynamic (effective) compression that is realized by a running engine.

An engine's dynamic compression ratio is controlled in part by such things as cam duration, valve overlap and rpm level. A long duration cam with a long overlap period can essentially raise an engine's volumetric efficiency (the amount of cylinder fill) at high rpm. The result is an increase in the dynamic compression ratio.

In general, changing the compression ratio from 8:1 to 9:1 will have a greater effect than going from 13:1 to 14:1. In some cases, engines that do not respond to tuning changes may be helped by an increase in compression ratio. Since fuel burn time is reduced by higher pressure, higher compression may require less ignition advance.

To make reasonable power with a Big Twin engine, the mechanical compression ratio should be no less than 9:1. Maximum effort engines using racing gasoline are typically between 13:1 and 15:1. Street engines are usually limited to between 9.5:1 and 10:1 due to low 92 octane unleaded pump gas. When increasing the engine's mechanical compression ratio by changing piston dome shape, it is important that the piston does not restrict cylinder breathing or interfere with spark plug flame travel.

In general, increasing squish increases the air/fuel mixture turbulence and this allows the use of higher compression ratios. Also, long duration, high overlap cams require higher than stock mechanical compression ratios to increase cylinder pressure and torque at low rpm.

In summary, the ideal combustion chamber space includes a relatively shallow roof (chamber) with a small included valve angle and a flat chamber floor (piston) for a minimum of bumps, angles and edges. It also should include a centrally located spark plug for short, unobstructed flame travel and sufficient squish area to produce a high degree of turbulence for excellent air/fuel atomization and homogenization. Nooks and crannies where fuel gets trapped and goes unburned should be minimized or preferably eliminated and the overall chamber design should concentrate the compressed mixture toward the spark plug for a faster, more complete burn.

REDUCING FRICTIONAL LOSSES

There are numerous approaches to making a motorcycle more performant. One approach is to increase the engine's power. Another more subtle method is to reduce frictional losses, both inside and outside the engine.

Frictional losses, sometimes referred to as frictional horsepower, are defined as the sum of all mechanical losses in the engine and the drivetrain. The most notable engine parts contributing to these losses include: rings and pistons, valvetrain, flywheels, clutch, bearings, seals, chains, oil pumps and alternators. Oil pumping and churning losses also are included in this category.

The largest single source of engine friction loss is directly attributed to piston ring drag on the cylinder walls and accounts for approximately seventy-five percent of total friction loss. Piston rings need proper ring land gaps and end gaps to eliminate unnecessary friction. In certain applications, low tension oil-control scraper rings can reduce friction, while still providing proper seal. The finish of the cylinder bore can also make a major difference. Smooth finishes, 400 to 600-grit, work well with moly-faced rings. Furthermore, accurately machined piston pin bores and crankcase cylinder decks, along with straight connecting rods can help ensure each piston resides straight and true in its cylinder. This will help keep cylinder wall friction to a minimum while maximizing cylinder seal.

The valvetrain is another major source of friction losses. Luckily, the Big Twin includes roller cam lifters for reduced friction and cam wear. Roller rocker arms are available from several aftermarket performance companies. Rollers attached to the end of each rocker arm not only cut friction by decreasing the rubbing resistance between the arms and valve stems, but also can reduce cylinder head temperatures. Unnecessarily high valve spring pressures not only waste horsepower, but also prematurely wear out valve seats and guides while increasing oil temperature. Therefore, don't run more spring pressure than the cam profile and engine rpm require.

Reducing reciprocating and rotating mass can also reduce friction and significantly help acceleration. Friction is generated by the inertia of reciprocating parts. Inertia is the energy or force required to stop or reverse the direction of moving parts such as pistons, rods, valve lifters, pushrods, rocker arms, valves and anything else that moves up and down. Remember that inertia increases as the square of rpm, so there is four times as much inertia at 6000 rpm as there is at 3000.

Rotating parts such as cams, flywheels, clutches and sprockets sometimes can be lightened to reduce friction and help acceleration. Also, keep in mind that rotating objects with different mass distributions behave differently even though their weights are the same. Therefore, reducing diameter is as important as reducing the weight of any rotating part. Light parts will make an engine quicker revving, however, for the engine to last to the finish, durability and reliability never must be compromised.

The Big Twin's crankcase breather system can be the source of a significant amount of hidden horsepower. The engine's crankcase pressure varies with each piston stroke and this

Builders of maximum effort dragsters use all the winning combinations discussed here to achieve gobs of horsepower in the needed rpm band for maximum acceleration. Bob "Rat" Taft on his Evolution Pro Gas dragster. He is an AMRA Top Gas record holder, with an E.T. of 8.406 sec at 159.01 mph. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

causes internal air-pumping losses. Enlarging the crankcase oil breather opening along with timing it to open and close at precisely the correct moment can reduce air-pumping losses and maximize the efficiency of the oil scavenging system.

Oil churning in the crankcase sloshes around and interferes with the flywheels, resulting in additional friction losses. Again, oil breather system modifications offer major improvements for removing unwanted oil from the crankcase. Additionally, oil restrictors can be added to certain oil passages to limit the amount of oil circulating through the engine. Flywheel oil scrapers, which are built into the crankcases, can be checked for proper clearance. This ensures the oil is efficiently scraped from the flywheels and directed to the oil pump for quick return to the oil tank. In some cases, cylinder head oil returns can be routed to bypass the crankcase and drain directly into the oil pump pickup area. This reduces the amount of oil retained in the crankcases and provides quicker oil return to the tank. On some engines, usually Sportsters, an oil windage tray can be mounted in the crankcase under the flywheels to help keep unwanted oil off the flywheels.

Oil viscosity is not only the needed element that keeps engine parts separated from one another, but it is also the cause of major amounts of friction. For race only engines, friction can be reduced by using the thinnest petroleum based engine oils that still provide sufficient film strength. However, remember there is a thin line between retaining enough oil viscosity for engine reliability and premature engine failure. Synthetic oils have seen major improvements in recent years and can significantly cut friction losses. Many, if not most of the top racers use synthetic based oils.

The stock Big Twin drivetrain (includes the primary chain, clutch, transmission and rear chain or belt) absorbs approximately 15 to 18 percent of the engine's power. Using proper assembly techniques, along with lighter and smaller diameter parts can help minimize often overlooked friction losses in the drivetrain. Rusty, red powder spots on the rear chain links indicate a lack of lubrication and can account for up to five horsepower in friction losses.

A bike's chassis can be a significant source of friction losses. Wheel bearings need to be clean and have smooth surfaces. The front and rear wheels need to be in alignment with one another and brake calipers need to retract from the rotors.

Today, with the enormous availability of performance parts, it is relatively easy for anyone with a few dollars to build or have built for them a large displacement engine. However, many big-inch engines never perform up to their potential. They are fast when compared to stock engines, but relatively speaking they are not fast. Poor breathing intake and exhaust ports, inefficient combustion chamber design, deficient pump seal and inadequate crankcase scavenging are the most common reasons for their substandard performance. Too much cam for the engine combination is also a common problem.

After reading this chapter, you should understand that engine modifications involve more than just installing a set of long stroke flywheels, big cam and large carb if the engine's true power potential is to be reached. It's the combination of many subtle techniques that individually almost go unnoticed that allow an engine to perform up to its potential. However, when all the little things are added up, they make the difference between winning and losing. When it is all said and done, it's the accumulation of things such as proper planning, the right combination of parts, attention to detail and trial-and-error testing that differentiates one high-performance engine from another. Remember, a "happy" engine is the combination of related parts working together in harmony to their maximum potential. ❖

Chapter 3

Motor Recipes

Combinations Are the Key

Today, there are numerous possibilities for increasing the Big Twin's performance.

However, before deciding what changes to make, you first must determine the objective for your bike and how the bike will be ridden. For example, you need to decide whether the bike is for casual or serious drag racing, general touring, trailer towing, sidecar pulling, or short Friday night ego trips. Also, you need to answer for yourself, the following questions. Do you want to run at the front of the pack or are you just looking for more performance? Where in the engine's rpm band do you want the performance increase? What kind of reliability and longevity are you willing to accept? Are you willing to trade reduced gas mileage for big-inch horsepower? What size budget do you have? The answers to these and other questions will narrow down your performance options.

If you intend to race, the class you compete in governs many of your options. Once you decide upon a class, it's helpful to know approximately what engine displacement or engine combination the class' front runners are using. The point here is if you want to compete in a class where all the top-guns are running 100 plus cubic inch engines, it's a good bet you're not going to win consistently with an 80-inch engine. This is particularly true if the guy with the big inch engine has a good fundamental knowledge of engine building and racing. Now this doesn't mean that "big" is always better and it doesn't advocate copying every crazy trend that's in vogue. However, it is helpful to know what the competition is running to minimize your learning curve and to give yourself a reasonable chance to win. This is especially true if you're new to the world of Harleys and your experience is limited.

Street riders first need to determine how frequent they are willing to rebuild their engine and whether reduced gas mileage is acceptable. A properly built stroker engine can easily last 15,000 to 50,000 miles between rebuilds. In the end, engine longevity is dependent upon how well the engine is built, broken-in and maintained, along with the length of its stroke and how hard it is ridden.

For street riding, an engine with 80 or more horsepower makes it easy to pass vehicles at highway speeds on a heavy touring bike while riding two up into a head wind. Most Big Twins weigh about 175 pounds less than a touring model, so their performance is enhanced for any given engine displacement. It is relatively easy to get 80 horsepower from a stock displacement 80 cubic inch engine and without getting too exotic power can be upped to about 100 horsepower. However, the added torque and power from an 89 or 96 cubic inch stroker or big bore engine makes riding even more exhilarating.

Before discussing various engine combinations, keep in mind that this is an area where there are no hard-and-fast rules. Nothing is set in concrete and there are few black-and-white answers. Instead, there is a great deal of mushy gray areas along with exceptions for just about anything. In certain situations there is more than one correct method for accomplishing an objective, while at other times there is only one — the correct one.

LARGE VERSUS SMALL

Since the introduction of the Evolution engine, the displacement of modified engines has been on the increase. This is primarily due to performance shops having invested a great amount of time developing new parts and complete kits that make it easier to build a big engine. Furthermore, the Evolution's cylinder head and crankcase design make it easier to build large engines.

Often someone asks, how big is big and how small is small? Well, it just depends. At one time 98 cubic inches was considered big, now it's only midsize. Today, engines between 90 and 100 cubic inches are considered medium size and anything below 90 cubic inches falls into the small category. Although there is no fixed rule defining what is big, 103 to 114 cubic inches probably fits the definition as well as anything. Also, engines from about 117 inches and up fall into the ultra large category.

LARGE DISPLACEMENT ENGINES

Increasing engine displacement is accomplished through either big bore cylinders, long stroke flywheels or a combination of both methods. History has proved that Big Twins respond to an increase in stroke extremely well. For classes restricted to 85 cubic inches, a large 3-13/16" bore combined with a short 3-11/16" stroke gives a high revving engine that is proving to be very potent. Yet, for the best of both worlds and the lowest cost in the long run, increase both the engine's bore and stroke at the same time.

A large displacement engine with stock heads will run out of airflow at high rpm. This means that power gains will be limited mostly to the lower and middle rpm ranges. As a result, it is highly recommended always to include a set of professionally ported heads when displacement is significantly increased.

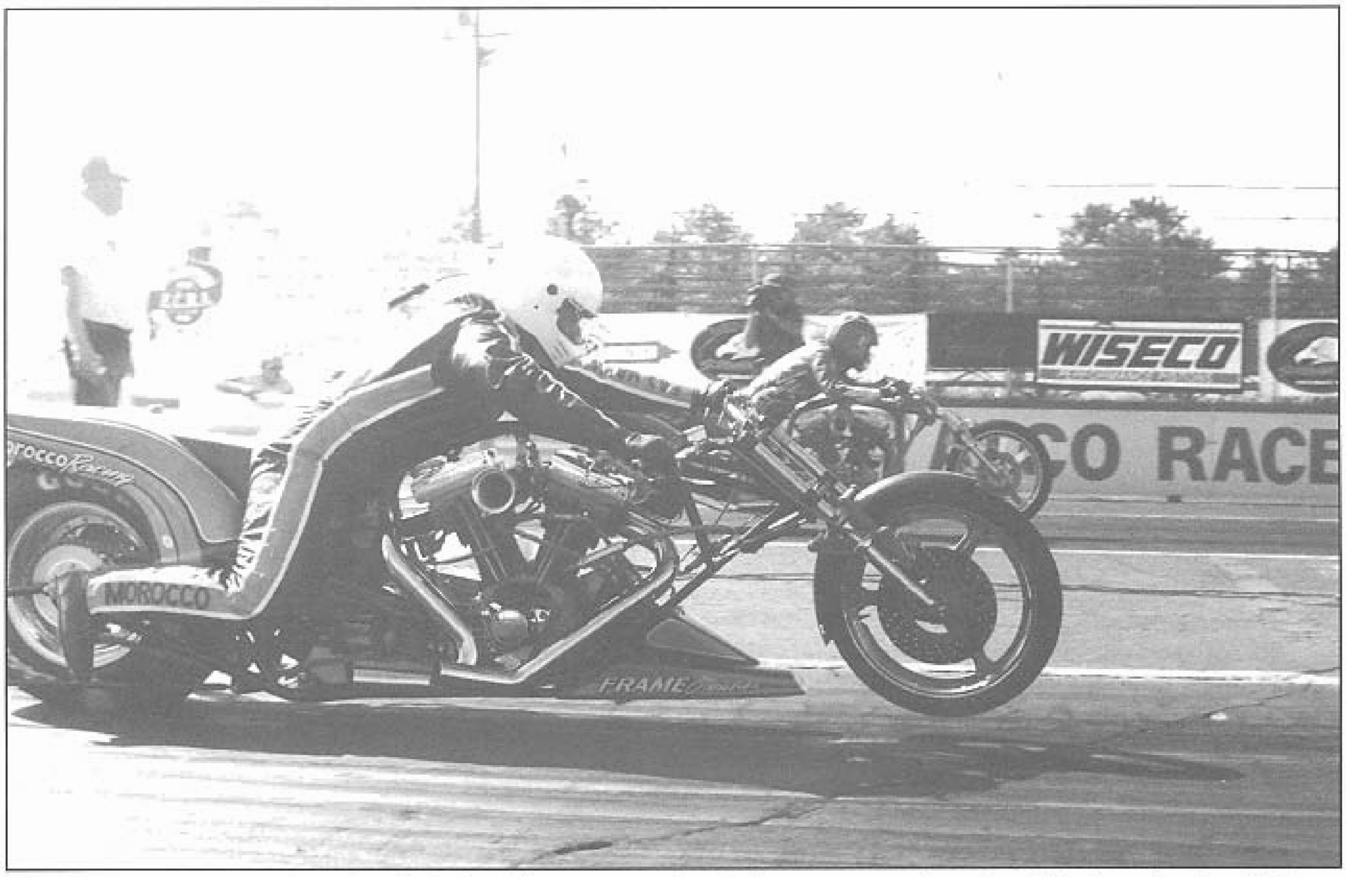
BIG TWIN ENGINE DISPLACEMENT

BORE (Inches)

STROKE (Inches)	3-7/16	3-1/2	3-5/8	3-13/16	4	4-1/4
3-31/32*	73.6	76.3	81.8	90.6	99.7	112.6
4-1/4**	78.8	81.7	87.6	97.0	106.8	120.6
4-1/2	83.5	86.5	92.8	102.7	113.1	127.7
4-5/8	85.8	88.9	95.5	105.6	116.2	131.2
4-3/4	88.1	91.3	97.9	108.5	119.4	134.8
5	92.7	96.1	103.1	114.1	125.7	141.9
5-1/8	95.1	98.6	105.8	117.0	128.8	145.4
5-1/4	97.5	101.0	108.4	119.9	132.0	149.0
5-1/2	102.1	105.8	113.5	125.6	138.2	156.0

Table 3.1 * Stock 74" = 3-7/16" Bore by 3-31/32" Stroke
** Stock 80" = 3-1/2" Bore by 4-1/4" Stroke

Regardless of your approach to achieving large displacement (long stroke, big bore or both), the engine must be removed from the frame, the crankcases need to be split apart and some amount of machining needs to be performed. Some long stroke combinations require extra long cylinders or spacer plates under the cylinders. Long valvetrain components and a wide manifold are also usually needed in these situa-



Jaime Morocco on his 149 ci Top Fuel Evolution. A few years ago Top Fuelers were averaging about 117 ci; now they're a lot bigger. Jaime has run a 7.13 E.T. at 193.0 mph. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

tions. The long parts may require clearancing between the rear cylinder rocker cover and the frame.

Furthermore, the Big Twin's transmission and clutch are marginal at best with a stock engine and only get worse when coupled to one with high power. To avoid missed shifts and gear damage, serious racers should budget the necessary dollars for blueprinting the entire transmission, including magnafluxing and backcutting the gears.

STOCK DISPLACEMENT ENGINES

If you're short on dollars or not interested in pulling the engine totally apart, then you are limited to working with either a stock 80 cubic inch (actually 81.7) Evolution engine, or a 74 or 80 cubic inch Shovelhead.

Without getting too exotic, an 80 cubic inch Evolution engine can be modified to put out about 100 horsepower. This provides about a 75 percent power improvement over stock and gives a significant increase in acceleration. And this can be accomplished while the engine remains in the bike's frame because only the heads, cylinders, cam gear cover and carburetor need to be removed. An exception to this is where a racer competing in a stock class needs to remove the engine for complete disassemble and blueprinting to stay competitive. Also, without pulling the heads or cylinders you can get between 75 and 80 horsepower for up to a 40 percent performance increase.

From a performance viewpoint, 75 or 80 horsepower is enough power to make a Big Twin interesting and fun to ride. And the fun only gets better from this point on up. Depending on how exotic the engine is built, as a general rule you can estimate about 1-1/4 to 1-1/2 horsepower per each cubic inch of displacement.

Shovelhead engines can be modified to perform about as well as an Evolution engine. The only difference is that it takes more work to get the same amount of power. Without major head modifications, Shovelhead engines are probably down at least five horsepower from an equivalent Evolution. This is primarily due to the poor air/fuel turbulence of their large hemispherical combustion chamber. However, experienced head porters can modify the chambers for better turbulence. Also, dual plugging the heads shortens flame travel, reduces the potential for detonation and allows the Shovelhead engine a better chance for reaching its maximum potential.

The key to building a good Shovelhead engine is proper combustion chamber design and not making the ports too big. The ports are actually too big in stock form. Even with its deficient combustion chamber, the Shovelhead is still a good engine and worth modifying.

STOCK BORE & STROKE DIMENSIONS

Since its inception in 1984, the Evolution Big Twin engine has shipped with a 4-1/4 inch stroke and 3-1/2 inch bore for a total of 81.7 cubic inches. Note that from here on 80 inches will be used for reference purposes. Also, the actual bore diameter is 3.498-inches, but for ease of reference 3-1/2 inches will be used.

Shovelheads shipped with a 74 cubic inch engine (3-31/32" stroke by 3-7/16" bore) from 1966 through mid 1979 and an 80-inch engine (4-1/4" stroke by 3-1/2" bore) from 1978 through 1984.

Over the years, many bore and stroke combinations have been used. Table 3.1 shows the displacements for the more popular combinations.

EVOLUTION COMBINATIONS

In general, the Evolution engine displacements that make the most sense from a performance and cost standpoint include the 89, 96, 103 and 114 cubic inch engines. Your performance objectives and riding characteristics will ultimately determine the best combination for you.

The 89 cubic inch (4-5/8" stroke x 3-1/2" bore) engine uses stock bore cylinders and a 3/8" longer stroke. This combination makes a good, small engine that allows for big bore cylinders to be added later. The 89-inch combination is a good setup for general street and touring bikes, but like other big inch engines, gas mileage will be reduced somewhat. This engine should run high 11s to low 12s at about 110 to 115 mph in the quarter mile when installed in an FX.

The 96 cubic inch (4-5/8" stroke x 3-5/8" bore) combination not only offers the most performance for the buck, but it also has proved to be one of the best performance combinations available. This engine has a 3/8" longer stroke and a 1/8" larger bore than stock, resulting in rod-to-stroke and bore-to-stroke ratios that work very well in the Big Twin. The engine's stroke length is limited to 4-5/8" so it fits into the stock frame without frame modifications. With this engine, a light weight FX that is properly setup should run low to mid 11s at about 116 to 123 mph in the quarter mile.

Although the 103 cubic inch (5" stroke x 3-5/8" bore) and 114 cubic inch (5" stroke x 3-13/16" bore) engines are frequently run on the street, realistically they are primarily for drag racing and short ego trips to the local burger hangout on the weekends. These engines have a 3/4" longer stroke and either a 1/8" or 5/16" larger bore than stock. When built properly, these engines definitely are more powerful than smaller engines, but they also are harder on parts and require more maintenance. They also may require a modified starter motor. For the most part, these engines are more for the serious street stroker or dragster classes. A light weight FX, properly setup for a street stroker class can run high 10s to low 11s in the quarter mile at about 120 to 125 mph with these engines.

Until recently, most top fuel and gas drag bikes were running 114 to 120 cubic inch engines. Now some engines are in the 150 cubic inch range.

MOTOR RECIPES

Since there are many approaches for gaining more performance with the Big Twin, the following cookbook of recipes was assembled to remove guesswork and to offer a logical, progressive path to more performance.

Although these combinations include components that work together very well, keep in mind that they do not represent the only possibilities for engine modifications and should be considered only as a baseline starting point.

EVOLUTION MOTOR RECIPES

Before various engine combinations are discussed, lets review the basic performance characteristics of a stock Big Twin Evolution engine. An average stock 80 cubic inch engine produces about 57 horsepower at 5250 rpm and 70 pound feet of torque at 3200 rpm, when measured at the rear wheel. These values are between 15 and 18 percent higher when measured at the engine's crankshaft. This engine includes a factory ignition module that limits rpm to about 5250 rpm.

The basic problems with this engine are that it has poor intake and exhaust breathing and it is rpm limited by the ignition module. As a result, all initial performance modifications are directed toward enhancing these areas. Enabling the engine to redline at approximately 6200 rpm allows it to benefit fully from modifications that increase breathing, while keeping piston speed at a reasonable 4400 feet per second.

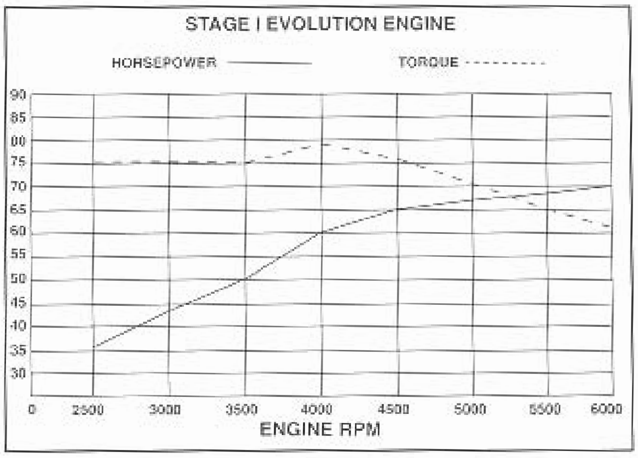
A stock 575 pound FX Big Twin setup with street tires, a 170 pound rider and 25 pounds of gas weighs in at 770 pounds. This stock combination should be able to turn high 13s or low 14s at approximately 90 to 93 mph in the quarter mile. An 800 pound "dresser" with the same rider should run low 15s at about 85 mph.

As engine horsepower increases, chassis, tire, clutch and transmission modifications must be made to keep up with the power increases. Also, keep in mind that each 50 pounds of weight will change quarter mile E.T.s approximately 1/3-second.

Take note that the following conventions are used when listing parts for each stage of modification: A check mark next to a performance part indicates the part is installed at this stage of modification. Parts without a check mark were previously installed.

Stage I Evolution
80 Cu. In. Engine
Low to Mid 13 Second Bracket
95 to 100 MPH — About 70 HP

- ✔ Performance Exhaust System
- ✔ Mikuni HS40 or S&S Super E Carburetor
- ✔ Performance Ignition Module

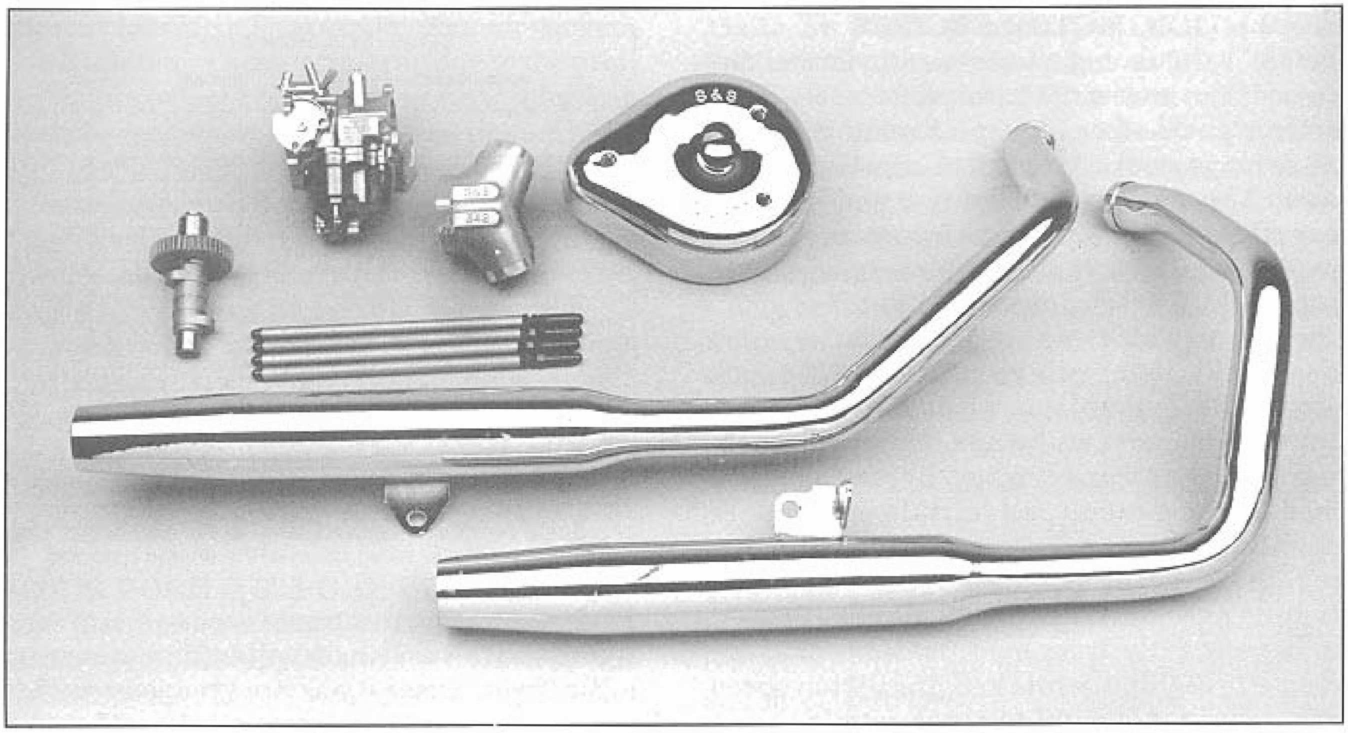


This graph represents the approximate horsepower and torque of a Stage I engine when measured at the rear wheel. Both values will be higher when taken at the engine sprocket.

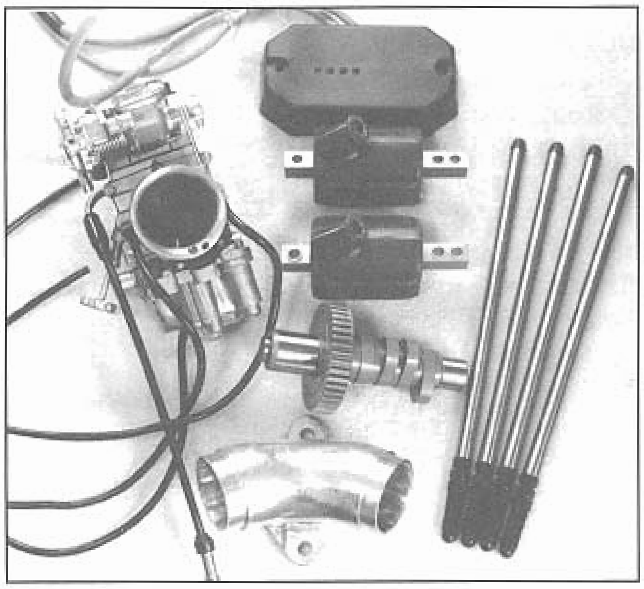
Performance enhancements need to start somewhere and these changes will definitely liven up a Big Twin. These items are grouped together because they address major deficiencies and because they are all external modifications. If you wish, you can add them individually. California engines have such a mild camshaft that you might consider changing the cam before anything else or at least including a cam with this package.

At this development stage a good exhaust system such as a set of free flowing staggered duals or a 2-into-1 system is good for about a five or six horsepower gain. For the drag strip, 40" by 1-3/4" O.D. straight pipes are a good place to start. Change the pipe length in 2" increments to fine tune the system to the engine combination. Both Harley-Davidson and SuperTrapp make performance turn-out mufflers for "dresser" models. To make decent power, the factory turn-outs need their small internal baffle removed. Since "dressers" have unequal length pipes for each cylinder, it's best to retain the crossover pipe connecting the two cylinders. Remember to enrich the carburetor jetting when installing a free flowing exhaust system.

A 40mm Mikuni flat slide or 1-7/8" (39.6mm venturi) S&S Super E carburetor adds about eight more horsepower. The Mikuni offers good bottom and midrange power while the S&S provides a strong top end. The S&S also includes a performance manifold. By using an adapter ring the Mikuni can mount directly to the 1990 and



Shown are various parts for the Stage I and Stage II engines: S&S Super E carburetor along with Carl's Speed Shop cam, pushrods and exhaust system. Photo courtesy of Carl's Speed Shop.



Shown are some additional choices for Stage I and Stage II engines: Mikuni HS40 carburetor, Branch manifold, Power Arc ignition, Dyna coils, Andrews EV3 cam and adjustable pushrods.

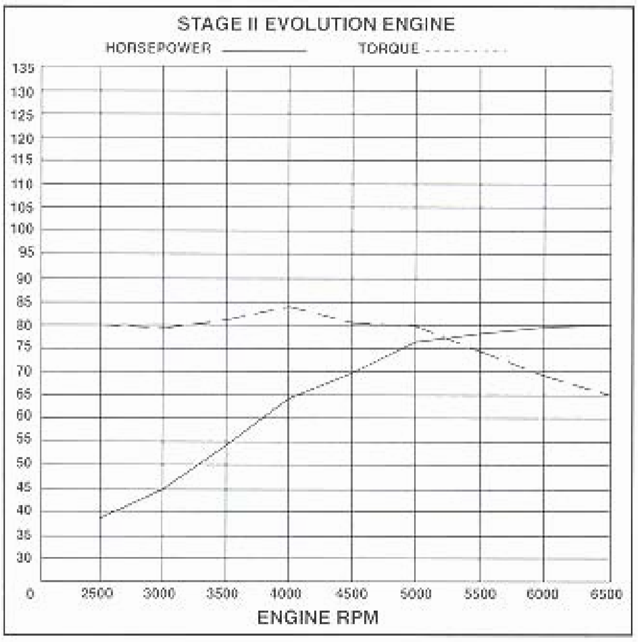
later manifold. The 1984 through 1989 Big Twins use a manifold sealed with compliance fittings and it usually ends up leaking. Branch and Bartels' offer performance manifolds that fit the Mikuni and eliminate the compliance fittings. Although the stock 1990 and later manifold eliminates the need for compliance fittings, you will gain flow by replacing it with a performance model.

The stock ignition module starts retarding the ignition timing at about 5250 rpm. This limits rpms and will hurt E.T.s by about 1/3-second at this performance level. Replacing the stock module with an Accel, Crane, Dyna, M.C. Ignition or Screamin' Eagle ignition will eliminate the problem. Any ignition with a single-fire capability provides added benefits and works especially well. The engine can now be redlined between 6000 and 6300 rpm for maximum power, although keeping the revs down to 5200 helps longevity. Also, adding a rev limiter to keep rpm below 6300 rpm is highly recommended.

This combination produces about 70 horsepower at 5800 and 79 pound feet of torque at 4000 rpm. With a good rider, it should be able to turn very low to mid 13s at 95 to 100 mph.

Stage II Evolution
80 Cu. In. Engine
Very high 12s to Low 13 Second Bracket
100 to 105 MPH — About 80 HP

- ✔ Performance Camshaft
- ✔ Chrome-moly Adjustable Pushrods
- ✔ High Output Coil and Plug Wires
- • Performance Exhaust System
- • Mikuni HS40 or S&S Super E Carburetor
- • Performance Ignition Module



This graph represents the approximate horsepower and torque of a Stage II engine when measured at the rear wheel. Both values will be higher when measured at the engine sprocket.

To get to 80 horsepower, the engine needs to breathe better and a mild cam change gets the job done.

Installing a cam into an Evolution takes more work than it does for a Shovelhead. It involves removing the exhaust pipes, cam gear cover, rocker covers, rocker arms, pushrods and lifter blocks. Numerous methods have been tried to install an Evolution cam without pulling the rocker arms. However, in the end it's easier to just bite-the-bullet and pull them off. This usually requires removing the gas tank and with some models it also requires loosening the engine mounts to lower the engine for necessary clearance.

Since the heads are not removed at this stage, consider a *bolt-in* cam that works with the stock valve springs. An Andrews EV13, EV3 or EV46; Bartels' BP 20 or BP 40; Carl's Speed Shop CM46; Crane 1-1000; Screamin' Eagle 406; Sifton 143-EV, 145-EV or 140-EV; and S&S 502 are possible choices. For heavy "dresser" models, a cam with less duration (such as an Andrews EV13, Crane 1-1100 or 1-1103, or Sifton 143-EV) will help retain power in the 2000 to 2500 rpm range where this bike is frequently ridden. Remember, it is better to under cam than over cam. When in doubt, choose the next smaller cam. A low cost S&S Hydraulic Lifter Limited Travel Kit also is worth installing because it helps keep the lifters from pumping up at high rpm.

Some early Evolutions do not have notched pistons; therefore the heads must be removed to check piston-to-valve clearance. In this case, it is recommended to consider spending extra time adjusting each cylinder's length to get the piston-to-head squish band clearance between 0.025 and 0.040-inch.

Adjustable chrome-moly pushrods are recommended instead of aluminum because they are more rigid and can be used in the future when a more radical cam may be installed.

A high-performance coil that throws a hotter spark is now recommended. Accel, Andrews, Dyna and Screamin' Eagle each offers a solution. Competition spark plug wires also should be installed. Accel, Andrews, Dyna and Taylor wires are good choices, but make sure they are compatible with the ignition.

For information regarding Stage II exhaust systems, carburetors and ignition module modifications, refer to the "Stage I Evolution" modifications since they are identical.

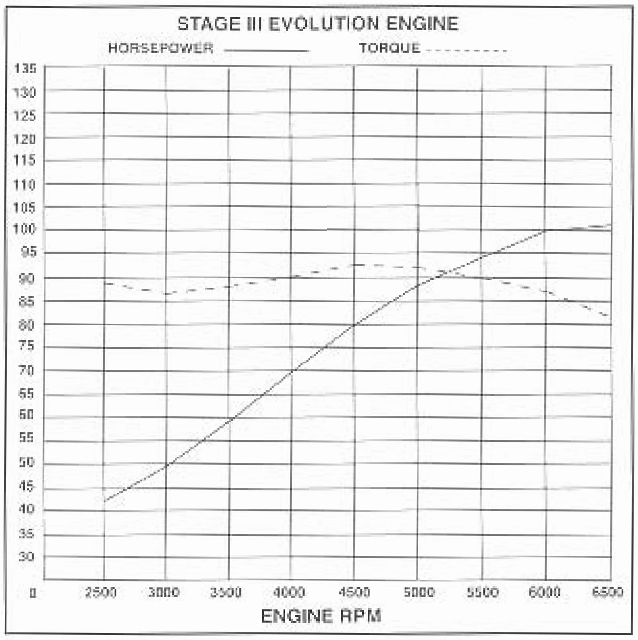
This engine combination is now producing about 80 horsepower between 5800 and 6300 rpm and about 84 pound feet of torque at 4000 rpm. A 770 pound FX bike should be good for very high 12s to low 13s at 100 to 105 mph in the quarter.

Stage III Evolution
80 Cu. In. Engine
Very high 11s to Mid 12 Second Bracket
105 to 112 MPH — About 100 HP

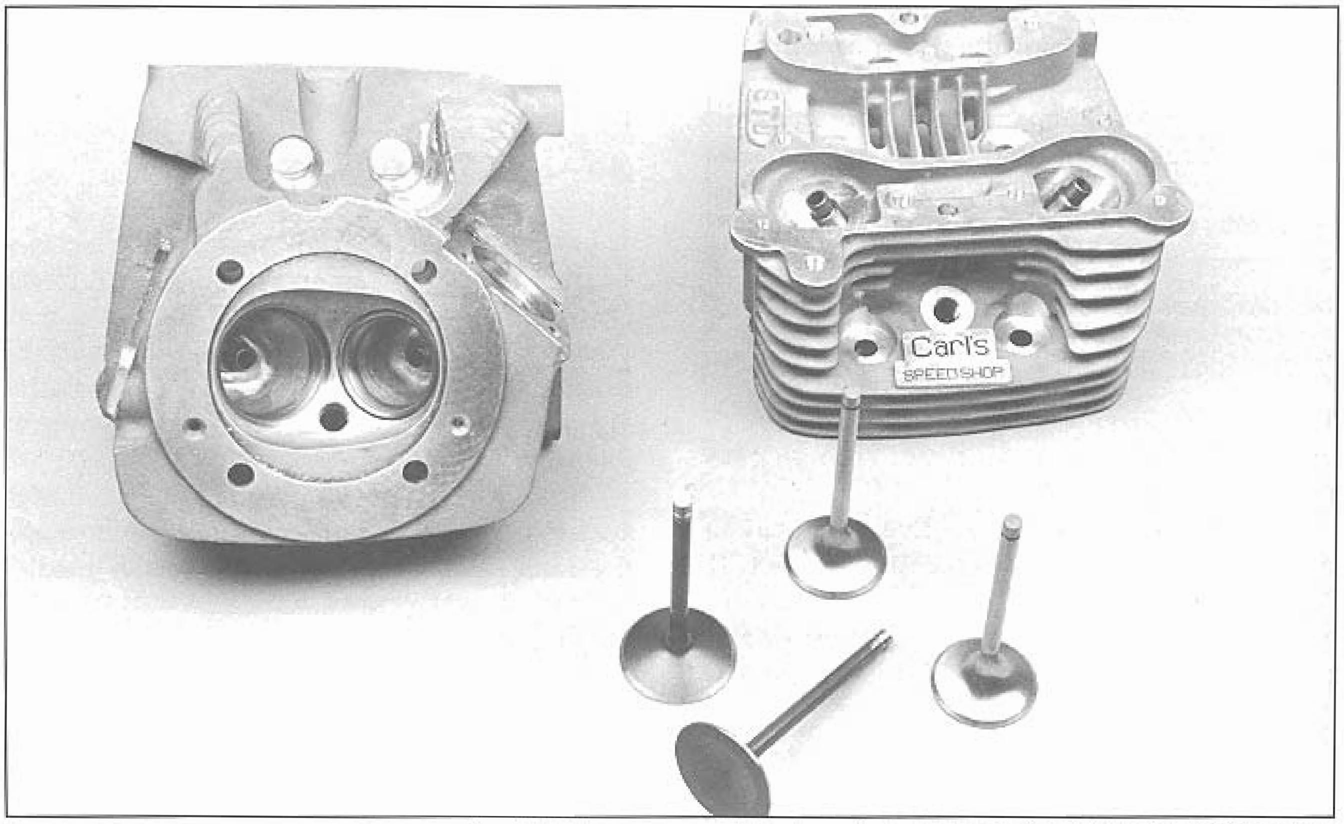
- ✔ Professionally Ported Heads
- ✔ Higher Compression
- ✔ Blueprinted Transmission
- ✔ Performance Clutch
- • Performance Exhaust System
- • Mikuni HS40 or S&S Super E Carburetor
- • Performance Ignition Module
- • Performance Camshaft
- • Chrome-moly Adjustable Pushrods
- • High Output Coil and Plug Wires

The Stage III combination includes all Stage II modifications, along with high flowing heads, higher compression and a beefed up clutch.

The next major bottleneck of this engine is the heads. A professionally ported set with performance valves, springs and collars is now in order. Modifying a head to flow big numbers is



This graph represents the approximate horsepower and torque of a "streetable" Stage III engine when measured at the rear wheel. Both will be higher when taken at the engine sprocket. This graph does not represent a maximum effort engine. With higher compression, more radical cams, maximized induction and exhaust systems, and proper engine assembly techniques, power can be increased beyond this level. Also, maximum horsepower will then be reached between 6800 and 7000 rpm.



A set of professionally ported heads can easily add 20 horsepower at this stage of development. Shown is a set of STD bathtub chamber heads reworked by Carl's Speed Shop. Photo courtesy of Carl's Speed Shop.

both a science and an art. Professional head porters usually have scrapped many heads in the process of learning how to do it right. Therefore, it is recommended to let the professionals do the job.

Before sending your heads for modification, be sure to discuss your objectives with the head porter. Explain your riding style, engine displacement, cam specifications, compression ratio, gearing, bike weight and rpm range. A top notch performance head job includes not only porting, but also oversize valves and seats, blending the seat to the port and chamber, and reworking the combustion chambers. In situations where a head needs replacement, S.T.D. Development makes an excellent performance head. Also, consider dual plugging the heads at this time. Dual plugs won't help the Evolution as much as the Shovelhead, but they are still recommend. At the very least, they will smooth out the engine.

The stock compression ratio is 8.5:1. Higher compression helps low and midrange power by replacing cylinder pressure lost due to a long duration cam. On pump gas with good squish, the engine can tolerate somewhere between 9:1 and 10:1 mechanical compression before reaching detonation levels. For good performance, run at least a 9.5:1 compression ratio. Modified heads with welded chambers or high compression pistons should easily end up with compression in this range. When running on race gas, 12.5:1 compression gives good performance as long as flame travel and airflow are not shrouded.

A good squish area for high mixture turbulence is crucial for making maximum power. Adjust the cylinder height so the piston comes within 0.025 to 0.040-inch of the head's squish band. Bartels', Custom Chrome and others make various thicknesses of cylinder base gaskets for adjusting cylinder height. Axtell, Rivera and Zipper's offer various high compression pistons.

If a bolt-in cam was previously installed, this is the time to consider a higher performance non bolt-in version. Reworked heads can usually benefit from higher valve lift than a bolt-in cam offers. If low speed torque is important, be sure the new cam's duration and intake closing timing do not kill power down low. Be careful not to add too much duration if you have a heavy weight "dresser" because power can be hurt below 3000 rpm, particularly between the 2000 and 2500 rpm range. Lighter bikes, such as the FX models, can handle more duration than heavier bikes without feeling a low rpm power loss. Refer to the "Cam and Valvetrain" chapter for more information.

Possible cams for this stage of modifications are as follows: Andrews EV35 or EV5; Carl's Speed Shop CM5; Crane 1-1001; Head Quarters HQ-23 or HQ-26; Leineweber E-2; Red Shift 575; S&S 561; Screamin' Eagle 400 or 433; and Sifton 144-EV or 141-EV.

For any bike with ported heads and situations where low end power (2000 to 3000 rpm) takes priority over the top-end (greater than about 5200 rpm), consider the following: Get the compression ratio as high as possible without incurring detonation. Also, discuss with your head porter the option of sacrificing some top end power in exchange for more at the bottom end. This can be accomplished by keeping port velocity high with a smaller than optimum port size and by retaining stock size valves.

At this performance level, the drivetrain starts to get strained. A Barnett Kevlar clutch pack helps keep the clutch from self destructing during high rpm shifts. To get good E.T.s at this power level, missed shifts and chipped gears must be eliminated. This requires disassembling the transmission and blueprinted it to specifications. The modifications should include magnafluxing and backcutting the gears for quicker shifts and less wear. Also, an Andrews close ratio first gear set will help minimize rpm reduction between first and second gear.

The stock rear tire can be replaced with sticker Avon, Dunlop, Metzeler or Michelin rubber. Finally, if you want to run a very wide rear tire or would like the extra strength of a rear drive chain, H/E/S and others offer a 24 tooth chain sprocket conversion kit.

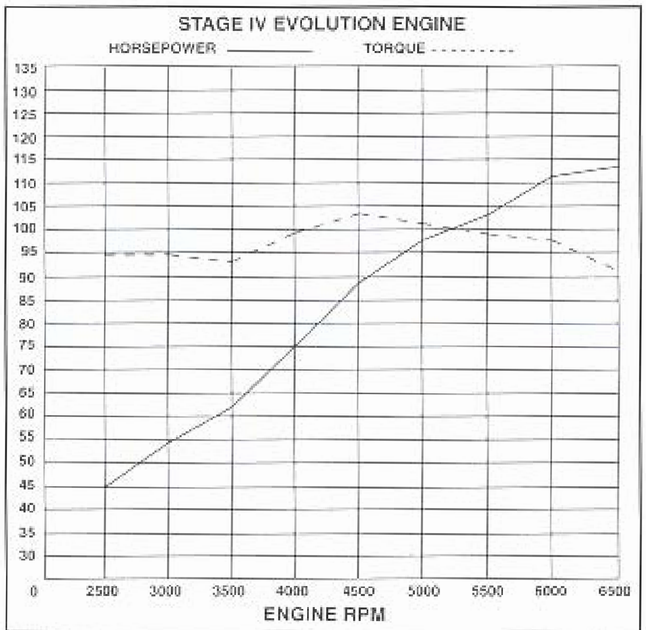
If you want to enhance an 80-inch engine any further, you're looking at complete blueprinting to race specifications and possibly more cam and carb. For the race track a more radical cam and bigger carb should help top end power, but they also will cut the bottom end.

When assembled properly, this engine will produce about 100 horsepower between 6300 and 6700 rpm and between 90 and 93 pound feet of torque between 4500 and 5000 rpm. Notice that the maximum amount of torque is slightly higher than the Stage II engine and it occurs higher in the rpm range. The major increase in power is primarily attributed to higher levels of torque over a much broader rpm range.

This engine combination should be good for very high 11s to mid 12s at 105 to 112 mph in the quarter.

Stage IV Evolution
89 Cu. In. Stroker Engine
Mid 11s to Low 12 Second Bracket
108 to 115 MPH — About 112 HP

- ✔ 4-5/8" Stroker Kit
- ✔ 9:1 to 10:1 CR — Pump gas
- ✔ Performance Camshaft
- ✔ S&S Super E or Super G Carburetor
- • Blueprinted Transmission
- • Performance Exhaust System
- • Performance Ignition Module
- • Chrome-moly Adjustable Pushrods
- • High Output Coil and Plug Wires
- • Professionally Ported Heads
- • Performance Clutch



This graph represents the approximate horsepower and torque of a "streetable" Stage IV engine when measured at the rear wheel. Horsepower and torque will be higher when measured at the engine sprocket.

Stroker flywheels and pistons are the easiest way to get to a Stage IV 89 cubic inch engine. Add 3-5/8" big bore cylinders and you have a Stage V 96 cubic inch engine. Shown is the S&S Hot Set Up 96 kit. Remember that large displacement engines need free breathing induction and exhaust systems to reach maximum potential. Photo courtesy of S&S Cycle.

This performance level requires complete removal of the engine from the frame. Stroker flywheels with a 3/8" longer stroke are installed along with stroker pistons. For pump gas, the mechanical compression ratio should be kept between 9:1 and 10:1. With race gas, 10.5:1 to 12.5:1 gives better performance. S&S or Truett & Osborn can supply the flywheels, while Axtell and S&S have pistons.

This is the time to rebuild the connecting rods or install a set of heavy duty S&S rods. The engine should be balanced and blueprinted to correct specifications and the crankcase oil breather system should be modified and timed to open and close for maximum oil scavenging.

This engine can use more cam than the stage III version. An Andrews EV5 or EV57; Carl's Speed Shop CM5; Crane 1-1001 or 1-1002; Head Quarters HQ-26; Leineweber E-31 or E-5S; Red Shift 575; Screamin' Eagle 400; Sifton 141-EV or 146-EV; or S&S 561 is the group to choose from. Remember to match the valve springs, spring spacing and lifters to the particular cam.

The S&S Super E is a good carb for this size engine, although the Super G may help top end performance. The rest of the modifications listed above are essentially the same as those described for the Stage III engine, including drivetrain modifications.

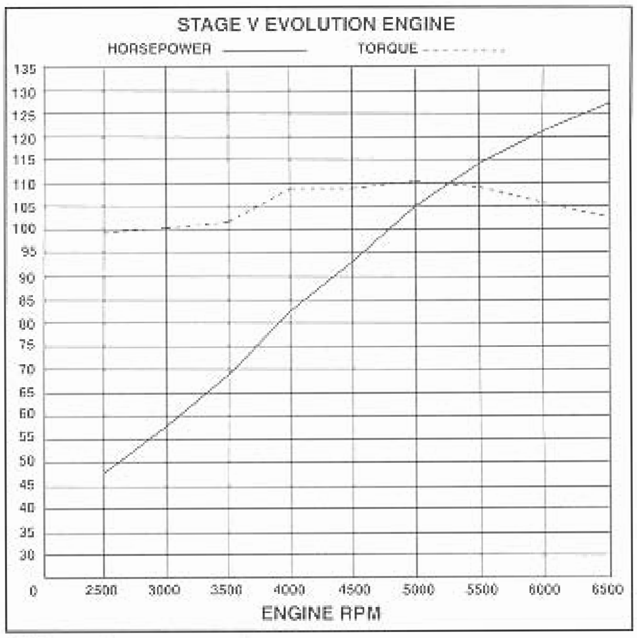
This engine is capable of producing 110 to 115 horsepower between 6000 and 6700 rpm and 100 to 105 pound feet of torque between 4500 and 5200 rpm. With a good chassis and drivetrain setup and a capable rider, quarter mile times of mid 11s to low 12s in the area of 108 to 115 mph can be expected.

Stage V Evolution
96 or 98 Cu. In. Big Bore Stroker Engine
Low to Mid 11 Second Bracket
114 to 120 MPH — About 120 to 125 HP

- ✔ 4-5/8" or 4-3/4" Stroke,
  3-5/8" Bore Kit
  (A stock stroke and 3-13/16" bore kit gives 97 cu. in.)
- ✔ 9:1 to 10:1 CR — Pump gas
  10.5:1 to 13:1 CR — Race gas
- ✔ S&S 2-1/16" Super G Carb — Street
  S&S 2-1/4" Super D Carb — Drag
- ✔ Performance Camshaft
- ✔ Adjustable or One-Piece Chrome-moly Pushrods
- ✔ Roller Rocker Arms
- ✔ Performance Single Fire Ignition
- ✔ 1-3/4" or 2" Exhaust System
- • Professionally Ported Heads
- • High Output Coil and Plug Wires
- • Performance Clutch

A Stage V engine uses either the same 4-5/8" stroke flywheels as the Stage IV engine for 96 cubic inches or longer 4-3/4" stroke flywheels for 98 cubic inches. In either case, Axtell or S&S 3-5/8" bore cylinders are added for a total of either 96 or 98 cubic inches. As an alternative to a long stroke, the stock 4-1/4" stroke can be combined with Axtell 3-13/16" bore cylinders for a 97 cu. in. engine. Note that the crankcase spigot holes need to be bored out for the larger cylinders. New big bore pistons are connected to either S&S or Carrillo heavy duty rods. Compression ratio recommendations are the same as for the Stage IV engine.

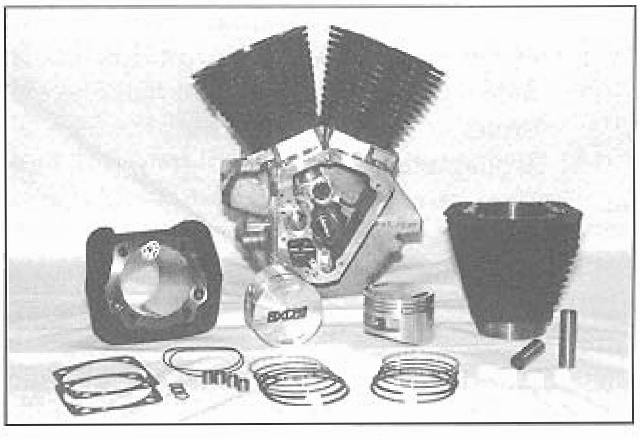
For a cam that complements this engine,



This graph represents the approximate horsepower and torque of a "streetable" Stage V engine when measured at the rear wheel. Both values will be higher when measured at the engine sprocket.

consider an Andrews EV7 or EV79; Carl's Speed Shop CM6; Crane 1-1003; Head Quarters HQ-26; Leineweber E-3 or E-5; Red Shift 625; Sifton 142-EV; or S&S 562. Again, remember to match the valve springs and lifters to the cam. Also, to reduce valvetrain weight, one-piece non-adjustable pushrods should be considered.

This engine can handle more carburetion, so an S&S 2-1/16" Super G for the street or a 2-1/4" Super D for the street or drag strip gets the job done. Some engine builders prefer to use a bored out S&S Super B carb on the street.



You can increase displacement to 97 cubic inches while retaining the stock stroke by installing 3-13/16" bore cylinders. Shown is Axtell's big bore Shortblock Kit. Photo courtesy of Axtell Sales.

On the race track try experimenting with 2" O.D. headers to determine if they help performance with your engine combination. Also, if you can afford the cost, this is the time to add roller rocker arms from either Baisley, Crane or Rivera along with a Dyna or M.C. Ignition single-fire ignition system.

Like the Stage IV combination, this engine should be balanced and blueprinted to specifications, including the crankcase oil breather system. For frequent, hard launches, a Bandit Machine SuperClutch will solve any potential clutch problems.

A Stage V engine is capable of about 120 horsepower between 6000 and 6500 rpm and approximately 118 pound feet of torque between 4500 and 5200 rpm. Assuming the chassis can get the power to the ground, this engine should put a 770 pound FX through the quarter mile in the very low to mid 11 second bracket at 114 to 122 mph.

Stage VI Evolution
103 or 114 Cu. In. Big Bore Stroker Engine
High 10s to Low 11 Second Bracket
120 to 125 MPH — 125 to 145 HP

- ✔ 5" Stroke x 3-5/8" Bore or 5" Stroke x 3-13/16" Bore
- ✔ 9:1 to 10:1 Compression-Pump gas 10.5:1 to 13:1 Compression-Race gas
- ✔ Performance Camshaft
- ✔ 2" O.D. Exhaust System
- • S&S 2-1/16" Super G or 2-1/4" Super D Carburetor
- • Performance Single Fire Ignition
- • Roller Rocker Arms
- • Professionally Ported Heads
- • Adjustable or One Piece Chrome-moly Pushrods
- • High Output Coil and Plug Wires
- • Performance Clutch

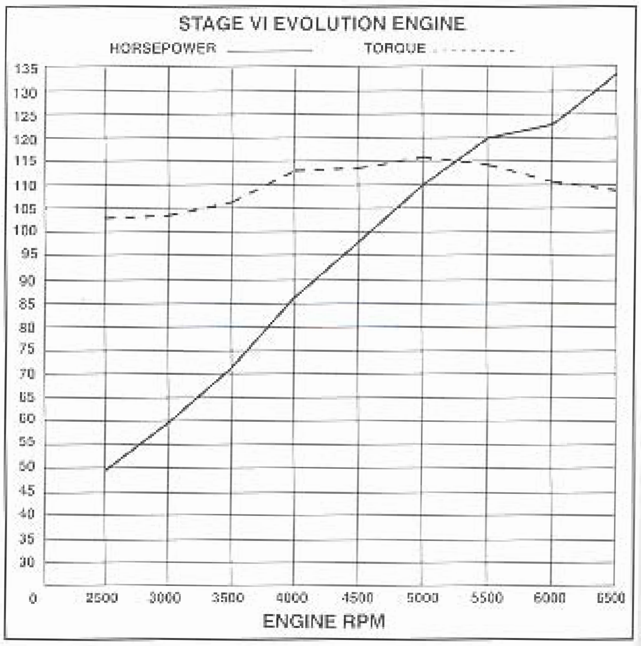
Although these engines are frequently run on the street, they are more of a drag engine than anything else. Without a doubt, the 114 cubic inch combination is harder on components than the 103 cubic inch version. Nevertheless, some racers have gone through an entire year of drag racing (about 200 to 250 runs) with this engine without disassembling the bottom end. When compared to a Stage V engine, this level of engine requires a greater degree of maintenance and a more watchful eye to ensure everything is in proper working order.

To achieve 103 cubic inches, the stroke is increased 1/4" beyond a Stage V engine for a full 5-inches. The bore remains the same at 3-5/8". For the 114 cubic inch engine, again the stroke is 5-inches, but larger 3-13/16" cylinders are now used.

Cam selection for this engine is similar to the Stage V engine, with the addition of the Andrews EV9; Carl's Speed Shop CM48; Head Quarters HQ-27; Leineweber E-7; Red Shift 710; Sifton 147-EV; and S&S 563 or 631 as possible choices.

Engines over 100 cubic inches can benefit from the additional high rpm scavenging offered by 2" O.D. headers. The remaining components are the same as for the Stage V engine.

These engine modifications have the potential to produce 125 to 140 horsepower between 5800 and 6500 rpm and about 115 pound feet of torque near 5000 rpm. High 10s to low 11s at 120 to 125 mph are possible with a properly setup street chassis and drivetrain.



This graph represents the approximate horsepower and torque of a "streetable" Stage VI engine when measured at the rear wheel. Both values will be higher when measured at the engine sprocket.

SHOVELHEAD COMBINATIONS

Modifications for the Shovelhead engine fall into two categories: the early model 74 cubic inch engine and the later 80 cubic inch engine. Although there are many similarities between the two engines, the most practical upgrade path for each sometimes differs.

Upgrading 74 cubic Inch Shovelheads

When starting with a small 74 cubic inch engine, the most practical displacement upgrades are the 84, 88, 93, 98, 103 and 114 cubic inch sizes. For highlights of the 103 and 114 cubic inch Shovelhead combinations, refer to the equivalent engine size described under "Evolution Combinations" because the dimensions and characteristics of the engines are identical.

The 84 cubic inch engine (4-1/2" stroke x 3-7/16" bore) includes a stock bore and longer stroke and is a practical option only when upgrading from 74 cubic inches. Over the years it has been one of the most popular combinations for anyone interested in more power while keeping stroke length relatively short. This combination is also an excellent choice if it is found during an engine rebuild that the flywheels need replacement due to worn shaft tapers.

When only increasing stroke, the 88 cubic inch engine (4-3/4" stroke x 3-7/16" bore) offers the most power and reliability for the buck. It also provides an excellent stepping stone for adding large bore cylinders later.

The 93 cubic inch engine (4-1/2" stroke x 3-5/8" bores) is a short stroke combination that keeps wear-and-tear to a minimum. It's a good choice for general street and touring bikes or when it is desired to keep piston speed down. In the quarter mile, this engine runs slightly quicker than the 88 cubic inch version.

The 98 cubic inch (4-3/4" stroke x 3-5/8" bore) engine offers the best overall combination of performance and reliability for the buck. This engine has good rod-to-stroke and bore-to-stroke ratios that work very well with the Big Twin engine. This engine should put a light weight FX street bike into the low to mid 11s at about 116 to 123 mph in the quarter mile.

Upgrading 80 Cubic Inch Shovelheads

When adding displacement to an 80 cubic inch engine, the 91, 93, 98, 103 and 114 combinations are the logical upgrades. Refer to "Upgrading 74 Cubic Inch Shovelheads" for characteristics of the 93 and 98 cubic inch engines since they are identical. Refer to "Evolution Combinations" for highlights of the 103 and 114 cubic inch engines since they have the same dimensions and characteristics as the Shovelhead versions.

The 91 cubic inch engine (4-3/4" stroke x 3-1/2" bore) is similar to the 88 cubic inch engine and gives the most power and reliability for the dollar when only the stroke is increased.

SHOVELHEAD MOTOR RECIPES

An average 74 or 80 cubic inch Shovelhead engine produces about 50 to 55 horsepower near 5000 rpm and a little more than 65 pound feet of torque at about 3500 rpm. Like the Evolution, this engine also suffers from poor intake and exhaust breathing, but in a different way. In general, its ports are too large and they suffer from low air velocity and high turbulence. One additional major problem is that its combustion chamber design is very inefficient due to low turbulence and large volume.

Although the Shovelhead's airflow capability is similar to an Evolution's, less of its air/fuel mixture is burned in the combustion chamber. This results in a higher percentage of unburned mixture being pushed out the exhaust. To make power with the Shovelhead engine, combustion chamber turbulence must be increased and flame travel must be shortened and kept unshrouded. The effectiveness of the chamber's squish area and the installation of dual plugs both play a crucial role towards this end.

A 575 pound Shovelhead FX with street tires, a 170 pound rider and a load of gas will weigh in at about 770 pounds. This setup should turn approximately mid to high 14s at 90 to 92 mph in the quarter mile. However, remember that as horsepower and torque increase, chassis, tire, clutch and transmission modifications are required made so the power can be transmitted to the ground.

For stock or near stock displacement engines, it is generally easier and less costly to increase horsepower to a given level when modifying an Evolution engine than a Shovelhead. But for engines 98 cubic inches and larger, the playing field is leveled due to cylinder airflow capability. Consequently, there is little performance advantage for either engine when displacement is large. From a long term point of view, however, the Evolution holds one trump card over the Shovelhead in that most research and development is being concentrated on it and over time this should provide the Evolution with an advantage. Nevertheless, if you currently own a Shovelhead and cannot afford to move to an Evolution, don't worry because the Shovelhead can be made to perform just as well — you just need to know what to do.

Many Shovelhead modifications use components similar to an equivalent Stage Evolution engine. In these cases, reference will be made to the Evolution engine.

The following conventions are used when listing parts for each stage of modification: A check mark next to a performance part indicates that the part is installed at this stage of modification. Parts without a check mark were previously installed.

Stage I Shovelhead
74 & 80 Cu. In. Engine
Mid 13s Second Bracket
95 to 98 MPH — About 65 HP

- ✔ Performance Exhaust System
- ✔ Mikuni HS40 or S&S Super E Carburetor
- ✔ Performance Coil & Plug Wires

A set of free flowing staggered duals and a pair of 1-3/4" O.D. header pipes or a 2-into-1 system will work well for an exhaust. The larger the muffler volume, the better the exhaust should scavenge. For the drag strip, 40" by 1-3/4" O.D. straight pipes are a good place to start. Change the pipe's length in 2" increments to fine tune them to your engine. Remember to enrich the carburetor jetting when installing a free flowing exhaust system.

Either a 40mm Mikuni or 1-7/8" S&S Super E carburetor is suitable for this engine and either one will help the engine breathe better.

For the ignition, a high voltage coil from Accel, Andrews or Dyna and performance spark plug wires will provide a good, hot spark for more complete combustion. For 1979 and earlier models, make sure the mechanical advance unit works smoothly. For 1978 and earlier engines with points style ignition, consider replacing the stock breaker points backing plate with an Accel Super Plate. On 1980 and later engines with an electronic ignition, make sure the Vacuum Operated Electric Switch (V.O.E.S.) is working properly or consider replacing the entire ignition system with either a points style unit or a single-fire Dyna or M.C. Ignition electronic unit.

This engine will produce about 65 horsepower at 5500 rpm and should be able to turn mid 13s at 95 to 98 mph.

Stage II Shovelhead
74 & 80 Cu. In. Engine
Low 13 Second Bracket
100 to 103 MPH — About 75 HP

- ✔ Performance Camshaft
- ✔ Aluminum Pushrods and Solid Lifters
- ✔ Performance Clutch
- • Performance Exhaust System
- • Mikuni HS40 or S&S Super E Carburetor
- • Performance Coil & Plug Wires

The next step toward enhancing the Shovelhead's breathing capabilities requires a more performant cam along with solid lifters and lighter pushrods. Consider the following cams for this engine level: Andrews A and B; Crane 288B and 308B; Head Quarters HQ-11/12 or HQ-13/14; Leineweber L1 and L2S; Sifton 107 and 112; and S&S 514. Be aware that some of these cams are bolt-in, while others require performance valve springs and proper spring spacing. If you do not want to go through the work of pulling the heads now, be sure to select a bolt-in cam. Also, remember it is better to under cam than over cam. When in doubt, choose the next smaller cam.

Since the Shovelhead's hydraulic lifters do not handle high rpm as well as the Evolution's, the stock lifters should be replaced with solids. Although solid lifters require periodic adjustment, they are the best performance option for the Shovelhead unless a set of Velva-Touch hydraulic lifters is installed. Remember to match the lifter design (solid or hydraulic) to the cam.

To keep up with the added power, the stock clutch should now be upgraded with a Barnett performance kit.

This engine combination should be near 75 horsepower at 5500 to 6000 rpm and should be able to do low 13s at 100 to 103 mph in the quarter.

Stage III Shovelhead
74 & 80 Cu. In. Engine
Low To Mid 12 Second Bracket
105 to 108 MPH — About 95 HP

- ✔ Professionally Ported Heads
- ✔ Compression Ratio 8.5 -1 to 9-1
- ✔ Dual Spark Plugs
- ✔ Single-Fire Ignition
- Performance Camshaft
- Aluminum Pushrods and Solid Lifters
- Performance Clutch
- Performance Exhaust System
- Mikuni HS40 or S&S Super E Carburetor
- Performance Coil & Plug Wires

At this point the engine has reached the limit of its breathing capability. Now professionally ported heads with performance valves, springs and collars are in order. Modifying a head to flow big numbers is both a science and an art. Professional head porters have usually scrapped many heads in the process of learning how to do it right. Therefore, it is recommended to let the professionals do the job. Before sending your heads for modification be sure to provide your head porter with the following information: objectives of bike, riding style, engine displacement, cam, compression ratio, gearing, bike weight and rpm range. A top notch performance head job includes not only porting, but also oversize valves and seats, blending the seat to the port and chamber, and reworked combustion chambers.

Professionally ported heads will increase performance about 25 percent. If you don't have the money for a professional set, you can make slight modifications yourself and get up to an 8 or 10 percent improvement. In general, Shovelhead ports are already too large so don't make them any larger. Instead, first radius the short-side turn (located on the floor of the port) then blend the bottom of the valve seat to the port bowl and remove the rough casting marks on the port walls. Next, finish the intake ports with a 60-grit paper roll and polish the exhaust ports. Install new valves, guides and springs and remember to space the springs to the cam's specifications. If a head needs replacement, S.T.D. Development makes an excellent performance head.

The stock Shovelhead compression ratio ranges between 7.4:1 and 8.0:1, depending on the year and model of the bike. Higher compression helps low and midrange power by replacing cylinder pressure lost due to longer cam duration. When running on pump gas, the engine usually can tolerate between 8.5:1 and 9.5:1 compression before reaching detonation levels as long as the heads are dual plugged and there is some amount of squish area. It's hard to make reasonable power unless the compression is at least to this level, so make every effort to properly setup the combustion chamber to handle increased compression. Rivera and S&S can supply pistons that will get the compression into this range.

Remember, it's difficult to get much squish area with a Shovelhead combustion chamber and stock bore unless the chamber is welded into a bathtub or hemitub shape. The squish area generates turbulence that is critical to making power with this engine. To achieve squish area without welding, the radiused bottom edge of the combustion chamber must be machined at an angle equivalent to the piston dome's. Then the cylinder length can be adjusted so the piston comes within .040 to .060-inch of the newly machined chamber area (squish band area). Refer to the chapter "Compression Ratio" for more information.

Just about any Shovelhead will run better with two plugs per cylinder and to run higher compression without detonation most Shovelheads need two plugs. Dual plugs not only shorten the flame travel in the chamber, but also unshroud its path. Although, many shops are capable of installing dual plugs, make sure the second plug is installed about at the same height in the chamber as the original plug. Also, if the second plug has a short 3/8-inch reach, use a heli-coil thread insert in the plug hole.

Ignition timing for engines with lower compression and single plug head will usually range between 35 and 40 degrees BTDC, however, some engines may need as much as 45 degrees. High compression engines may only need 30 degrees advance. Also, it's important to retarded the timing between 5 and 10 degrees from these values when using dual plugs.

Dual plugs require two coils instead of one. So if you haven't already done so, this is a good time to upgrade to a performance single-fire ignition like a Dyna or M.C. Ignition Power Arc unit.

This engine should be in the area of 95 horsepower between 5800 and 6300 rpm and is capable of getting a 770 pound bike into the low to mid 12s at about 105 to 108 mph range.

Stage IV Shovelhead
84, 88 or 91 Cu. In. Engine
High 11s To Low 12 Second Bracket
108 to 112 MPH — About 100 to 110 HP

- ✔ 4-1/2" or 4-3/4" Stroker Kit
- ✔ Performance Camshaft
- ✔ Blueprinted Transmission
- Professionally Ported Heads
- Compression Ratio 8.5:1 to 9:1
- Dual Spark Plugs
- Single-Fire Ignition
- Aluminum Pushrods and Solid Lifters
- Performance Clutch
- Performance Exhaust System
- Mikuni HS40 or S&S Super E Carburetor
- Performance Coil & Plug Wires

For simplicity the 4-1/2" and 4-3/4" stroker kits are described together. The 4-1/2" stroker kit only makes sense when rebuilding a 74 cubic inch engine or where the flywheels of an 80 cubic inch engine need replacement. Depending on the bore size, the 4-1/2" stroke provides either 84 or 86 cubic inches while the 4-3/4" stroke gives either 88 or 91 cubic inches.

This performance level requires removal of the engine from the frame. New, long stroke flywheels are installed along with stroker pistons. S&S or Truett & Osborn can supply the flywheels, while Axtell and S&S have pistons.

With pump gas and single plug heads, mechanical compression should be kept between 8.5:1 and 9:5:1. Some engines may handle up to 10:1 compression with dual plugs. With race gas, you can run between 10.5:1 and 12.5:1 compression. Maximum effort engines are setup with about 14:1 compression. It's imperative to attain as much air/fuel turbulence as possible with squish area and good chamber design. Keep the piston dome as low as possible, while striving to achieve the desired compression ratio. Refer to the "Stage III Shovelhead" description for further information regarding the squish area.

At this time, rebuild the connecting rods or install a set of heavy duty S&S rods. The engine should be balanced and blueprinted to proper specifications, and the crankcase oil breather system should be modified and timed to open and close for maximum oil scavenging.

This engine can use more cam than the stage III. An Andrews 6 and M; Crane 310B or 304B; Head Quarters HQ-15/16; Leineweber L-3 or J-4; and Sifton 105 or 107 are the cams to choose from. Remember to match the valve springs, spring spacing and lifters to the specific cam.

To eliminate missed shifts and chipped gears, the transmission needs to be disassembled and blueprinted to proper specifications. Also, it helps to magnaflux and backcut the gears and install close ratio first and third gear sets.

In general, the Mikuni HS40 or S&S Super E are good carbs for these engines, although the 88 and 91 cubic inch engines might benefit from the larger S&S 2-1/16" Super G.

Finally, the stock rear tire can be replaced with sticker Avon, Dunlop, Metzeler or Michelin rubber. The remaining modifications listed above are essentially the same as those described under Stage III engine modifications.

Depending on displacement, a Stage IV engine should produce between 100 and 110 horsepower at about 5800 rpm and is capable of high 11s to low 12s at 108 to 112 mph in the quarter mile.

Stage V Shovelhead
93 or 98 Cu. In. Engine
Low To High 11 Second Bracket
112 to 120 MPH — About 114 to 125 HP

- ✔ 4-1/2" Stroke x 3-5/8" Bore Kit or 4-3/4" Stroke x 3-5/8" Bore Kit
- ✔ Performance Camshaft
- ✔ S&S Super G or D Carb - 98" engine S&S Super E or G Carb - 93" engine
- ✔ Roller Rocker Arms
- ✔ Chrome-moly Pushrods and Solid Lifters
- ✔ 1-3/4 or 2" O.D. Exhaust System
- Blueprinted Transmission
- Professionally Ported Heads
- 9:1 Compression-Pump gas 9.5:1 to 14:1 Compression-Race gas
- Dual Spark Plugs
- Single-Fire Ignition
- Performance Clutch
- Performance Coil & Plug Wires

Stage V engines use either 4-1/2" or 4-3/4" stroke flywheels along with either Axtell or S&S 3-5/8" bore cylinders for a total of 93 or 98 cubic inches. The crankcase spigot holes must be bored for the larger cylinders and new big bore pistons are connected to either S&S or Carrillo heavy duty rods.

When running on pump gas keep the mechanical compression ratio to 9:1 to 9.5:1 with single plug heads; however, the engine may handle up to 10:1 with dual plugs. With race gas, run as much compression as you can without shrouding the flame front or airflow. Maximum effort drag engines are running 14:1 or higher mechanical compression. It's imperative to attain as much air/fuel turbulence as possible with squish area and good chamber design. Keep the piston dome as low as possible, while striving to achieve the desired compression ratio. Refer to the "Stage III Shovelhead" description for further information regarding squish area.

For a cam that complements these engines, choose either an Andrews 6, C, M or 9; Crane 310B, 304B or 320B; Leineweber J-4, L-5 or L-51; Red Shift 550; Sifton 117, 118 or 108; or an S&S 495. Remember to match the correct springs and lifters to the cam. Also, extra rigid chrome-moly pushrods are now recommended and if you can afford the cost, this is the time to add roller rocker arms from either Baisley, Crane or Rivera.

Try an S&S Super E or G carburetor with the 93 cubic inch engine. The 98 cubic inch engine works well with the Super G and in some situations the 2-1/4" Super D.

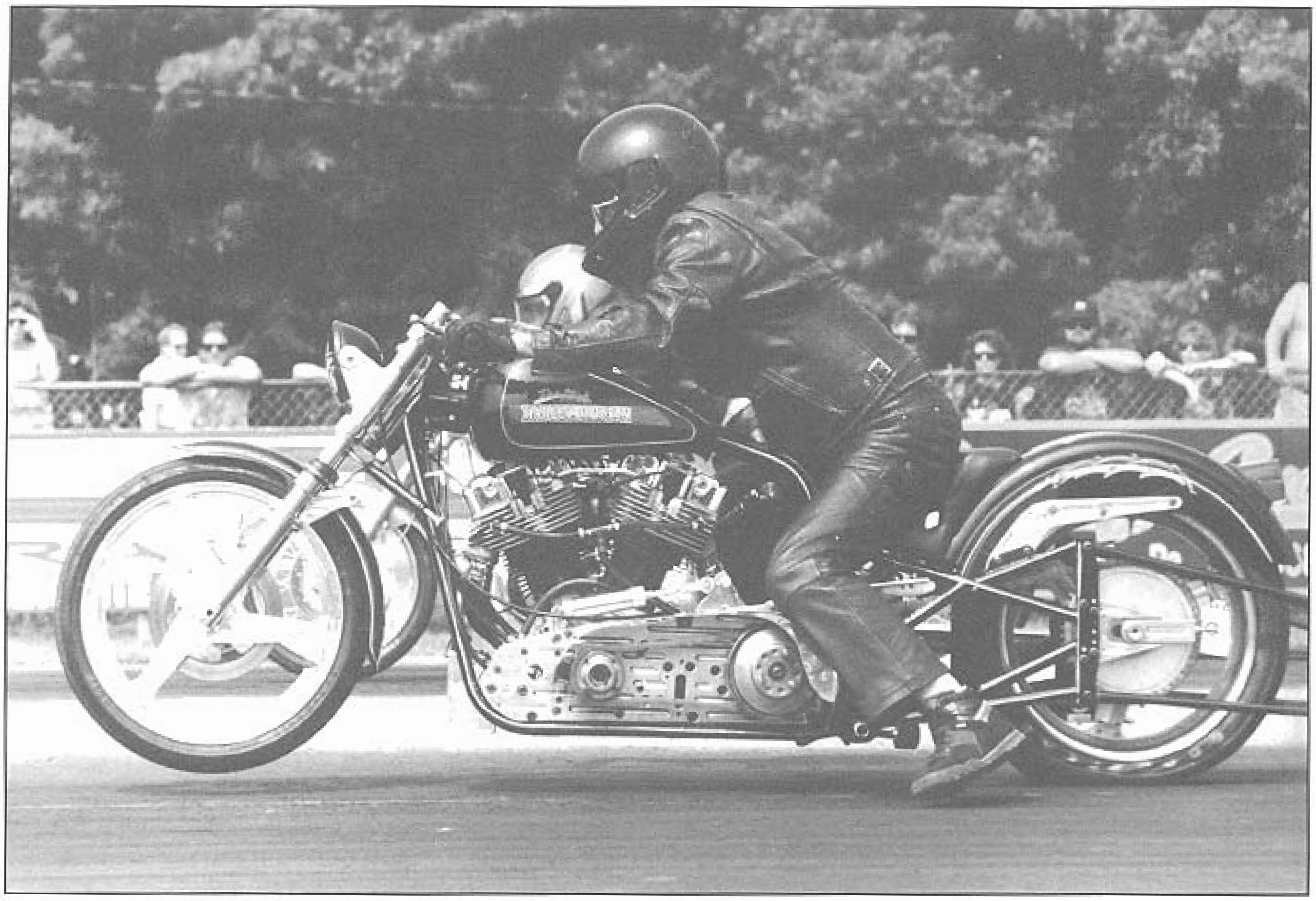
Like the Stage IV combination, this engine should be balanced and blueprinted to specifications, including the crankcase oil breather system. For frequent hard launches, a Bandit Machine SuperClutch will solve any potential clutch problems.

With a 98 cubic inch engine it's worth experimenting at the race track with 2" O.D. headers. For drag pipes on the race track, start with about 30" lengths for the 1-3/4" diameter pipes and about 40" for the 2" O.D. pipes.

A Stage V engine is capable of between 112 and 120 horsepower at 5800 to 6000 rpm. Assuming the chassis can get the power to the ground these engines should put a 770 pound FX through the quarter in the very low to high 11 second bracket at 112 to 120 mph.

Stage VI Shovelhead
103 or 114 Cu. In. Engine
High 10s to Low 11 Second Bracket
120 to 125 MPH — About 125 to 145 HP

- ✔ 5" Stroke x 3-5/8" Bore or 5" Stroke x 3-13/16" Bore
- ✔ Performance Camshaft
- ✔ S&S 2-1/16" Super G or 2-1/4" Super D Carburetor
- ✔ 2" O.D. Exhaust System
- Roller Rocker Arms
- Chrome-moly Pushrods & Solid Lifters
- Blueprinted Transmission
- Professionally Ported Heads
- 9:1 Compression-Pump gas 9.5:1 to 14:1 Compression-Race gas
- Dual Spark Plugs
- Single-Fire Ignition
- High Output Coil and Plug Wires
- Performance Clutch



Shovelheads are still running strong on the racetracks. If you build a Shovelhead right, it can run with the best of them. Robert Smith on his 114 ci Shovelhead Pro Stock. Note the external oil drain lines from the heads. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

Although these engines are frequently run on the street, they are more like a serious drag engine than anything else. Without a doubt, the 114 cubic inch combination is harder on components than the 103 inch version. However, with this engine some racers have gone through an entire year of drag racing, about 200 to 250 runs, without disassembling the bottom end. When compared to a Stage V engine, these engines require a greater degree of maintenance and a more watchful eye to ensure everything is in proper working order.

To achieve 103 cubic inches, the stroke is increased 1/4" beyond Stage V for a full 5-inches, while the bore remains at 3-5/8". For the 114 cubic inch engine, the stroke again is 5-inches, but larger 3-13/16" cylinders are now used.

For cam selection, choose either an Andrews C, 9 or 10; Crane 320B or 330B; Leineweber L-5, L-61 or L-6; Red Shift 550, 625 or 665; or a Sifton 108, 118 or 111. Remember, match the appropriate springs and lifters to the cam.

Carburetion is provided by either an S&S Super G or Super D. Additionally, these engine can benefit from the additional high rpm scavenging provided by 2" O.D. headers. The remaining components are the same as for the Stage V engine.

These engines have the potential to produce between 125 and 145 horsepower near 6000 rpm with maximum torque at about 5000 rpm. High 10s to low 11s at 120 to 125 mph are possible with a properly setup street chassis and drivetrain. ❖

Chapter 4

Induction System

Heavy Breathing

The engine's induction system includes everything from the air cleaner or velocity stack to the intake port.

This chapter covers the carburetor, intake manifold and the air cleaner assembly.

After installing a free-flowing exhaust system and possibly a performance ignition system, the induction system becomes the next bottleneck for the Big Twin. In stock form, the Big Twin Evolution engine puts out about .80 horsepower per cubic inch when measured at the engine sprocket (this figure is reduced by 15 to 18 percent when measured at the rear wheel). The Shovelhead has a slightly lower horsepower to cubic inch ratio than the Evolution. Today, many state-of-the-art naturally aspirated engines are producing two or more horsepower per cubic inch on gasoline. With the proper combination of parts and assembly techniques, the Evolution and Shovelhead can easily pump out on gasoline between 1-1/4 to 1-1/2 horsepower per cubic inch at the rear wheel.

The major reasons for the Big Twin's low horsepower to cubic inch ratio are due primarily to three factors: poor intake tract breathing, poor turbulence and incomplete combustion, and poor exhaust scavenging.

The main bottleneck of the Big Twin's induction tract is the carburetor and the intake manifold. Over the years the Big Twin has come equipped with many carburetors such as the Linkert, Tillotson, Bendix and finally the Keihin. Most of these carburetors have a venturi diameter of between 34 and 38mm or about 1-5/16 to 1-1/2 inches. The latest carburetor, the Keihin CV that was introduced in 1990, has a venturi diameter of 40mm or 1.575-inches.

The ratio of carburetor venturi to cylinder displacement is rather low on a stock engine. For example, the stock 40mm diameter venturi is equal to 1.948 square inches of area. Since the Big Twin's carb feeds each cylinder separately, it essentially feeds 40.35 cubic inches. This translates into 20.71 cubic inches for each square inch of venturi area. When compared to the Sportster, XR-750 racing engine or state-of-the-art racing engines, this carburetor is relatively small and low in airflow capacity for a 40.35 cubic inch cylinder. The situation is worse for 1989 and earlier Evolution engines because they have a 38mm carburetor. And it's worse yet for California engines equipped with 34mm carburetors.

The Y-shaped intake manifold also restricts airflow in the intake tract. The stock Shovelhead manifold includes sharp edges instead of gentle radius bends. These edges make it difficult for the airflow to bend around the corners. Consequently, flow is restricted. The early model Evolution manifold (1984 to 1989) not only has restrictions similar to the Shovelhead, but it also has a mounting system prone to leaks. The late model Evolution manifold (1990 and later) has a decent short-side radius and is free of sharp edges, but its long-side (backside) radius is too straight for optimum airflow.

Because of these restrictions, the carburetor and intake manifold must be replaced with a high flowing design to make any substantial horsepower with a Big Twin engine. Since the carb and manifold are externally located, they are relatively easy to replace without having to open up the engine. The only other major component that may need removal is the gas tank.

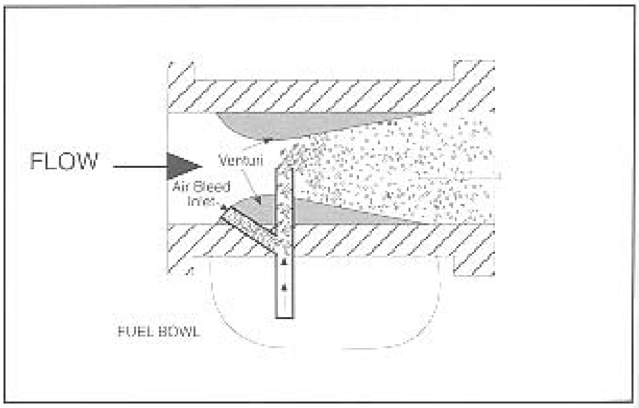
On a stock displacement engine with a free flowing exhaust and an unrestricted ignition, a performance carb will generally improve power about 15 to 20 percent. As engine displacement increases, the percentage of horsepower gain also increases. However, with a stock or near stock displacement engine, no one carburetor is going to provide enormous gains over another. Major power gains will normally be realized starting at 3500 or 4000 rpm because below this point most carbs and manifolds can satisfy the engine's airflow requirements.

Any engine with increased displacement should always include a high flowing carburetor. Significantly increasing engine displacement without also increasing the intake tract's breathing capability (read: "performance carburetor and head work") causes the engine's torque to peak at a very low rpm and then it drops off like a lead balloon. This is because the engine's large displacement runs out of breathing ability (volumetric efficiency) rather quickly. Since torque multiplied by rpm equals horsepower, it is easy to see why power drops off. Actually this phenomenon is also happening with a stock engine, but to a lesser degree.

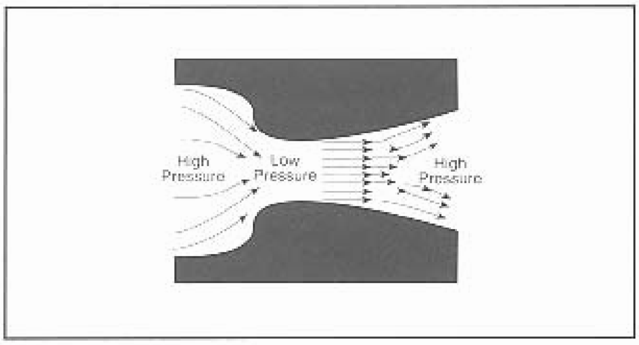
CARBURETOR OPERATION

Before discussing specific high-performance carburetors, it should be helpful to understand the basic principles of carburetor operation. This will help not only when you're selecting a carburetor, but also when you're tuning it for maximum performance.

The principal function of a carburetor is to regulate the flow of gasoline or other volatile liquid into the engine and help atomize the liquid fuel with a fast moving airstream for optimum combustion. The challenge the carburetor faces is to mix the fuel and air properly over the engine's entire operating range. It just so happens that some carburetors perform this function better than others.



Air speeds up as it flows through the restricted part of the venturi. The high speed air causes a pressure drop, which allows fuel to flow from the higher pressure float bowl area.



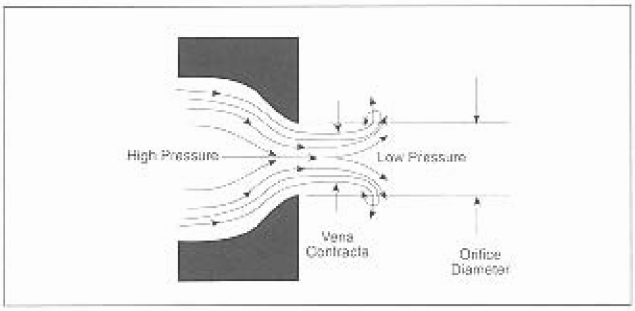
A venturi flows more air than a basic orifice because it reduces turbulence by gradually slowing the airstream down and then allowing air pressure to rise smoothly through a controlled tapered section. The more laminar flow magnifies the low and high pressure areas and increases the flow area of the venturi.

A basic carburetor has about five major components: venturi, fuel bowl, spray tube, jet and throttle valve. Of these components, the fundamental element is the venturi.

The *venturi* is located in the carburetor's body and forms an air restriction by sharply narrowing and then gradually increasing in size to as large as or larger than the original opening. As air passes through the restriction, it speeds up the same as when it moves over an airplane wing. The fast moving air creates a drop in pressure (vacuum) that causes fuel to be drawn.

The fuel is stored in the *fuel bowl* and is kept at a predetermined and relatively constant level by a float mechanism. The fuel bowl is vented to atmospheric pressure (14.7 psi at sea level) through a vent tube. Fuel is forced or drawn from the fuel bowl's high pressure area to the venturi's low pressure area and then mixed with the fast moving airstream. The higher the fuel level in the float bowl the easier it is for the venturi's low pressure area to draw fuel and the richer the air/fuel ratio becomes. As the fuel level drops, it becomes more difficult for the venturi's low pressure to draw fuel and this results in a leaner the air/fuel ratio. Therefore, it is important for the fuel system to sustain a constant fuel level in the bowl to maintain a consistently correct air/fuel ratio.

The *spray tube* is sometimes referred to as an emulsion tube, discharge tube or needle jet. It connects to the fuel bowl and is designed to help mix fuel with air before supplying the mixture to the low pressure area at the venturi. The outlet end of the spray tube is located at the venturi and is positioned higher than the fuel in the float bowl. Fuel flows up the spray tube because the venturi end of the tube is under a lower pressure than the vented fuel bowl. The decreased pressure at the venturi is called the *signal*. The strength of the signal is extremely important for good carburetion. The stronger the signal is, the better performance will be at low speed, given all things are equal. Although the highest signal strength would be expected at the venturi's smallest diameter, it actually occurs about .030-inch beyond this point due to fluid friction. This is why the spray tube is frequently placed off center in the venturi.



Airflow through a basic orifice loses kinetic energy due to turbulence on the exit side. The turbulence essentially reduces the flow area of the orifice. The vena contracta is the area of lowest pressure and is located down stream from the orifice's smallest diameter.

Ideally, fuel should be delivered by the spray tube in small, dense, cone-shaped droplets that spread out over a large area. This is important for good fuel atomization and vaporization because fuel droplets tend to attract each other. If they emerge in unevenly distributed, tightly bunched patterns, they are more likely to combine and separate from the air. This results in fuel puddling, which reduces throttle response and power because a lower percentage of fuel is burned in the combustion chamber. The amount of unburned fuel is significant and it gets dumped out the exhaust and is wasted.

The *jet* is a small threaded restriction that has a small hole to limit the amount of fuel that can pass. The size of the hole controls the overall air/fuel ratio of the mixture supplied by the spray tube.

The *throttle valve* is designed to regulate the amount of air/fuel mixture that is drawn into the combustion chamber. Most carburetors use either a butterfly, slide or cylindrical valve design.

The ideal carburetor is one that would satisfy the engine's airflow requirements at high rpm, wide-open-throttle (WOT) conditions. It would also supply an air/fuel mixture that has a nearly constant ratio regardless of the engine's rpm or load conditions. A basic carburetor can meter fuel *almost* proportional to airflow because the low pressure area at the venturi produces a signal at the spray tube that *almost* doubles as airflow doubles.

However, the laws of physics that govern the flow of both liquid and gaseous fluids from restricted apertures stipulate that as the low pressure in the venturi increases, the amount of fuel flowing from the jet and spray tube will also increase, but at a quicker rate than the increase in airflow through the venturi. As a result, the basic carburetor described here will provide a constantly richer air/fuel mixture as the engine's rpm increases. This is a key point to understand when tuning a butterfly style carb on the Big Twin. Also, sudden variations in the engine's rpm rate will affect the air/fuel ratio and its homogenization. Consequently, additional jetting circuits and devices are needed to address these conditions.

CARBURETOR CONSIDERATIONS

Air speed through a carburetor is determined by the venturi's diameter, position of the butterfly or slide, engine displacement and rpm. Increasing the size of the venturi decreases the air velocity through the carb for a given rpm and engine displacement, and reduces the signal at the spray tube. The reduced signal delays starting the fuel flow until a higher rpm is reached. When the signal falls below a threshold level, the fuel becomes less atomized and drops out of suspension from the air supply. This results in poor throttle response and reduced torque due to poor combustion. It also increases the potential for detonation and reduces fuel mileage at that rpm.

When the rpm level increases, air velocity through the carb will also increase. The higher velocity increases the signal at the spray tube, which increases fuel draw, improves atomization and eliminates undesirable performance characteristics. Additionally, greater fuel atomization also increases the possibility of more complete combustion that leads to increased power potential. At high rpm, a large venturi works well because air velocity is high. The high velocity produces a strong signal strength, which generates high mixture draw and good atomization. Although a large carb is great for large displacement engines or high rpm power, it can reduce power on small engines or at low rpm operation. Conversely, too small a carb will give great low and midrange performance, but top-end power will be reduced.

When comparing two different carburetors with similar airflow capabilities, performance differences can exist. For example, one carb may meter fuel more precisely over a broader rpm range or one may introduce less turbulence into the airstream.

About half of the carbs available for the Big Twin regulate airflow through a fixed size venturi with a throttle plate. A throttle plate is commonly referred to as a butterfly. Other carbs use either a flat or round slide to vary the size of the venturi and regulate the airflow. Constant velocity (CV) carbs such as the SU and the stock 1990 and later Keihin use a butterfly as well as a vacuum operated slide (round piston) to control airflow.

Butterfly carbs have a fixed venturi size and a round throttle plate (butterfly) which is mounted on a shaft located a short distance behind the venturi. As the high speed air hits the shaft and butterfly, turbulence is generated. The turbulence causes the air to tumble as it moves through the manifold and port. Turbulence in the intake tract reduces power because cylinder filling is generally reduced.

Some butterfly style carbs are designed with a very large entry diameter preceding the venturi and a large exit diameter following the venturi. The large entry and exit compensate for the turbulence and restriction caused by the butterfly and shaft. This allows the carburetor to flow about five to ten percent more air than another carb that has an identical size venturi, but smaller entry and exit paths.

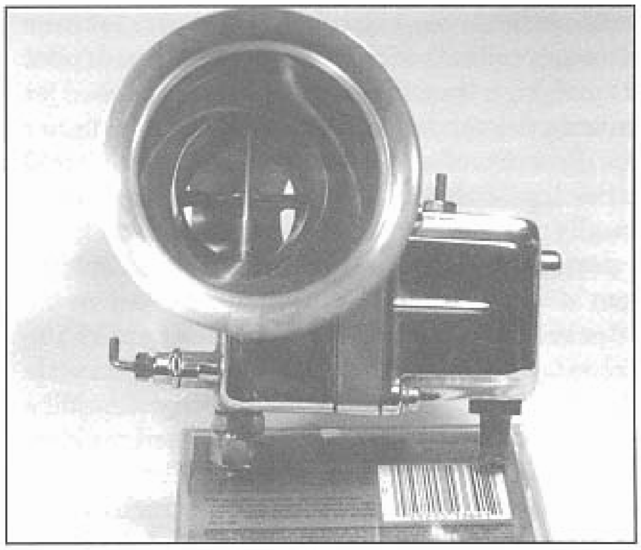
A flat slide carb has no butterfly and shaft and is relatively smooth through its bore. This keeps turbulence to a minimum and allows the air to travel more smoothly down the manifold and port. Its flat slide creates a variable size venturi when raised by a throttle cable. The venturi is small when air demand is low and increases in size as air demand increases. This keeps the air velocity high, which creates a high pressure drop that generates a strong signal for quick fuel delivery and good atomization. The strong signal provides excellent throttle response and tractability at low rpm while providing outstanding peak power.

At low rpm levels, the opening required to flow a given amount of air with the slide carburetor is about 40 percent the size of a butterfly carb's opening. The smaller opening generates an air velocity about four times that of a typical butterfly carb. Consequently, the signal at the spray tube is stronger, so better atomization and fuel draw is achieved.

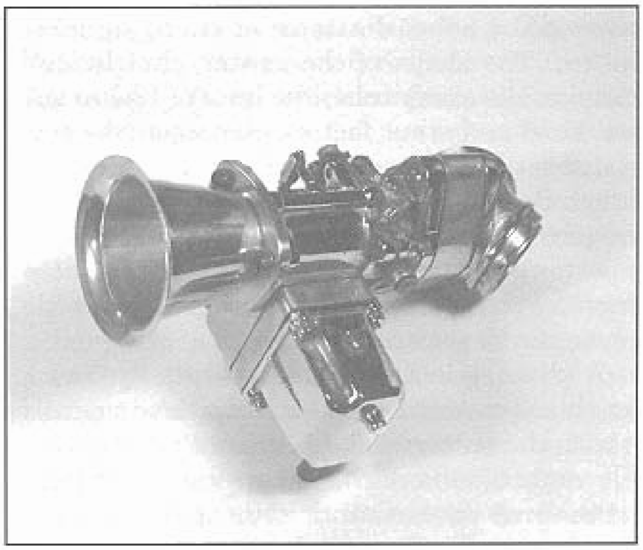
Another carburetor design is the variable venturi, constant velocity (CV) type. This style of carb is original equipment on 1990 and later Big Twins. The venerable SU carburetor also uses this design. A constant velocity carb has a vacuum operated round piston that regulates the size of the venturi along with a butterfly and shaft that regulates the airflow. The engine's demand for air regulates the up-and-down movement of the piston, which in turn regulates the size of the venturi. At low speed, the venturi is small and air velocity is kept high for maximum booster signal and throttle response. Conversely, at high speed the piston raises and creates a large venturi to satisfy the engine's greater demand. This design provides the benefits of both small and large fixed venturi carburetors. The downside is that its butterfly shaft generates high turbulence in the airflow.

JET CIRCUITS

Venturis of currently available performance carburetors range in size from about 40mm (1.575-inch) to 49.53mm (1.950-inch) and soon larger sizes will be available. The major difference between these carbs is not only the size of their venturi and flow capacity, but also how well they meter fuel throughout the rpm range. Since an engine's air/fuel requirements vary according to its rpm, load level, and other factors, there must be a way for the carb to adjust for the different conditions. These adjustments are normally handled by different jetting circuits. In the previous example of the basic carburetor, one jet circuit was represented. It is equivalent to what is called the main jet circuit.



Here is an example of a 1965 vintage Linkert butterfly style carburetor modified for racing. The horizontal butterfly has been thinned down and can be seen located toward the rear of the carb's throat. The vertical spray tube is protruding from the bottom of the throat. The venturi is bored out to 1-9/16", which leaves the carb's body paper thin. It also leaves the venturi with very little taper. The minimal venturi creates little pressure drop, so a large engine displacement is needed for good throttle response at low speed. Notice the fuel tickler on top of the float bowl and the large fuel inlet on the bottom. During the 1960s, this style carb was successfully used for gas and fuel drag racing. In the mid 1960s, Leo Payne successfully used a fuel version to lay down mid 9 second runs with his 74-inch Sportster fueler. The introduction of the S&S carb in 1968 ended the Linkert's racing career.



Most butterfly type carbs incorporate three jetting circuits: the idle mixture, low speed (pilot jet) and high speed (main jet). The low speed jet controls the mixture at idle and slightly higher rpm. A series of small holes is generally located in the top of the throttle bore. These holes are usually referred to as either transition, idle or bypass holes. As the butterfly disk starts to open from a closed or idle position, the holes are uncovered one by one. These holes supply the fuel mixture up to about one quarter throttle opening. At this point, the air velocity should be high enough to create sufficient vacuum (low pressure) in the venturi area to allow fuel to start flowing out of the spray tube, which is part of the main jet circuit. As the throttle disk moves past the series of transition holes, the flow of mixture through the holes starts to slow and then stops. At this point, the flow of mixture from the main jet circuit should start supplying the fuel. This point is referred to as the *transition period* and it's a period where large bore butterfly carburetors sometimes have a difficult time properly controlling the air/fuel mixture. This is the point where flat spots and hesitation are frequently experienced.

Sometimes these problems can be reduced or eliminated by adding more transition holes, increasing the holes' diameter or changing their pattern. The shape of the venturi, height and design of the spray tube, the level of fuel in the float bowl and other factors play a part in how good throttle response is during the transition period. Because of the butterfly carb's inherent design (low number of metering circuits), it is important to select a venturi size that is not too large for a street driven engine, otherwise throttle response will suffer.

A slide style carburetor essentially has a variable size venturi, instead of a fixed venturi as with the butterfly style carbs. With the variable venturi, air velocity is kept relatively high at the spray tube at all throttle openings. This provides a quick, forceful draw on the fuel mixture and good atomization. Instead of transition holes, this style carb uses a moving needle and needle jet along with a low speed pilot jet circuit and a high speed main jet to provide the correct air/fuel mixture at practically any throttle opening.

FUEL DELIVERY

The float bowl supplies fuel to the venturi area for metering to the engine. The fuel level in the bowl is regulated by a float mechanism connected to an on/off valve called a needle and seat assembly. The float, needle and seat are located inside the float bowl. As the float rises from the incoming fuel, the needle contacts the seat and shuts off the fuel supply. Conversely, the level stops rising as fuel is drawn out of the bowl. Now the float starts to lower, which drops the needle away from the seat and a new supply of fuel enters the bowl. The needle and seat are intermittently opening and closing the flow of fuel to the bowl.

If the engine draws fuel from the bowl faster than it can be replaced, the fuel level in the bowl drops and the air/fuel mixture becomes increasingly leaner and leaner. This condition is most likely to occur during hard acceleration at WOT because fuel requirements are greatest at this time. The longer the engine is at WOT, the greater the chance the fuel level will drop enough for the air/fuel ratio to go lean before the end of the quarter mile or straight-away.

A performance oriented carburetor must have a needle and seat with sufficient flow capacity to supply enough fuel to the engine under high acceleration conditions. A large fuel bowl capacity can aid the needle and seat by reducing the rate at which the fuel level drops in the bowl under high demand conditions.

CARBURETOR SELECTION

The ideal high-performance carburetor has high flow capacity, maintains high air velocity and generates a minimum amount of turbulence. It also includes enough adjustments for precise fuel metering over the entire rpm range and allows for easy jetting changes.

Selecting a carburetor for a street driven engine can be more difficult than a drag engine because the street engine encounters a broader rpm range. For a street bike application, good air/fuel metering from 2000 rpm to redline (usually 6000 to 6500 rpm) is needed to eliminate flat spots and to provide good gas mileage. A carb that maintains high air velocity is desirable for quick throttle response, especially at low rpm.

When selecting a carburetor, consider one that will deliver the highest *signal* strength for your engine's displacement and the rpm range it will be operating at. High airflow capacity needs to be balanced against good fuel metering and throttle response. Sometimes it is better to select a carb not because it flows more air, but because it delivers a better fuel curve for the application. Keep in mind that the most important factor for acceleration is engine torque, not horsepower, and too large of a carb kills torque in the low and midranges and can even kill torque at high rpm.

Other considerations for street driven engines include adequate gas tank and leg space clearance. A large five gallon gas tank generally provides less space for the carb than a smaller tank and therefore carb selection may be limited. Rubber mounted engines shake violently in the frame so more clearance is generally required between the carb and tank than with a solid mounted engine. Also, carburetors and manifolds that protrude a great amount from the engine can create a very cramped riding position for your leg and this can become very uncomfortable, particularly on long rides. A manifold that changes the carb's position by as little as one half inch can create interference problems between some carbs and large five gallon tanks. Finally, ease of installation should be considered. Carb kits that include the proper intake manifold and throttle cables eliminate many frustrations and hassles.

With a drag application, carburetor considerations are slightly different. High airflow is a greater priority than with a street carb and gas mileage is not a consideration. Depending on factors such as the bore/stroke combination, gearing, etc., a drag only engine is revved between 4500 and 5000 rpm on the low end, and 6500 and 7800 rpm on the high end. As a result, high air velocity (read: "good fuel metering and throttle response") is a major consideration only above 4500 or 5000 rpm because the engine will normally not drop below this level during race conditions. Consequently, the rpm range the carb meters fuel is narrowed and good throttle response can be achieved with a larger carb for a given engine displacement.

Just remember that for a given engine displacement, a larger carburetor requires a higher rpm level to bring the high speed jetting circuit into operation. Conversely, a smaller carb, due to its higher air velocity, brings the high speed circuit in at a lower rpm. A high rpm engine can handle a larger carburetor because at high rpm air velocity is kept high for good fuel metering. Remember, the point at which the high speed circuit starts delivering fuel plays a major role in throttle response and driveability.

A large carb can increase top-end power, but may hurt low-end throttle response and tractability. A carburetor too small for a particular engine displacement will provide good fuel atomization, throttles response and gas mileage, but will reduce top-end power.

Still, there are other more subtle factors that can play a part in carburetor selection — factors such as cylinder head airflow capacity. For example, an engine with high flowing heads, in general, can handle more carburetion than an equal displacement engine with lower flowing heads. As a result, it is sometimes difficult to identify the optimum size carb for a particular application.

Unfortunately, motorcycle carburetors are not flow rated as are automotive carbs. This fact and differences in carb design mean that you are frequently comparing apples to oranges when selecting a performance carb for your engine. Additionally, there is a myriad of carbs available for the Big Twin, which confuses the selection process even more. In fact, an entire book could be dedicated to only carburetors. Therefore, this chapter is limited to specifically discussing two proven performance carburetors, besides the stock Keihin (1990 and later).

One carburetor has a slide design while the other uses a throttle butterfly. Most performance carbs for the Big Twin use one of these designs. As such, many of the concepts and techniques described below can be applied to another carburetor with a similar design.

Since its introduction in 1968, the S&S carburetor has powered more Big Twin race winners than any other carburetor. This carb is a true legend and its performance records are well documented. It offers performance, simplicity, good looks and is readily available in kit form.

The Mikuni HS40 is a high flowing 40mm carburetor that provides excellent low and midrange power and throttle response. Additionally, it offers a wide tolerance in setup, is easy to dial in and is available in kit form.

Although the S&S and Mikuni HS40 are not the only carburetors that offer better performance for the Big Twin, they are proven performers, easy to install, readily available and do not interfere with the rider's leg position.

Two carburetors not discussed here are the venerable 2-inch SU and the 44mm round slide Mikuni. However, these carburetors make excellent power and are used in many successful performance applications.

To help the carburetor selection process, Table 4.1 lists possible choices based on engine displacement. The Table is valid for both Evolution and Shovelhead engines and although the displacement categories are arbitrary, they make sense from a logical view point.

CARBURETION SELECTION GUIDE

74–86 Cu. In.	88–93 Cu. In.	96 Cu. In. & Up
Mikuni HS 40	Mikuni HS 40	S&S Super G*
S&S Super B	S&S Super B	S&S Super D
S&S Super E	S&S Super E	
	S&S Super G	

* For these engine displacements, the S&S Super G can be bored out up to .100-inch. Refer to Appendix E for additional information.

Table 4.1

CARBURETOR SETUP

Unless extremely lucky, few racers are winning races with a carburetor that was dumped out of a box and bolted on their engine. A box-stock carburetor may not be far off in performance, but it won't have that extra performance advantage.

Today, carburetors are mass produced within a specified tolerance range to help keep them affordable. As a result, due to the wide tolerances, some carbs perform better than others. Rough shipping and handling can change a carburetor's pre-set factory tolerances and this can result in a performance reduction at the very least, or a poor running engine at the worst. Furthermore, instances have occurred where a new carburetor, straight out-of-the-box, had an O-ring missing and leaked as a sieve when the fuel valve was first turned on.

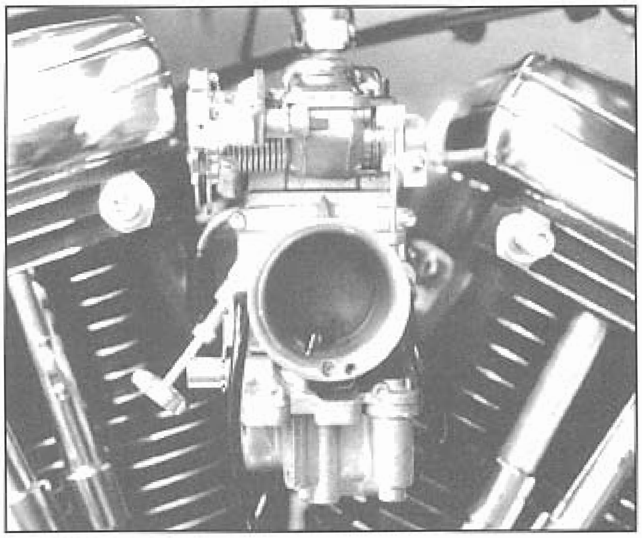
Therefore, if you want the maximum performance from your carburetor or want to save yourself some inconveniences, you should consider spending the necessary time to check the carburetor before installing it. If you're not familiar with your carburetor or this is the first time you are installing a carb, you can learn about it by carefully and patiently checking it over. This is an excellent way to get familiar with it and the knowledge you gain will help you not only during installation, but also during tuning.

Checking a carburetor is divided into two categories of procedures: basic blueprinting and advanced blueprinting. Basic blueprinting includes procedures such as: examining movable parts for proper functioning; verifying installed jet sizes; checking tolerances and settings of adjustable components; and ensuring that all parts, gaskets and seals are properly installed and seating. On the other hand, advanced blueprinting includes: machining procedures that change the shape or size of the venturi or throttle bore; modifications to the butterfly shaft or CV piston; modifications to the fuel and air circuits; or evening up the airflow and fuel signal of multiple carbs. Both categories of modifications are included below.

MIKUNI HS40 DESCRIPTION

The Mikuni HS40 is a 40mm smoothbore carb manufactured by Mikuni American specifically for installation on Harley-Davidson engines. It first entered the market place in late 1988 and is available in kit form for Evolution and Shovelhead models. The HS40 should not be confused with the Mikuni 40mm round slide (RS) carburetor.

Due to the clean airflow through its smooth bore, the HS40's venturi flows as if it were about 42mm. This carb is very versatile and offers excellent low and midrange throttle response because air velocity is kept high and airflow moves smoothly through its variable size smooth bore venturi. Its quick throttle response is partly due to the strong vacuum signal across its tuning circuits, which is a direct result of its narrow radial flat slide. Additionally, its fuel level is kept relatively high in the float bowl, which places the fuel closer to the mouth of the needle jet. As a result, fuel is drawn to the signal area quickly. The HS40 also offers good top-end performance for smaller displacement engines.



The Mikuni is noted for its excellent fuel curve over a broad rpm range. The carburetor provides excellent throttle response and comes in a kit, which includes a high flowing K&N air filter, extra jets and mounting hardware.

The HS40 has six fuel metering circuits to handle the entire range of throttle operation. A replaceable pilot air jet and pilot fuel jet along with an adjustable pilot fuel screw are used for controlling idle and low speed operation. An adjustable/replaceable needle and needle jet primarily control midrange operation and a removable main jet controls high speed operation above 3/4 throttle position. An adjustable accelerator pump is built into the carburetor and helps throttle response at any rpm.

Cold weather starting is handled by a mixture enrichment circuit instead of a choke butterfly and shaft. This circuit provides a clean path for the airflow through the carb's bore, unlike a choke butterfly setup.

Each HS40 carburetor kit includes a high flowing K&N air filter, throttle cables, mounting hardware and extra jets. A velocity stack is optionally available. If desired, the factory Evolution air cleaner cover can be reused for a stock looking appearance.

The Mikuni mounts to 1989 and earlier Evolution intake manifolds with a rubber flange, however, it requires an aluminum manifold adapter ring for mounting to 1990 and later manifolds. For optimum performance, the stock intake manifold should be replaced with either a Bartels', Branch, Ram Jett or S&S aftermarket manifold.

At one time, White Brothers in Garden Grove, California offered a kit that allowed retention of the stock cruise control for late model (FLHTCU) "dresser" models. The kit may still be available. Rebuild kits are available for the HS40 carburetor.

MIKUNI HS40 SETUP

The Mikuni HS40 carburetor can be separated into about three major component areas: the fuel float bowl, float, needle and seat assembly; jetting circuits; and accelerator pump. However, before any modifications or changes are made to any of these areas, a baseline check of all jet sizes should be performed and the results recorded for future reference. Also, the settings of adjustable items such as the float level, accelerator pump, jet needle and pilot screw should be verified and recorded.

Unless the Mikuni is used on a large displacement engine, install it with the factory jets and run the engine before making any jet changes. For tuning purposes, be aware that HS40 main jets are rated by size, while the pilot jets are flow rated.

The throttle return spring has three preload positions. The HS40 is shipped with the spring in the stiffest position, but this position can be fatiguing on the wrist. Moving the throttle preload to the middle position is usually preferred because it reduces fatigue. However, for safety reasons, ease of tuning and to achieve maximum airflow, you need to ensure that the radial slide opens fully and always returns completely to the idle position.

Float Bowl Components

If this carb has a shortcoming, it is its limited ability to flow a sufficient volume of fuel during high acceleration conditions. From a performance view point, anything modifications that increase fuel flow into the HS40 carb should help high rpm performance.

When you view the carb's fuel inlet swivel fitting that the fuel line connects to, you will notice that it makes a 90 degree bend into the carb body and the fuel passage makes an additional 90 degree bend at the needle and seat area. Any improvement to fuel flow in this area, such as radiusing the internal 90 degree bends with a Dremel tool, is beneficial. The needle and seat assembly can be removed and the top of the seat can be slightly shortened in a lathe for improved flow. The end of the fuel passage also can be radiused. Be careful not to damage the surface of the needle's seat.

An aluminum stop-pin protrudes from the bottom of the float bowl. The pin stops the float from dropping below the pin's end. The pin should be ground down to one half its normal height so the float can drop down lower. This will allow the fuel needle to move farther away from its seat and allow fuel to refill the bowl quicker.

The Viton tip needle and seat assembly acts as a valve and allows fuel to enter the float bowl. When the bowl is full, the needle must provide a good seal against the seat to shut off the fuel flow completely so the bowl does not overflow. The needle and seat assembly also includes an O-ring for sealing the seat against the carb body. With a new carb, leave the O-ring alone because it can easily get damaged when the needle and seat assembly is removed from the carb body. Even if the needle seals properly against its seat, a damaged O-ring can still allow the fuel bowl to overflow. If the O-ring needs replacement, the needle and seat assembly first must be removed.

The fuel level (float level) in the bowl is important for proper carburetion. The float level is set at the factory, but it can change during shipping or rough handling. The fuel level can have an affect on how rich or lean of mixture the carburetor delivers. The higher the fuel level in the float bowl, the easier it is for the vacuum at the venturi to draw fuel. Conversely, the lower the fuel level, the harder it is for the venturi's vacuum to draw fuel. In general, the carburetor tends to provide a richer air/fuel mixture with a high fuel level and a leaner mixture with a lower fuel level. This is the reason it is important to have the fuel level set properly

To check the float level, turn the carb upside down and tilt it at about a 15 degree angle with the needle and seat positioned on the high side. The highest point of the float, when measured opposite the needle and seat assembly, should be 19mm (about 3/4-inch) above the surface of the carb body. The float level is adjusted by gently bending the actuator tab that contacts the needle. Make sure the O-ring is properly positioned when replacing the fuel bowl.

TUNING CIRCUITS-HS40

There are six jet circuits on the HS40. Five are replaceable with different size jets and one (pilot screw) is adjustable. The six jet circuits can be categorized into three tuning circuits: the *idle / low speed*, *intermediate* and the *main*.

The idle/low speed circuit controls the mixture up to about 1/4 throttle open and includes the following tunable parts: pilot fuel jet, pilot air jet and the pilot screw. The intermediate tuning circuit controls the fuel mixture between 1/16 and 3/4 throttle position and includes the needle jet and jet needle. The main tuning circuit starts supplying fuel at about 3/4 open throttle and includes the main jet.

Most late model Mikuni HS40 carbs are shipped from the factory with the following jet combinations: #17.5 pilot fuel jet, #1.1 pilot air jet, #165 main jet, #9DJY4-96 jet needle, #Y-6 needle jet and the pilot screw set about three turns out from fully bottomed. Early model HS40 carbs were shipped with a 9DJY-1 jet needle.

IDLE/LOW SPEED TUNING CIRCUIT-HS40

The idle/low speed circuit influences the air/fuel mixture up to about 1/4 throttle opening. This circuit must be fine tuned before the other circuits.

Pilot Screw

The adjustable *pilot screw* (externally located on the lower backside of the carb) must be adjusted before other jets can be tuned. Start by warming the engine and hold the engine rpm between 1400 and 1500. Then adjust the screw either half way between its too-lean and too-rich positions or to a slightly over-rich position. Turn the screw *out* (counterclockwise) to enrich the mixture. The acceptable range for the screw is between 1/4 turn and 3-1/2 turns out. If the engine runs best with the screw beyond 3-1/2 turns out, either the pilot fuel jet is too small or the pilot air jet is too large. If the engine runs best with the screw less than 1/4 turn out, either the pilot fuel jet is too large or the pilot air jet is too small.

When the pilot screw is beyond about 2-1/2 turns out, it may vibrate loose. You can either stretch the spring or apply a small amount of RTV silicone sealer to hold the screw in position.

Pilot Fuel Jet

The *pilot fuel jet* (located inside the float bowl area) controls the maximum amount of fuel that flows through the idle/low speed circuit and works in conjunction with the pilot air jet. Available sizes are #15 to #30 in increments of 2.5 and #35 to #60 in increments of five. These jets are flow rated so the jet's number directly can be used to calculate flow changes during tuning.

Pilot Air Jet

The *pilot air jet* (externally located at the bottom right-hand side of the carb's air intake) controls the amount of fuel that flows through the pilot fuel jet by regulating the amount of air through the pilot system. For a given size of pilot fuel jet, increasing the size of the pilot air jet leans the pilot circuit. Decreasing the pilot air jet enriches the pilot circuit. Available pilot air jet's are #1.1 through #2.0 in increments of 0.1. These jets are flow rated like the pilot fuel jet.

The pilot *air* jet is normally selected by riding between 15 and 30 mph. The engine will tend to surge or possibly detonate if the pilot air jet is too lean. In this case, select a smaller size (lower number). If the engine misfires or fuel mixture is detected burning in the exhaust system, the air jet is too small (rich) and a larger size (larger number) should be selected. Also, an engine with too small of pilot air jet (mixture too rich) will have good acceleration through the low and mid rpm ranges up to about 80 or 90 mph. Beyond this point, the engine will stop pulling hard and will stop accelerating. On the other hand, if the pilot *fuel* jet is too small, the engine may tend to "cough" and "pop" through the carb at idle. Also, the engine will be slow to drop from about 2000 rpm to idle (about 1000 rpm). Changing to a one-size smaller pilot air jet has approximately the same effect as installing a one-size larger pilot fuel jet.

Refer to Table 4.2 for suggested starting points. Keep in mind that due to different engine combinations and ambient atmospheric conditions, your best jetting requirements may change.

Although the idle/low speed circuit primarily affects operation up to about 1/4 open throttle, be aware that it can influence the air/fuel mixture of the intermediate tuning circuit.

INTERMEDIATE TUNING CIRCUIT-HS40

The intermediate tuning circuit controls engine operation between 1/4 and 3/4 throttle operation. This circuit includes the *needle jet* and *jet needle*.

Needle Jet

The *needle jet* is a long tube that screws into the carburetor body. When you look through the carb's throat, the tip of the needle jet is seen slightly protruding from the bottom of the venturi. It can be removed with a 5/16-inch deep-well socket after removing the fuel bowl drain plug.

The function of the needle jet is to work in conjunction with a tapered jet needle to control the air/fuel mixture mostly between 1/16 and 1/4 throttle opening. Beyond 1/4 throttle opening, the tapered portion of the jet needle begins to emerge from the mouth of the needle jet and takes control of most of the air/fuel mixture up to about 3/4 throttle. Beyond 3/4 throttle, the jet needle's taper has no influence on the mixture.

Due to the low pressure area at the venturi, fuel is drawn from the fuel bowl through the needle jet to the venturi. The HS40 needle jet is a non-bleed design, which means it does not include air holes (emulsion tube design) intended to help the fuel mix and atomize with the incoming airflow. Instead, the pilot air and fuel jets handle this task. As air velocity increases, a non-bleed design increases fuel flow at a higher rate than an emulsion tube design.

The standard needle jet is a number Y-6. A number Y-8 is richer while a Y-4 is leaner. In most applications, the standard Y-6 needle jet is adequate; however, a richer Y-8 needle jet may be required for large displacement engines or in situations where a large pilot air jet is used.

Jet Needle

The tapered *jet needle* is attached to the aluminum radial slide by a metal E-ring. The needle consists of a straight section and a tapered section. The needle extends down vertically from the slide and passes through the center of the needle jet. As the throttle opens (slide raises), the needle raises with it.

From 1/16 to 3/4 throttle opening the amount of air/fuel foam or froth drawn into the carburetor's venturi is governed by the diameter of that part of the needle in the mouth of the needle jet and the inside diameter of the needle jet. As the tapered section of the needle is raised farther out of the needle jet, the orifice formed by both parts increases in size and allows more fuel mixture to flow through. From the point where the tapered section of the needle leaves the mouth of the needle jet (about 1/4 throttle opening), it has a significant affect on the air/fuel mixture. The needle's greatest influence is between 1/2 and 3/4 throttle opening because in this range it controls most of the fuel flow.

Keep in mind that the tapered section of the needle has no effect until the throttle is about 1/4 open. If the air/fuel mixture is too lean or too rich when the straight section of the needle is in the mouth of the needle jet (1/16 to 1/4 throttle opening), the needle jet (not the jet needle) is incorrect and must be changed. Raising or lowering the needle to rectify a mixture problem in the 1/16 to 1/4 throttle opening will have either no effect, or if the change does cure a problem a rich or lean condition will result between the 1/4 and 3/4 throttle position.

The jet needle's adjustability provides the HS40 with excellent fuel metering between the idle/low speed and high speed tuning circuits. This results in excellent throttle response at lower speeds and elimination of flat spots.

Early model (1989 and earlier) HS40s were shipped with a number 9DJY01 needle while late models (1990 and later) include a 9DJY04-96 needle. The only difference between the two is their part number. Two additional needles, the Y02 and Y03 (both are richer than the standard needle) are also available.

The jet needle can be accessed by removing the carb's top cover. The needle includes five grooves at its top end. An E-clip in one of the grooves holds the needle in the slide. Be careful not to lose the plastic washer located under the clip. The standard position (factory setting) for the clip is the middle (third) groove. The needle can be manually adjusted either richer or leaner by moving the position of the clip. Moving the clip to a higher groove lowers the needle and leans the mixture. On the other hand, moving the clip to a lower groove raises the needle and enriches the mixture.

To tune the jet needle, the throttle must be positioned between 1/2 and 3/4 open. The throttle's position can be determined by marking the handlebar throttle grip at full closed and full open position with a small dot of paint. Be sure all the slack is out of the cable before marking. Next, mark the 1/4, 1/2 and 3/4 positions.

To tune the jet needle, the engine needs to be accelerated quickly from 1/2 to 3/4 throttle position in high gear starting at about 50 mph. Adjust the needle for the strongest acceleration. A soft or slow response indicates a lean mixture. Another symptom of too lean a needle setting is that the engine runs better when the throttle is quickly shut from 3/4 to 1/2 throttle position. A symptom of too rich is that the engine will tend to load up with fuel and hesitate and stagger when the throttle is suddenly closed from 3/4 to 1/2 position. Strive for fast, crisp acceleration without any symptoms of the engine loading up during sudden throttle close. Remember that either enriching or leaning the jet needle not only will affect the air/fuel mixture during midrange, but also at the low and high speed ranges.

Long periods of storage can cause the radial slide or throttle linkage to stick, so periodically apply WD-40 or a similar lubricant to ensure free movement.

MAIN JET CIRCUIT-HS40

The *main jet* circuit includes the main jet, which controls the air/fuel mixture from about 3/4 to full throttle opening. Below this throttle range, the main jet has no influence. In fact, it even could be removed from the carb without affecting carburetion. This is because at about 3/4 throttle opening, the orifice created by the tapered needle and needle jet becomes larger than the main jet's orifice, resulting in the main jet size as the limiting factor to fuel flow.

The main jet is always the last jet to fine tune and it is reached by first removing the float bowl drain plug and then removing the needle jet with a 5/16-inch deep-well socket. The main jet screws into the bottom of the needle jet. Now the main jet can be exchanged and the needle jet assembly replaced in the carburetor. The HS40 is normally setup at the factory with a #165 main jet.

Main jets are available from #50 to #200 in increments of 2.5. Take note that HS40 round style main jets are size rated. This means its number represents the diameter of its orifice in millimeters.

Determining the optimum main jet is best done at the race track because the highest mile per hour in a given distance is the best indicator of maximum horsepower and optimum main jet size. For stock displacement engines, start with a #155 main jet. Refer to Table 4.2 for recommended starting points. Keep in mind that altitude, air temperature and even humidity will have a significant affect on the correct jetting.

First, warm the engine and then make a run down the track noting the final mph, but disregarding the run's elapsed time (E.T.). Increase the jet size by five, make another run and again note the mph. Be as consistent as possible with the runs. Continue making runs until the mph drops. At this point, decrease the jet size (lean) by 2.5 and record the results. By including air temperature, humidity and altitude information you can establish a usable baseline for future reference.

For anyone limited to street riding, check the main jet by quickly closing the throttle from wide open to 7/8 position when the engine's rpm is 4500 or greater. If the engine accelerates slightly, the main jet is too small (too lean). If the engine hesitates or misses slightly, the main jet is too large (too rich). If the engine just slows a slight amount, the main jet is close to the correct size.

Another test is to accelerate rapidly through the gears at full throttle. If the engine backfires through the carburetor, misses, intermittently cuts out or quits running, the main jet is too lean and its size should be increased by five. On the other hand, if the engine has sluggish acceleration, will not accept throttle or has a flat sounding exhaust, the main jet is too rich and its size should be reduced by five.

Take note how easily and quickly the engine reaches the gear shift rpm level. The engine should accelerate smoothly and quickly through the gears. A rich condition will tend to make the engine's acceleration labored and sluggish.

MIKUNI HS40 JETTING GUIDE

Engine	Pilot Air Jet	Pilot Fuel Jet	Main Jet	(From Top) Jet Needle	Needle Jet
80 Cu. In. w/modified exhaust	1.1–1.2	17.5	155–160	2–3	Y-6
80 Cu. In. w/modified exhaust & cam	1.4–1.6	22.5–27.5	155–160	2–3	Y-6
80 Cu. In. w/modified exhaust, cam & heads	1.5–1.8	25–30	152.5–160	2–4	Y-6, Y-8
80 Cu. In. w/modified exhaust, cam & heads	1.5–2.0	25–30	160–175	2–4	Y-8

Keep in mind that due to different engine combinations and ambient atmospheric conditions, your best jetting requirements may differ.
Table 4.2

ACCELERATOR PUMP-HS40

The Mikuni HS40 includes an accelerator pump for enhanced throttle response when quickly opening the throttle at low rpm. When the throttle is quickly opened, air velocity through the carb drops and the air/fuel mixture leans out. The accelerator pump injects a squirt of fuel directly into the carb's throat as the throttle is opened to help maintain a more correct air/fuel mixture. Be careful not to flood the engine accidentally by unnecessarily opening and closing the throttle.

The total volume of fuel injected is controlled by the beginning and ending of the accelerator pump's flow. Both points can be adjusted with two screws externally mounted near the top right side of the throttle shaft.

Best engine performance is usually achieved when the pump's push rod begins its stroke immediately as the throttle is moved from the idle position and ends its stroke at 3/4 throttle position. Attempt to achieve the best throttle response with the minimum amount of pump travel. Mark the throttle grip as described above to help determine the throttle's exact position when adjusting the pump.

The pump injects fuel into the carb's throat from a thin brass spray nozzle that protrudes about 1/4-inch from the bottom of the throat. For both cylinders to receive an even distribution of fuel, the nozzle must be positioned so the fuel stream strikes the needle jet. The nozzle is held in position by friction from a rubber O-ring seal and it can be easily rotated by gently gripping it with needle nose pliers. The nozzle's position can be biased to favor either cylinder, but it's better to direct the fuel stream at the needle.

The accelerator pump has an aluminum plunger and a return spring located on the right side of the float bowl. The plunger pumps fuel out of the nozzle. If the bike is stored for long periods of time or dirt or foreign particles get lodged between the plunger and its cylinder, the plunger can seize in a lowered position and stop working. Sometimes a small squirt of WD-40 on the pump rod, throttle shaft linkage and return spring is all that is needed to free the plunger. In severe cases, the plunger may need to be disassembled and lightly sanded with fine, 600 grit emery cloth.

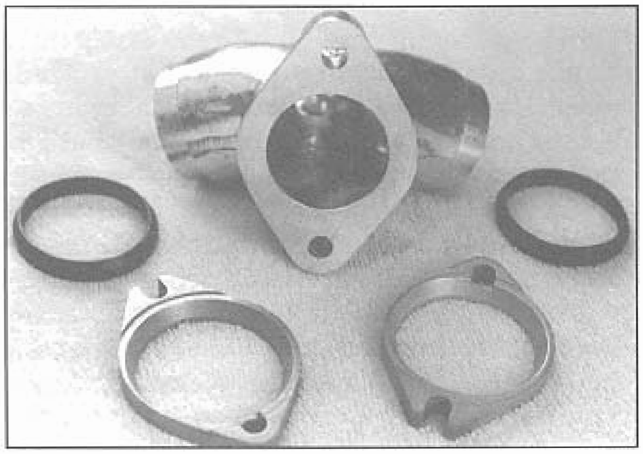
INTAKE MANIFOLD-HS40

As described earlier, the Mikuni HS40 mounts to an 1989 and earlier Evolution manifold with a rubber flange. One end of the flange slips over the male end of the carburetor, while the other end bolts to the stock manifold. For 1990 and later manifolds, an aluminum adapter ring is required for mounting. HS40 Shovelhead kits include a new manifold.

Regardless of the engine's year, for maximum performance the stock intake manifold should be replaced with either a Bartels', Branch, Ram Jett or S&S manifold. S&S offers various length manifolds for cylinders longer or shorter than stock. These manifolds have better radiuses and a larger plenum than stock manifolds. They also reduce turbulence by directing the airflow at a much better angle to the port than the 1990 and later manifold.

Some aftermarket manifolds come performance finished on the inside. For best performance, make sure the manifold runners are free from machine and casting marks, have gently curving radiuses and are finished with about a 60-grit cartridge roll. Also, never make the manifold's inside diameter larger than the intake port.

Always use new O-rings or rubberbands during installation and remember that the manifold mounting flanges for the 1990 and later Evolutions are different for the front and rear cylinders and cannot be interchanged. For Shovelheads that use a steel band clamp to fasten the manifold to the cylinder head, consider replacing the flimsy stock items with stainless steel aircraft-style clamps. These are available from S&S and other suppliers.



This Branch Evolution manifold comes reworked, flows a lot more air than the stock unit and is an excellent match for the Mikuni HS40 carburetor. S&S Cycle and Ram Jett Retainer also offer high-performance manifolds.

AIR CLEANER-HS40

Each Mikuni HS40 kit includes a high flow K&N air cleaner and chrome cover. K&N filters do an excellent job of filtering the air and have always been noted for their high airflow rating. Take note of the radiused airflow entryway located inside the cleaner where it mounts to the mouth if the carb. The radius entry increases airflow and reduces turbulence by directing and smoothing the air into the carb's throat. This technique was first used with modified Linkert carbs during the 1960s.

Be sure to clean the filter element periodically according to instructions (409 household cleaner works excellently) and reapply the necessary oil dressing using the correct oil. To retain a stock looking engine, the Evolution's factory chrome air cleaner cover can be installed over the K&N air cleaner. For a street engine, it is important to filter dirt and debris from the air entering the engine. However, for racing the air cleaner can be replaced with a radiused velocity stack for improved airflow into the carburetor.

SUPPORT BRACKET-HS40

Most Big Twin carburetors have a support bracket that prevents the carb from shifting, which could result in an intake manifold leak.

For the Evolution, the support mechanism is integrated into the backside of the air cleaner element and bolts directly to the cylinder head. For the Shovelhead, two separate steel brackets are included that bolt to the air cleaner and crankcase. In either case, make sure a carb support is always used and tightly fastened.

VACUUM FITTING-HS40

The Evolution factory ignition system includes a Vacuum Operated Electric Switch (V.O.E.S.) that helps reduce detonation during high load conditions. Some aftermarket high-performance ignition systems include provisions for the optional use of this switch. The V.O.E.S. is connected to the intake manifold by a thin rubber hose and works off manifold vacuum.

The Mikuni HS40 includes an external vacuum fitting on its lower right side toward the back of its body. If you are not using a V.O.E.S., make sure the vacuum fitting is plugged closed.

S&S CARBURETOR DESCRIPTIONS

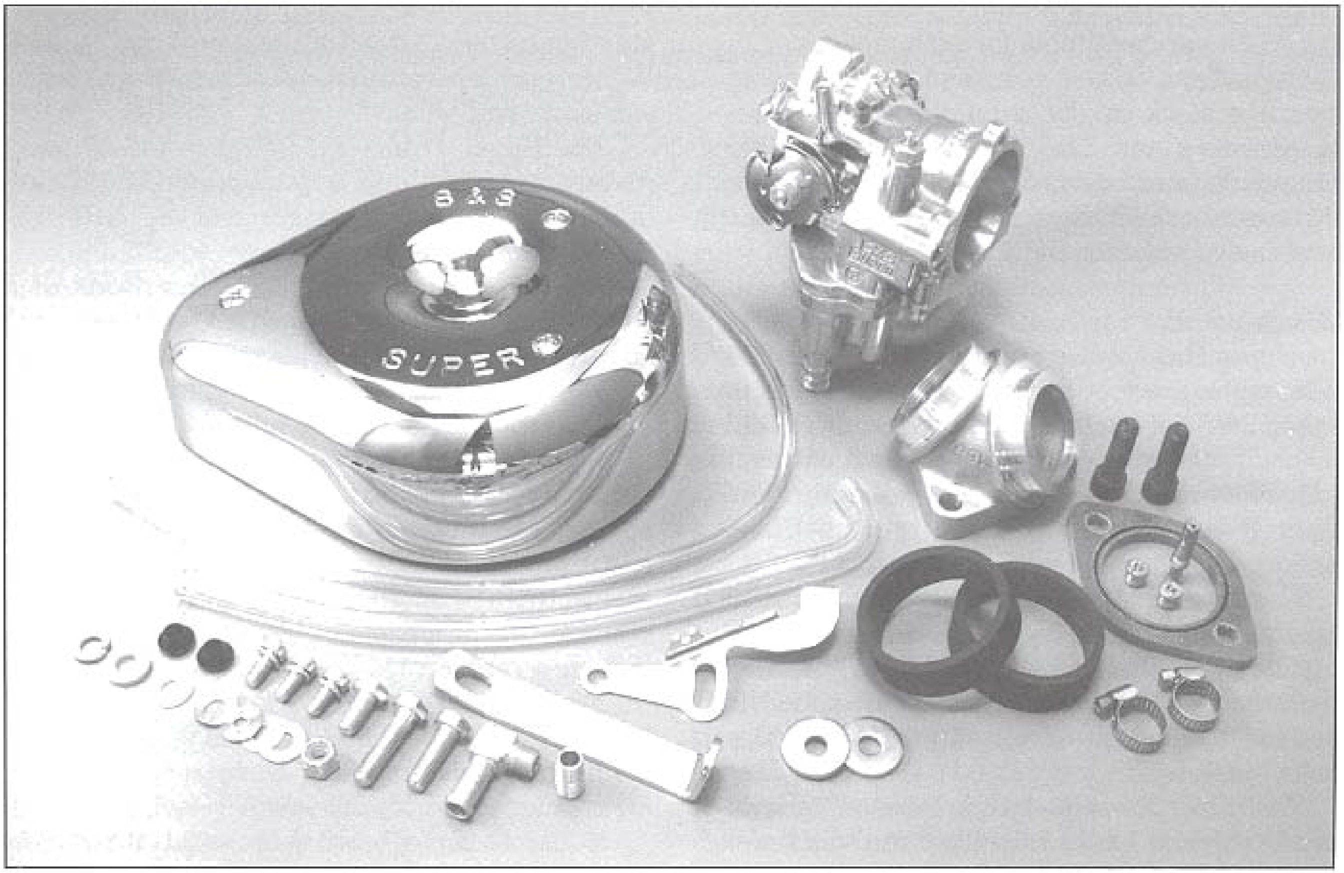
S&S Super E and G

These two S&S carburetors became available in late 1990 and are the latest evolution of the S&S gas carburetor since its introduction in 1967.

The Super E and G model carbs are butterfly style carbs and are identical in design except for the size of their venturi and throat. The Super E has a 1-9/16 inch (39.6mm) diameter venturi and a 1-7/8 inch diameter throat when measured at the butterfly valve. The Super G's venturi measures 1-3/4 inch (44.5mm) diameter and its throat 2-1/16 inch diameter. Each carb is identified by either an "E" or "G" cast into the throttle linkage side of the carb's body.

Both carburetors are designed with two fuel metering circuits: a replaceable midrange (intermediate) jet and a replaceable high speed (main) jet. It also has an adjustable idle mixture screw.

Optionally, a ThunderJet can be installed to add a third fuel circuit that provides a more balanced fuel curve between the intermediate and main jet circuits. Also, the internally located (float bowl area) main air bleed can be modified to accept an externally located replaceable jet. The replaceable air bleed significantly helps in tuning the fuel circuits.



The Super E is the latest evolution of the S&S carburetor. It has a shorter body than previous models and includes an accelerator pump. A Super G model is also available, but with a larger venturi. When setup properly, the S&S Super gives excellent performance and is the choice of racers. Photo courtesy of S&S Cycle.

Both carbs include an adjustable accelerator pump for better throttle response. This feature was not available on any previous model S&S carburetor.

Cold weather starting is handled by a variable mixture enrichment/fast idle circuit instead of a choke butterfly and shaft. This circuit provides a clean path for the airflow through the carb's bore, unlike a choke butterfly setup and offers improved engine starting and warm-ups.

The major differences between the Super "E" and "G" model carbs and the previous "B" model (other than the larger venturi of the "G" model) are that the new model carbs are about 1-7/16 inch shorter for improved rider leg room, include an accelerator pump for improved throttle response and have a redesigned higher flowing air cleaner. The reduced length has led to the nickname "Shorty" carb.

Each carburetor kit includes a teardrop air filter assembly, intake manifold, throttle cables, mounting hardware and extra jets. Two lengths of velocity stacks are optionally available and rebuild kits are available for both models.

Regardless of the year and model of Big Twin, the stock intake manifold is always replaced with a new S&S unit. Due to its larger plenum area and internal design, the S&S manifold flows about seven percent more air than the stock early Evolution manifold.

S&S Super B

The Super B carb was first introduced in 1976 and is still in production. It is distinguished from earlier S&S gas carb designs by the round fuel bowl mounted under its body and by its enrichment device instead of a choke butterfly. The Super B is also available in a nitromethane version.

The Super B superseded the almost identical Super A model carb. The Super A was produced only during 1975. Between 1967 and 1974, S&S gas carbs were available in L or G series models that included a side mounted fuel bowl and a choke butterfly.

The Super B model carb is a butterfly style carb that has a 1-9/16 inch (39.6mm) diameter venturi and a 1-7/8 inch diameter throat when measured at the butterfly valve. It is identical in size to the Super E model and is similar in that it has two fuel metering circuits plus an adjustable idle mixture screw and an enrichment device for cold weather starting. A ThunderJet fuel metering circuit can be added as a third metering circuit and the externally located main air bleed can be modified to accept a replaceable jet for greater tuning control. The Super B is available as a kit with components similar to the Super E. A velocity stack is optional.

The Super B differs from the Super E in that it does not include an accelerator pump and it is almost 1-1/2 inches longer. Although it flows the same amount of air as the Super E, some racers prefer the Super B because its longer body length provides different intake tract tuning characteristics and because of the shape of its throat.

S&S Super D

The S&S Super D gas racing carburetor was introduced in 1983 and except for its size is similar in design to the Super B. It uses a butterfly to control airflow, has a round float bowl mounted under its body and an enrichment starting system instead of a conventional choke. Either gasoline, alcohol or nitromethane versions are available.

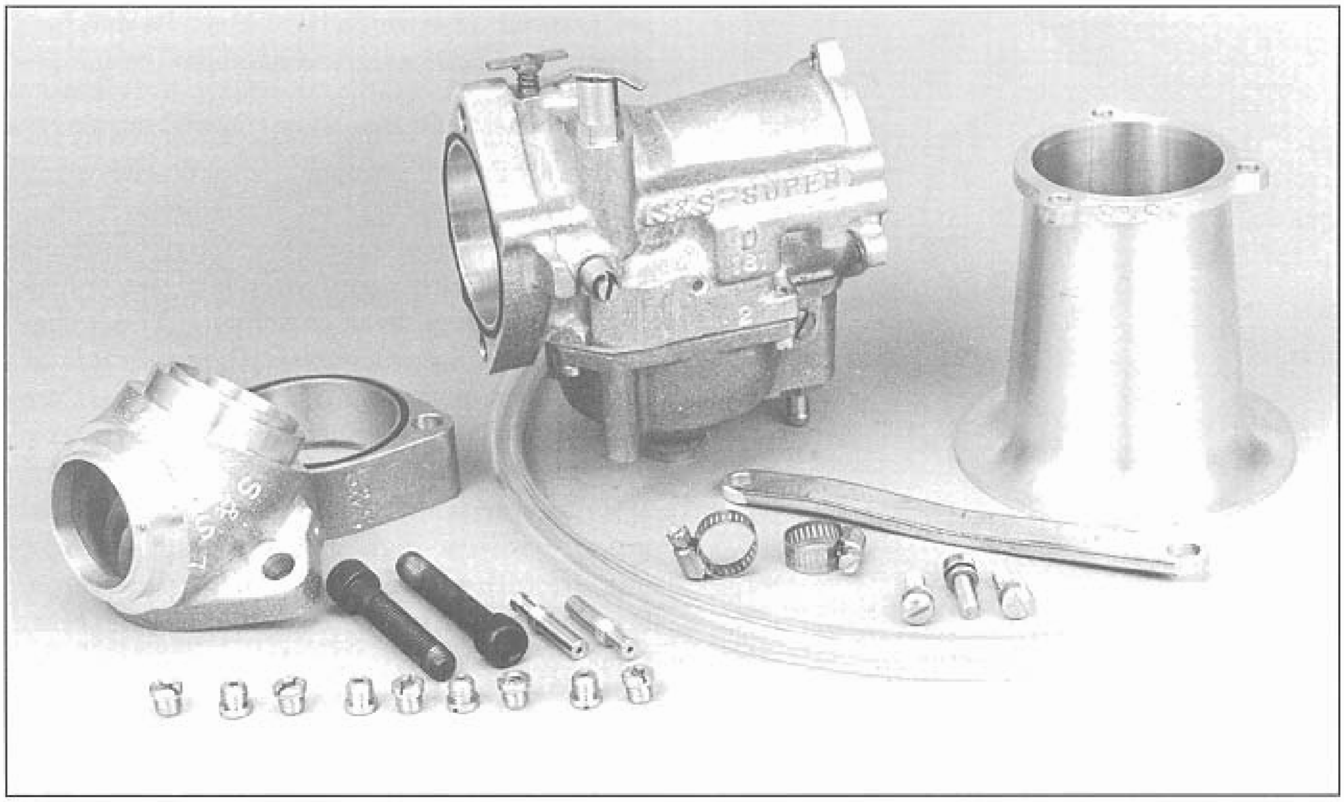
The Super D has a 1.950-inch (49.53mm) diameter venturi and a 2-1/4 inch (57.15mm) diameter throat when measured at the butterfly valve.

Just like the other Super model carburetors, the Super D is designed with two fuel metering circuits: a replaceable midrange (intermediate) jet and a replaceable high speed (main) jet. It also has an adjustable idle mixture screw.

As with other S&S carbs, a ThunderJet can be installed as a third fuel circuit for a more balanced fuel curve between the intermediate and main jets. Also, the externally located main air bleed can be modified on early model Super D carbs to accept a replaceable jet. Late versions include a replaceable (.040-inch) air bleed jet.

Each Super D carburetor kit includes a velocity stack, intake manifold, 1-inch spacer block, mounting hardware and assortment of jets. An air cleaner assembly is optionally available.

Super D carbs built before mid 1985 ("A" bowl design) include a flat tipped needle and seat assembly while those produced after this time ("B" bowl design) include a pointed tip needle



The S&S Super D was designed from the beginning as a racing carburetor. It has a 2-1/4 inch throat and works best on 98 cubic inch and larger engines. Although intended primarily for drag racing, some are used on large displacement street engines. Photo courtesy of S&S Cycle.

and seat assembly. Again, rebuild kits are available.

When compared to the Super B or E model carbs, the Super D offers a performance increase of approximately 2/10 second E.T. in the quarter mile when used on 96 cubic inch and larger engines.

S&S Early Models

Since the S&S carburetor has been around for over 25 years, there are many older models still in use. Most of these carbs are identified with a serial number and model designator. S&S usually can help if you need to verify a carburetor model or need replacement parts.

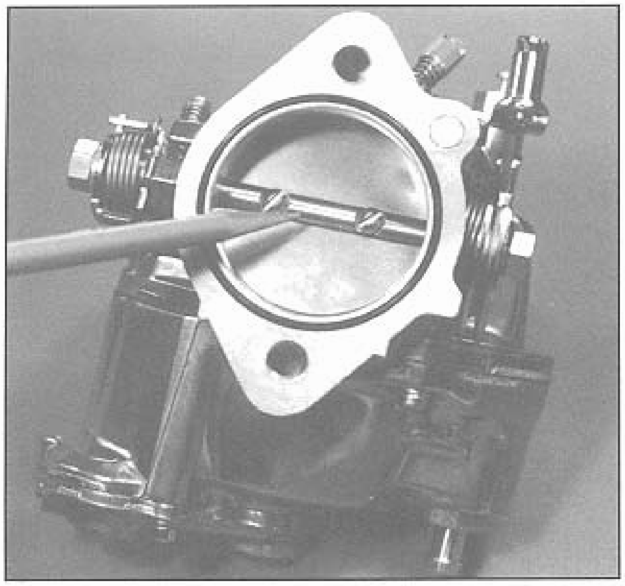
S&S CARBURETOR SETUP

The following information describes the setup of S&S Super E, G, B and D carburetors. Differences between the models are pointed out.

S&S Super carburetors are separated into two major component areas. The first area is the fuel delivery system, which includes the fuel float bowl, float, and the needle and seat assembly. The second area consists of the jetting circuits. The Super E and G also include a third major area — the accelerator pump.

Before making modifications or changes, a baseline check of the intermediate and main jet size should be performed and the results recorded for future reference. Also, the settings of adjustable components such as the float level, accelerator pump and idle mixture screw should be verified and recorded.

For all carbs, make sure the throttle butterfly shaft works smoothly and does not bind. Also, to eliminate air leaks, the throttle shaft must be tight fitting in its body. S&S rebuild kits include throttle shaft bushings that correct a loose fitting shaft. Also, for maximum airflow, make sure that when the butterfly is completely open it is positioned exactly horizontal within the carb's throat. Additionally, make sure the butterfly closes completely when it is shut. After verifying the butterfly completely closes against the carb's throat, slightly open it by turning the



Make sure the throttle linkage works smoothly and the butterfly closes all the way. The shaft can be thinned down and smaller screws used to hold the butterfly to the shaft. This will reduce turbulence and increase air flow. Use Loctite on the screws and then stake lock them. Double check that they are secure and will not come loose.

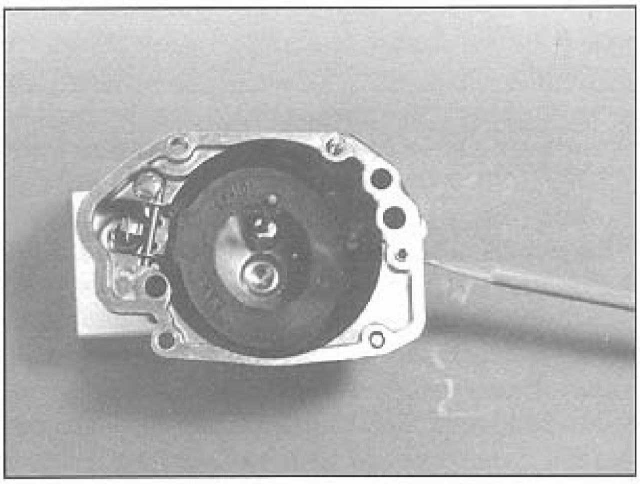
idle speed screw a half-turn clockwise after it contacts the throttle cable spool.

Small jet orifices are very susceptible to clogging from dirt and debris. When handling a carburetor, be sure that all passages are kept microscopically clean. It is easier to install the appropriate baseline jets before installing the carb on the engine. Refer to the Jetting Tables for recommendations. Starting about 1984, all S&S jets are rated by size. This means the jet's number corresponds to its orifice diameter in thousands of an inch.

Float Bowl Components

S&S Super gas carbs include a doughnut shaped Nitrophyl float (early models used brass) and a Viton tipped needle and seat assembly (some needles are solid brass) that is generally large enough for sufficient fuel flow. The standard fuel inlet measures .235-inch I.D., but a larger racing only needle and seat assembly is available from S&S. Four different designs of needle and seat assemblies have been used since the first Super carb. So if you need to order replacement parts, pay close attention to the model number. Some needles are shaped with a flat tip, while others are pointed. Also, some have a needle and fork design, while others have a needle, wire and tab design.

The fuel level in the bowl is governed by the float setting and it is important for proper carburetion. The fuel level is set at the factory; but it can change during shipping or rough handling. The level will have an affect on how rich or lean of mixture the carburetor delivers. The higher the fuel level in the float bowl, the easier it is for the vacuum at the venturi to draw fuel. Conversely, the lower the fuel level, the harder it is for the venturi's vacuum to draw fuel. In general, a carburetor tends to provide a richer air/fuel mixture with a high fuel level and a leaner mixture with a lower fuel level. For this reason, it is important to verify that the fuel level is set properly.



The float level can change during shipment and should be checked before installation. The float should be 1/8 to 3/16-inch below the gasket surface when the needle is seated and the spring fully compressed.

To check the float level, first make sure the float and needle work freely and the float does not contact the bowl gasket. For all Super models, remove the float bowl from the carburetor body and while holding the bowl level, gently raise the float until the needle is fully seated and the spring on the needle if fully compressed. With the E and G models, the highest part of the float should measure between 1/8 and 3/16-inch below the bowl's gasket surface. For the Super B and D models, the correct distance is 1/8-inch below the surface.

If the carburetor floods over there may be dirt in the needle and seat assembly, the float assembly may be contacting the bowl gasket, the seating surface of the needle and seat is damaged or the float level setting may be incorrect. If lightly tapping the carburetor with a soft object doesn't help, remove the float bowl assembly and check all float bowl components. If the needle and seat still do not seal properly, lapping them with toothpaste or household Comet cleaner (only on a solid brass needle and seat) is sometimes helpful.

Many shops carry quick change float bowl screws that reduce the time required for jet changes and maintenance on the B and D model carbs.

TUNING CIRCUITS-S&S

The S&S Super carburetor is relatively easy to tune if you understand its characteristics. Its two standard tuning circuits, the intermediate and main, control the fuel mixture across the entire rpm range. Depending on engine displacement and throttle butterfly position, the intermediate jet primarily controls the fuel mixture to about 3500 rpm. Starting between 3000 and 3700 rpm, the high speed main jet starts to activate and mostly controls the fuel mixture up to maximum rpm. There is an overlap or transition period between the two circuits that results in both circuits being active at the same time. This is discussed later. An adjustable idle mixture screw controls the fuel mixture at engine idle. The Super's two standard tuning circuits are defined as follows:

Intermediate Jet	Off idle response and low to midrange.
Main Jet	Midrange and top-end.

For correct results, the tuning circuits should always be adjusted in the order listed below.

A Super carburetor can be ordered with a complement of intermediate and main jets that are baselined for a particular engine displacement. These jets should only be used as a starting point because they may not be the best combination for your engine.

When tuning any jetting circuit, the engine should be thoroughly warmed to normal operating temperature and the carb's enrichment device must be fully disengaged for accurate results.

Idle Mixture Screw

The idle mixture screw regulates the air/fuel mixture at idle speeds. It is externally located either on the top left of the carb's body (Super E and G) or the top center (Super B and D). Be careful never to force this brass screw against its seat because this can distort the screw's tapered end and the aluminum seat in the carburetor. This will make it difficult to obtain the proper idle mixture. The idle mixture screw must be adjusted before any other jet circuits.

Initially turn the idle mixture screw clockwise until it gently seats and then turn it out (counterclockwise) 1-1/2 turns. This is a good starting position and should be relatively close to its final position. If this is the first time the engine is started with the Super carb, have a screw driver handy for quick idle mixture adjustment in case the engine refuses to keep running. Once the engine is running, warmed up and the enrichment device is fully disengaged, set the idle speed screw so the engine idles between 1000 and 1100 rpm.

Turn the idle mixture screw counterclockwise (out) to enrich the mixture and clockwise (in) to lean it. First, turn the screw in until the engine speed starts to drop and note its position. Next turn the screw counterclockwise until the engine speed starts to drop and again note its position. Now set the screw in the middle of these two points.

The idle mixture screw's final position can provide an indication whether the intermediate jet is the proper size. If the engine runs best with the mixture screw turned out less than one turn, the intermediate jet is too large. If it is greater than 1-3/4 turns, the intermediate is probably too small.

Intermediate Jet

The intermediate jet controls the fuel mixture from immediately off idle to approximately 3000 to 3500 rpm. It is located inside the float bowl area on all Super carbs. There are nine different intermediate jets available ranging in size from

.025 to .040-inch. Note that early model intermediate jets are marked #1, #2, etc.

After adjusting the idle mixture screw, take the bike for a ride and allow all engine parts to normalize in temperature. Now you're ready to tune the intermediate jet. The correct intermediate jet will allow the bike to accelerate smoothly from a stop while slowly rolling the throttle on. Also, slowly roll the throttle open from various steady speeds between 30 and 60 mph. If the engine backfires or spits through the carb, the intermediate jet is too lean (too small). If the engine hesitates, is sluggish, unresponsive or backfires through the exhaust, the intermediate jet is too rich. Another symptom of richness is a flat instead of sharp sounding exhaust note.

Each time the intermediate jet is changed, the idle mixture screw must be readjusted. A slightly rich intermediate jet will normally give the best acceleration while a slightly lean jet gives the best gas mileage. Keep in mind that most street riding is in the 2000 to 3500 rpm range — exactly where the intermediate jet controls the mixture. As a result, this jet has a major affect on gas mileage — either good or bad.

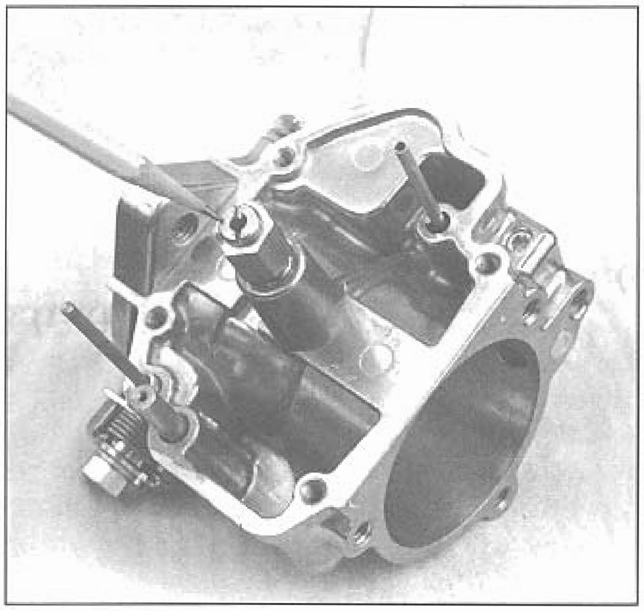
When tuning the intermediate jet, the accelerator pump should be turned off because it can conceal a lean mixture condition in the intermediate circuit. The pump can be reactivated when the intermediate circuit tuning is complete.

Main Jet

The main jet circuit includes the main jet, which controls the fuel mixture starting between 3000 and 3700 rpm and continues all the way up to maximum rpm. The main jet is always the last jet to tune.

The main jet screws into the bottom of the main discharge tube, which is the brass tube protruding from the bottom of the venturi area of the carbs throat. The jet can be easily reached by removing the drain plug located at the bottom of the float bowl. Main jets are available from .040 to .110-inch in increments of .002-inch. There also is a .115 and .120-inch jet. Remember that S&S jets are size rated.

Determining the optimum main jet is best done at the drag strip because the highest mile per hour in a given distance is the best indicator of maximum horsepower and optimum main jet size. Refer to the Jetting Guide Tables for a



The main jet for all S&S Super carburetors is located in the fuel bowl and screws into the bottom of the discharge tube (emulsion tube). Do not over tighten the jet because it could distort. The intermediate jet is seen protruding from the carb's body about 1-1/2 inches to the left of the main jet. Photo courtesy of S&S Cycle.

starting point. Keep in mind that altitude, air temperature and even humidity will have a significant affect on the correct jetting.

Warm the engine and make a drag run noting the final mph. Disregard the E.T. Increase the jet size (richer) by .004-inch, make another run and again note the mph. Be as consistent with the runs as possible. If you change to a larger jet size and the mph stays the same, keep going larger. Eventually this will force a change in mph to take place. Continue increasing the main jet size until the mph drops. At this point, decrease the jet size (leaner) by .002-inch and record the information. Also, by recording air temperature, humidity and altitude information you can establish a usable baseline for future reference.

For those limited to street riding, you can check the main jet by quickly closing the throttle from wide open to the 7/8 position when the engine is revving at least 4500 rpm. If the engine accelerates slightly, the main jet is too small (too lean). If the engine hesitates or misses slightly, the main jet is too large (too rich). If the engine just slows a slight amount, the main jet is close to being correct.

Another test requires accelerating rapidly through the gears at full throttle. If the engine backfires through the carburetor, misses, cuts out intermittently or quits running, the main jet is too lean so increase its size .004-inch. On the other hand, if the engine is sluggish to accelerate, will not accept throttle, has a flat sounding exhaust, or backfires through the exhaust, the main jet is too rich and should be reduced in size .004-inch.

Take note how easily and quickly the engine reaches the gear shift rpm level. The engine should accelerate smoothly and quickly through the gears. A rich condition will tend to make the engine's acceleration labored, strained and sluggish.

Experience has shown that a main jet .006-inch smaller (leaner) than the best size can make the engine intermittently cut out or quit running. Conversely, a main jet .006-inch larger (richer) than optimum will cause the engine to miss and sound flat. With a Super D carb, straight pipes, appropriate cam and head work, a good starting main jet size is about one thousandth of an inch for each cubic inch of displacement. For a Super B carb, use the same calculation, then subtract .010-inch from the total.

Accelerator Pump

The Super E and G carburetors include an adjustable accelerator pump for improved throttle response when quickly opening the throttle. The pump injects a squirt of fuel into the carb's throat, but only operates when the throttle is turned quickly during the first 1/3 turn of the throttle. The fuel volume delivered can be regulated by a pump adjusting screw located on the right end of the throttle shaft. By turning the adjusting screw clockwise until it contacts the pump's arm, the pump can be completely turned off.

Turn the adjustment screw three turns out from its full seated position and then test ride the bike from idle to 3500 rpm. Turn the screw clockwise in 1/2 turn increments until throttle response is reduced. Strive for the best throttle response with the minimum pump travel.

The accelerator pump cap assembly contains two small O-rings and two check balls. If the assembly leaks fuel, make sure the O-rings are not damaged or missing and the check balls are in place.

S&S Super 1-7/8" Carburetors Jetting Guide					
Carb Model	Super 1-7/8" A, B			Super 1-7/8" E	
Engine	74-86 Cu. In.	88-93 Cu. In.	96 Cu. In. & Up	74-84 Cu. In.	86-98 Cu. In.
Idle Mixture Screw	1-1/4 to 1-3/4 turns	1-1/4 to 1-3/4 turns	1-1/4 to 1-3/4 turns	3/4 to 1-1/4 turns	3/4 to 1-1/4 turns
Intermediate Jet	.0295 to .033	.031 to .035	.032 to .036	.028 to .033	.031 to .036
Main Jet	.068 to .075	.074 to .085	.078 to .090	.068 to .078	.072 to .086

Table 4.3

S&S Super 2-1/16" & 2-1/4" Carburetors Jetting Guide				
Carb Model	Super 2-1/16" G		Super 2-1/4" D	
Engine	88-93 Cu. In.	96 Cu. In. & Up	88-93 Cu. In.*	96 Cu. In. & Up
Idle Mixture Screw	3/4 to 1-1/4 turns	3/4 to 1-1/4 turns	1-1/4 to 1-3/4 turns	1-1/4 to 1-3/4 turns
Intermediate Jet	.0295 to .033	.031 to .036	.031 to .036	.032 to .036
Main Jet	.068 to .080	.072 to .086	.076 to .094	.082 to .108

* The Super D carburetor is recommended only for 96 cubic inch and larger Big Twin engines.

Table 4.4

REPLACEABLE AIR BLEED-S&S

In some situations a low speed stumble or flat spot may still exist after tuning a Super carburetor. Stock engines place relatively small fuel and airflow demands on the carburetor and the demands are only present between idle and 5200 rpm. High rpm, large displacement engines significantly increase the airflow demands against the carb and can extend the operating range for proper fuel metering to beyond 7000 rpm.

The two stock fuel circuits have a difficult time correctly metering fuel over the entire rpm range, particularly during the transition period between the intermediate and main circuits. As mentioned earlier, the intermediate jet primarily controls the fuel supply to about 3500 rpm. Between 3000 and 3700 rpm the main jet starts supplying fuel. To get good throttle response, the intermediate circuit must be jetted excessively rich. Tuning problems exist during the transition period because neither circuit operates independently of the other. With both circuits supplying fuel during the transition period, the air/fuel ratio becomes too rich and a flat spot develops somewhere in the engine's power band. If the main circuit is jetted for maximum top-end power, the midrange will be too rich. If the main jet is leaned down to eliminate the midrange flat spot, top-end power goes down due to an excessively lean condition. The Super's main air bleed system can be modified to help eliminate a flat spot during the transition period.

The main air bleed is an air passage that supplies air to the carburetor's main discharge tube (emulsion tube). This is the brass tube that extends vertically from the float bowl cavity to the venturi in the carb's throat. The main jet screws into the bottom of the emulsion tube. The emulsion tube includes a series of small holes that help mix air from the air bleed passage with fuel from the float bowl. This turns the mixture into an air/fuel emulsion before it is drawn to the venturi's low pressure area. The emulsified mixture vaporizes much easier when discharged into the high speed airstream flowing through the venturi. It also has a lighter viscosity than liquid fuel and responds quicker to any change in vacuum signal at the venturi. The emulsion's lighter viscosity allows it to start flowing quicker than liquid fuel. The number, size and pattern of the holes in the emulsion tube influence fuel delivery. A key point to note is that some racers have improved their drag strip 60 foot time by soldering closed the standard S&S air bleed holes and drilling a new set.

As airflow increases through the carb's venturi, fuel flow also increases through the emulsion tube, but at a disproportionately *greater* rate than the airflow. This enriches the mixture as rpm rises because the air's density is reduced as velocity increases. Additionally, because fuel is much heavier than air, the fuel tends to keep spraying from the emulsion tube even after the intake valve has closed, particularly at high rpm. The series of small holes in the emulsion tube, along with the total amount of air supplied to the cavity surrounding the emulsion tube provide air-correction and keep the mixture from over enriching.

The size of the main air bleed's orifice (.040-inch for the Super D, .042-inch for all other models) determines when the main jet circuit begins supplying fuel to the engine. With the standard size air bleed, fuel is supplied by the main jet circuit starting at a relatively low 3000 to 3700 rpm, depending on engine displacement and throttle butterfly position. A flat spot frequently occurs because the main jet circuit begins supplying fuel too early. Increasing the size of the air bleed raises the rpm level at which the main jet circuit begins supplying fuel and reducing its size lowers the point where the main jet kicks in. This is because air is lighter than fuel and the lighter air settles to the top of the emulsion tube's chamber. With more air at the top it takes longer for the fuel to start moving.

In essence, a larger air bleed creates more separation between the intermediate and main fuel circuits. This provides correction for the excessively rich transition period. Now the main jet can be jetted as rich as is necessary for maximum top-end power without creating a flat spot during the transition period. This takes place because the main jet begins working at a higher rpm, which essentially removes some fuel from the intermediate circuit. Now the intermediate jet can supply fuel during midrange rpm levels without the main jet supplementing it as much.

The adjustable air bleed was first discovered about 1970 by racers running an early model S&S nitromethane carb. They found that increasing the air bleed size corrected an engine cutting out problem caused by an excessively rich condition.

With an adjustable main air bleed, the Super's tuning circuits are now separated into three categories:

Intermediate Jet	Off idle response and low to midrange.
Main Air Bleed	Midrange fine tuning.
Main Jet	Midrange and top-end.

Tuning procedures for an S&S Super carburetor with an adjustable air bleed are described under "ThunderJet Tuning."

The main air bleed is made adjustable by installing a replaceable jet in the location of the air bleed's original fixed orifice. For smooth power over a broad rpm band, this modification is recommended for all S&S Super carburetors. However, if you don't achieve better results with it, just install a .042-inch jet to set the bleed hole back to stock size.

With the Super B and D carb, the main air bleed is located on the throttle shaft return spring side (left side) of the carburetor's body, slightly below and to the left of the model number designator. Late model Super D carbs are shipped from the factory with an adjustable air bleed.

Installation of an adjustable air bleed requires drilling the main bleed hole with a 1/4-inch drill bit until the cross passage in the carb body is reached. The hole's surface should be spot faced with a 1/2-inch end mill then tapped with a 5/16-24 start tap and followed by a bottom tap. The air bleed is now setup to accept an S&S main jet. A jet between .054 and .070-inch should allow tuning the Super B or D carburetor for maximum midrange throttle response and maximum top-end power.

On Super E and G carburetors, the main air bleed is found on the underside of the carb body in the fuel bowl cavity area. Its exact location is on the throttle cable spool side in the deep recessed area. This hole must be blocked off so the air bleed can be relocated to an external location for easier access.

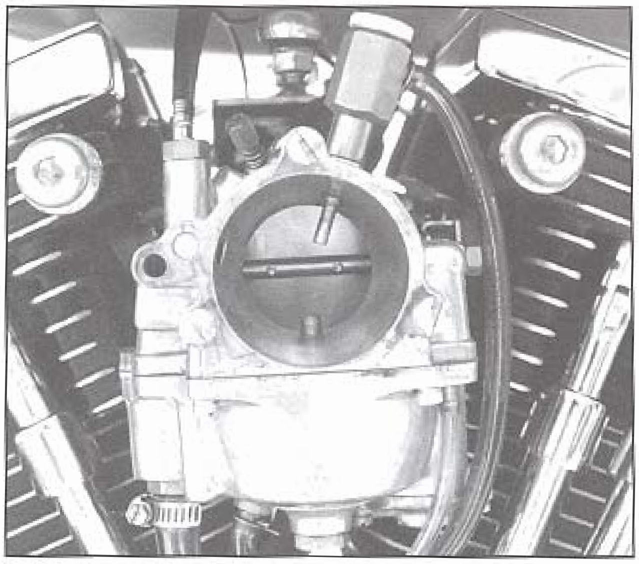
Start by boring the hole with a .159-inch (#21) drill bit until the cross passage in the carb body is reached. Then thread the hole with a 10-32 tap and plug it with a 10-32 Allen head screw.

Next, locate the external main air bleed passage plug that is positioned on the throttle cable spool side of the carb body (above and to the left of the S&S logo). Drill out the plug with a #19 drill bit and tap the hole with a 5mm by .8mm tap. Finally, install a Mikuni *round* main jet as the replaceable air bleed jet. A jet between 130 and 170 (.055 to .067-inch) should allow tuning of the Super E or D carburetor for maximum midrange throttle response and top-end power.

THUNDERJET

As pointed out earlier, an unmodified Super carburetor includes two fuel circuits. Even when an adjustable main air bleed is added, there still remains only two circuits. What the adjustable air bleed provides is more flexibility in tuning the existing two fuel circuits. On the other hand, installing a ThunderJet adds a fully adjustable, self-correcting third jetting circuit to the carb.

The ThunderJet works in conjunction with the main jet by supplying fuel to the engine at very high rpm. It senses acoustic wave pulses in the intake tract and instantaneously responds to the pulses by siphoning fuel from the fuel bowl. As the pulses get stronger, the ThunderJet supplies more fuel to the engine. The actual rpm it starts supplying fuel is dependent on the engine's displacement and the carb's throttle butterfly position, but it usually starts about 4500 rpm. Because the ThunderJet supplements the main jet in supplying fuel, the size of the main jet now can be reduced. This lessens the potential for the main jet to enrich the intermediate circuit to the point where a flat spot develops somewhere in the rpm band.



A ThunderJet is installed in this S&S Super E carburetor. The ThunderJet provides an additional fuel circuit that improves the fuel curve between the intermediate and main jet tuning circuits. A ThunderJet can be installed on many butterfly style carburetors. Photo courtesy of Perfection Cycle.

The ThunderJet consists of four major parts: ThunderJet body, fuel jet, air correction spacer, and fuel delivery tube. For installation, the carburetor body must be first drilled and tapped for the fuel delivery tube. The tube protrudes into the carb's throat and supplies the fuel. In the case of the Super B, the delivery tube is normally positioned horizontally through the carb's left side wall (throttle return spring side) at the nine o'clock position. Due to differences in the Super D's casting, the tube's position should be rotated 180 degrees to the three o'clock position. For Super E and G carb's, the delivery tube is installed at the one o'clock position.

The tube is positioned .750-inch from the air cleaner mounting surface on the Super B and D models and .950-inch on the Super E and G. The next step is to modify the standard main air bleed passage (as described under "Replaceable Air Bleed") to accept an adjustable jet. Now a small fuel line fitting is installed into the bottom of the float bowl to supply fuel to the ThunderJet circuit. Finally, the air correction spacer and ThunderJet body are screwed onto the fuel delivery tube and the thin fuel line is connected to the float bowl fitting and the ThunderJet body.

THUNDERJET TUNING

Installing an adjustable main air bleed does not require the installation of a ThunderJet; however, an adjustable air bleed is always installed in conjunction with a ThunderJet. With a ThunderJet installed, the S&S Super now has four tuning circuits — three supply fuel and one supplies air. The tuning circuits are now defined as follows:

Intermediate Jet	Off idle response and low to midrange.
Main Air Bleed	Midrange fine tuning.
Main Jet	Midrange and top-end.
ThunderJet	Top-end.

Since the ThunderJet adds a third fuel tuning circuit, the original tuning and jetting guidelines no longer apply. Therefore, the ThunderJet Jetting Guide (see Table 4.5) should be referenced for initial jetting recommendations.

THUNDERJET JETTING GUIDE

Engine	Intermediate Jet	Main Jet	Air Bleed	ThunderJet
74-86 Cu. In. Super B	.033 – .036	.080 & Up	.056" – .072"	100 – 130
74-86 Cu. In. Super E	.032 – .037	.078 & Up	135 – 165 .056" – .066"	100 – 125
88-93 Cu. In. Super B	.033 – .038	.084 & Up	.056" – .072"	100 – 140
88-93 Cu. In. Super E & G	.033 – .038	.084 & Up	135 – 165 .056" – .066"	100 – 135
96 Cu. In. & Up Super G	.038 – .040	.078 & Up	135 – 165 .056" – .066"	120 & Up
96 Cu. In. & Up Super D	.038 – .040	.056 & Up	.056" – .072"	120 & Up

Use these jet sizes only as a starting point. Final jetting will vary depending on the exact engine combination and atmospheric conditions.

Table 4.5

With four jets to tune besides the idle mixture screw, more experimentation, time and patience are required for precise carburetor adjustment.

1.) Start tuning a ThunderJet modified S&S carb by first installing the recommended baseline combination of jets. Turn off the accelerator pump (if present) and thoroughly warm up the engine. Make notes about the jet combinations you test, including information about performance and the rpm range of any problems.

2.) Notice that the ThunderJet Jetting Guide (Table 4.5) recommends a much richer intermediate jet than the Standard Jetting Guide. The intermediate jet directly affects off idle and low speed throttle response. The ThunderJet allows this circuit to be enriched for additional low range power without introducing tuning problems in other areas of the rpm band. In general, for the best throttle response, use as large an intermediate jet as possible. However, keep in mind that gas mileage will suffer.

3.) With a large intermediate jet, the standard range of adjustment for the idle mixture screw must be disregarded. This adjustment normally needs to be leaned from the standard setting. The final setting may be as little as 1/2 turn out from its seated position. Do not enrich this adjustment to compensate for a too lean intermediate circuit and remember that the idle mixture must be readjusted whenever any jetting is changed.

4.) With a relatively large intermediate jet, off idle throttle response should be improved. However, a flat spot may develop due to an over rich condition at the rpm level where the main jet starts to become active. An excessively rich condition during the transition from the intermediate to main circuit can be eliminated by increasing the air bleed size so the point where the main jet starts delivering fuel is delayed. If you're not sure which way to adjust, start increasing the air bleed size in small increments until throttle response gets worse.

5.) Once you have the intermediate and air bleed circuits as good as you can get them, start tuning the main jet circuit for high midrange and top-end power. Refer to the previously discussed "Main Jet" topic for the exact procedures. Remember that there is a relationship between the air bleed and the main jet.

6.) The ThunderJet circuit is tuned after the main jet and it's done in the same manner. Tune for highest mph in a given distance without encountering any backfiring, missing or intermittent running. Remember that at high rpm all three fuel circuits are supplying fuel to the engine. Adding fuel to one circuit enriches the mixture throughout the rpm range and may require that fuel be removed from another circuit.

INTAKE MANIFOLD-S&S

S&S carburetor kits include a new one piece rigid manifold. All S&S manifolds have good runner angles and the 2-1/16 and 2-1/4 inch models have a relatively large plenum area. This manifold offers excellent performance and in stock condition flows about seven percent more than the factory Evolution manifold. Its rigid design also eliminates the need for the troublesome compliance fittings used for sealing on the 1984 to 1989 Evolution Big Twin. Be sure to order the carburetor kit with the appropriate manifold (Evolution, Shovelhead, O-ring or rubberband type) for your engine.

Cylinders longer or shorter than standard generally require a non-standard manifold width. S&S offers a selection of manifolds that fit most non-standard cylinder lengths. Be sure to order the correct width manifold.

Before installation, remove all machine and casting marks in the manifold's interior and finish with a 60-grit cartridge roll. All bends should have smooth radiuses and the manifold's inside diameter, where it mates to the head, should never be larger than the port.

A one inch spacer block comes standard with the Super D and is optionally available for the other models. The block not only enlarges the plenum area, but also lengthens the intake tract, which changes the timing of the tract's pulses. This affects the engine's power band and can sometimes help control fuel "stand off."

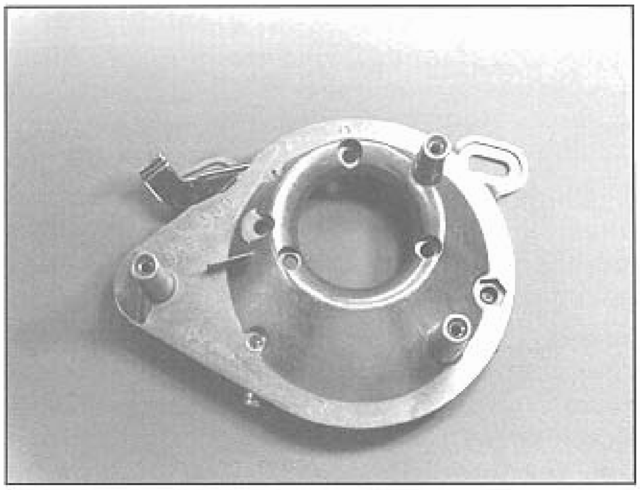
Always use new O-rings or rubberbands when installing the manifold and take note that the Evolution manifold's two mounting flanges cannot be interchanged. For Shovelheads, consider replacing the flimsy stock manifold clamps with stainless steel aircraft-style clamps that are available from S&S and other suppliers.

If your ignition does not use a Vacuum Operated Electric Switch and your manifold includes a vacuum fitting, block the fitting off.

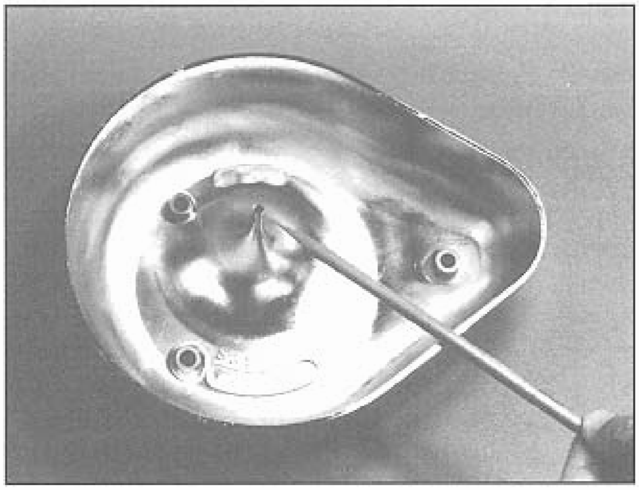
For additional information, refer to "Intake Manifolds."

AIR CLEANER-S&S

The traditional S&S teardrop air cleaner includes a radiused entryway on the backplate and an air directional cone on the cover's inside. This design captures turbulent air, smoothes it out



Notice the radiused entryway and air directional cone on the inside of the S&S air cleaner. This design increases airflow by capturing air and smoothing it out while directing it into the carb's throat.



and then directs it in a laminar manner into the carb's throat for maximum flow.

In large displacement engines where no air cleaner element is used, airflow can sometimes be improved by reducing the diameter of the air cleaner's backplate. This modification increases the space between the backplate and front cover where the air enters the filter area.

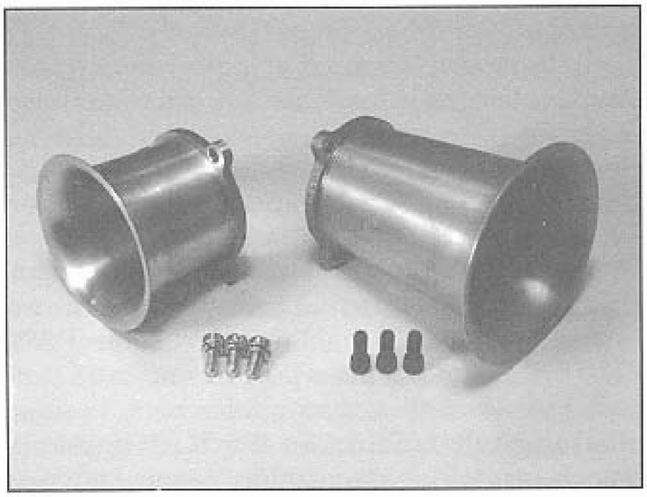
The standard filter element is made of synthetic foam. For increased flow, this unit can be replaced with a washable and reusable K&N filter element. This filter provides up to 45 percent greater airflow than a foam unit, besides offering excellent filtration. The K&N unit uses four layers of pleated cotton gauze material sandwiched between an aluminum wire grid. This results in low restriction and the pleats significantly increase the filtering area. K&N

filters periodically need washing with a household cleaner like Formula 409. After washing, thoroughly dry the element and reoil it with the correct filter oil to trap airborne debris.

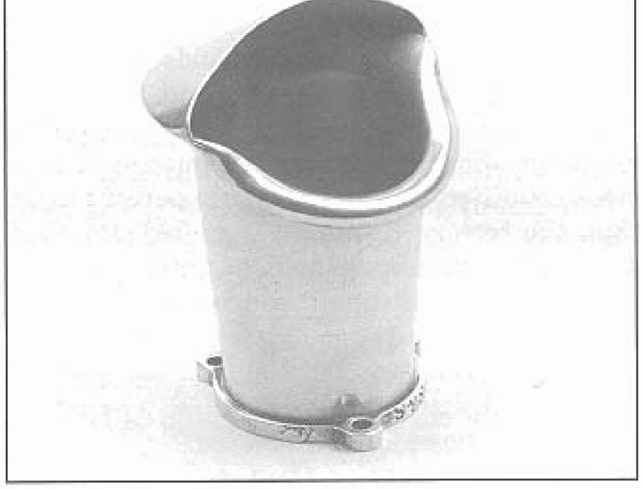
When using the air cleaner backplate and cover without a filter element or when using a non-standard filter element, be sure the float bowl vent located at the air cleaner end of the carb (at either the three o'clock or nine o'clock position) is not obstructed. With Super E and Super G models, remove the bowl vent screw located on the throttle cable spool side of the carb body near where the air cleaner backplate mates to the carb throat. This will ensure that the pressure in the fuel bowl cavity is equivalent to the ambient atmospheric pressure. Keep in mind, that less pressure in the bowl cavity makes the fuel mixture lean; more pressure makes it rich. Bowl vents can be too small, but they can't be too large.

VELOCITY STACK-S&S

For maximum airflow, the air cleaner assembly can be replaced with a velocity stack. Depending on many factors, a velocity stack can improve cylinder filling and control fuel "standoff." The Super B and Super D stacks are 4-inches long while 2-1/2 and 4-inch lengths are available for the E and G models. In most cases, the longer length seems advantageous.



A velocity stack is used during racing to reduce turbulence, improve airflow and change the intake tract's tuned length. Shown are 2-1/2" and 4" stacks. Photo courtesy of S&S Cycle.



A deflector shield is sometimes added to a velocity stack to increase the intake tract's air pressure and reduce turbulence from high velocity side winds. Photo courtesy of Carl's Speed Shop.

Carl's Speed Shop offers a custom designed hand made velocity stack that includes a deflector shield on its outer end. This design offers the advantage of increasing intake air pressure by scooping the air. It also helps when high velocity side winds and turbulent wind buffeting are present.

Changing the length of the velocity stack changes the length of the intake track, which changes the timing of the acoustical wave energy pulses traveling within the intake tract. This pulse can have a significant affect on cylinder filling at high rpm levels.

Keep in mind, that using a velocity stack increases airflow, therefore jetting usually needs to be enriched.

When using a velocity stack on a Super E or G carb, don't forget to remove the float bowl vent screw located on the throttle cable spool side of the carb body near where the velocity stack mates to the carb throat. This will ensure that the pressure in the fuel bowl cavity is equivalent to the ambient atmospheric pressure.

When using a deflector shield stack design, consider venting the float bowl cavity to the intake tract's high pressure instead of ambient atmospheric pressure. This ensures the float bowl pressure is matched to the high pressure in the intake tract and will allow fuel to flow freely from the float bowl through the jets to the carb's venturi. This modification won't show any benefits on a dyno, but it will help when the bike's speed is greater than 100 mph. The standard air bleed location also may need to be changed.

SUPPORT BRACKET-S&S

Engine vibration can easily cause the carburetor to shift, resulting in manifold leaks and carburetion problems. The Evolution uses a carburetor support that bolts the air cleaner backplate directly to the cylinder head. The Shovelhead uses a support bracket that extends from the backplate to the cylinder head. For either engine, always use a support bracket and keep it tightly fastened.

A different support bracket may be required when switching from an air cleaner to a velocity stack. Zippers Products sells a heavy duty bracket for the Super D carb. This bracket provides outboard support and does not interfere with mounting a velocity stack or air cleaner.

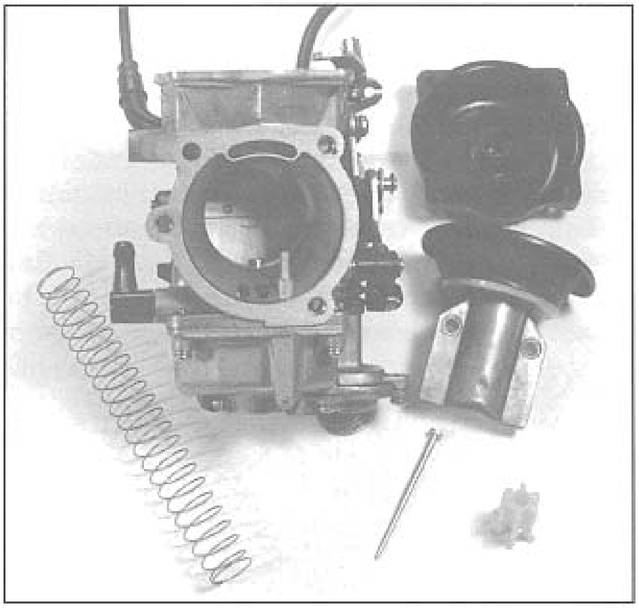
KEIHIN CV MODIFICATIONS

The stock 1990 and later Keihin CV carburetor flows more air and makes more power than earlier Big Twin carburetors. If you intend to retain this carb, a few modifications will help its performance. But be aware that the cost of modifying the Keihin can approach the cost of a high-performance carburetor.

The Keihin works by the constant velocity (CV) principle and uses a throttle butterfly to regulate airflow and a vacuum operated slide piston to create a variable size venturi. At any given time, the size of the venturi is determined by the vacuum level in the intake tract. The variable venturi keeps air velocity high, which results in good low speed throttle response.

Stock engines are too lean in the midrange (slow jet) and slightly rich on the top-end (main jet). DynoJet makes a jet recalibration kit that includes all necessary jets, needles, drills and springs to rejet the carb for better performance.

If you don't install this kit, you can still make modifications to help performance. First, the CV's slow jet must be enlarged to help midrange power. The standard non-California slow jet is a #45. Its orifice is .45 millimeter, which calculates to about .0177-inches. Its size should be increased to between .019 and .020-inch to help midrange performance.



Improvements can be made to the stock 1990 and later Keihin CV carburetor. The piston's vacuum hole can be drilled larger for better throttle response and its bottom edge radiused for better flow. The needle can be raised slightly and the jetting changed for a better fuel curve. DynoJet makes a recalibration kit that includes the necessary parts and instructions.

Raising the needle in the slide also helps midrange performance. Place a .025-inch washer under it to help enrich the midrange. Additionally, performance is sometimes helped by removing .0005-inch from the diameter of the needle's tapered section.

There is a vacuum hole (about .100-inch in diameter) in the top of the slide next to the hole the needle goes through. Use a #31 drill bit to increase its size to .120-inch. This makes the slide more responsive when quickly opening the throttle. A lighter tension slide spring will also increase the slide's reaction.

To increase the CV's airflow, you can gently radius the front bottom edge of the slide. This edge can be seen when looking through the carburetor's throat from its air cleaner mounting end.

The idle mixture screw is located behind the float bowl in the drop-down cylinder covered by an aluminum plug. Adjust the screw for a smooth idle when the engine is warm. If the engine backfires through the carb during acceleration, the idle mixture is too lean.

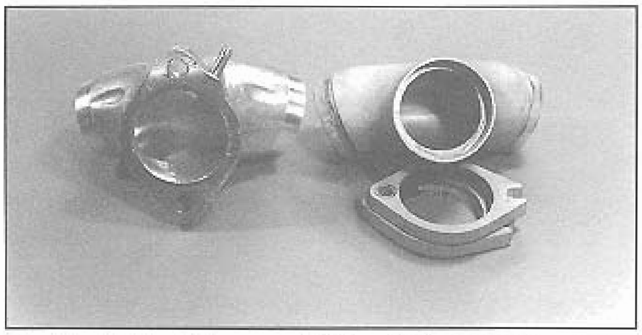
The intake manifold and air cleaner assembly must be replaced for greater airflow. A Screamin' Eagle performance manifold and air cleaner kit will improve flow. The manifold's interior provides a better angle to the ports and the air cleaner has a radius inlet ring that helps smooth air into the carb's throat, thereby reducing turbulence and increasing flow.

If the stock air cleaner assembly is retained, the internally located baffle plate can be removed to help increase airflow. This is the flat steel quarter-moon shaped plate located next to the air cleaner element.

INTAKE MANIFOLDS

The Evolution and Shovelhead both use a single stubby Y-style manifold to feed both cylinders. The interior shape of the Y-manifold is as important as the carburetor's flow capacity for producing maximum power. None of the Big Twin manifolds are very efficient at getting air into the cylinder.

The Shovelhead and early Evolution (1984–1989) manifolds direct airflow at a relatively decent 45 degree angle toward the ports, but the interior corners are too sharp and cause turbulence and air/fuel separation. The early Evolution manifold is also plagued by troublesome compliance fittings that allow air leaks to develop easily. The 1990 and later Evolution manifold solved the compliance sealing problem and has a smooth interior with gentle radiuses, but its back wall is too straight, which forces the airflow to turn at close to a 90 degree angle. Additionally, with large carburetors, the volume of all stock manifolds is too small for maximum power. Regardless of the carburetor used, to achieve maximum performance the intake manifold must be replaced with one that has its interior reworked for improved flow.



The Big Twin's intake manifold is a big bottleneck to performance. Notice the difference between the stock Evolution manifold (right) and a reworked S&S. The 45 degree bend should be properly radiused and the manifold must fit to the heads without a step.

The intake manifold should be viewed as an extension of the intake ports and most of the flow principles that apply to the intake ports also apply to the manifold. The manifold should always be included when reworking cylinder head ports and both should be developed and flow tested together.

Compared to an automotive V8 intake manifold, the Big Twin manifold is relatively simple. It includes a common plenum area and two intake runners. Flow loss is mainly attributed to a change in air direction and a decrease in air velocity due to expansions and bends. The major concern is getting the air to bend around the 45 degree corner between the carburetor and the port. This corner needs a large gentle radius for maximum air velocity and flow. The center ridge that splits the two manifold runners also can cause turbulence and must be radiused and smoothed. Additionally, the manifold's plenum volume may be increased and its runners may be shortened or lengthened.

Intake Pulses

Because of the V-Twin's design, one spark plug is often colored lighter than the other. Usually the plug coloring indicates the front cylinder is running leaner than the rear. Some racers attempt to cure the situation by running a colder plug in the front cylinder, but this is only a stopgap measure.

The Big Twin's 45 degree cylinder angle dictates that the time between the intake pulses for each cylinder is different. The rear cylinder fires 315 degrees after the front and the front fires an additional 405 degrees after the rear, completing the 720 degrees of crankshaft rotation required for the four engine strokes. The front cylinder's intake valve opens and closes and is then followed very closely by the opening and closing of the rear cylinder's intake valve. The time interval between the front and rear cylinders filling is less than between the rear and front cylinders filling. Consequently, the rear cylinder gains from the air/fuel movement that already has been initiated in the manifold and port during the filling of the front cylinder.

The longer time before the filling of the front cylinder reduces the momentum of the air/fuel stream and causes the heavier fuel to drop partially out of suspension with the air. Since air is lighter than fuel, it responds quicker than the heavier fuel to the changing intake pulse initiated by the opening intake valve. This results in the front cylinder receiving a greater amount of air and less fuel, which causes a lean condition.

Some racers address this problem by biasing the carburetor through angle-milling the manifold's face two to three degrees towards the front cylinder. Other racers increase the plenum area to reduce the strength of the intake tract's energy pulses.

Most V-Twins that have a common intake manifold experience some degree of uneven cylinder filling. Factors such as camshaft specifications, rod ratio, engine displacement, rpm range and head design are all linked to how pronounced the characteristic is. For example, a Shovelhead's front cylinder head that is modified by porting, but without any welding, usually will flow more than the rear head. The different flow ratio between the heads seems to minimize the lean condition and less manifold correction is required.

Plenum Area

The Big Twin's 45 degree cylinder angle is not the only factor that affects the induction tract. As each cylinder draws from the plenum, the air/fuel streams are constantly changing direction due to intake pulses. The strength of the pulses is partially determined by cam timing, valve overlap and whether the negative exhaust pulse reaches the intake tract during the overlap period.

When the engine is operating in the rpm band for which the exhaust system is tuned, the negative pulse from the exhaust arrives during the valve overlap period and starts the intake flow early. Outside this rpm band, the exhaust's positive wave arrives during overlap and it can blow back through the intake tract and negatively affect the carburetor's ability to meter fuel accurately. Since the carburetor allows gases to move in either direction, air/fuel mixture from the combustion chamber and exhaust gases are blown into the intake tract past the carb's venturi. The mixture then reverses direction and passes the venturi for a third time as the other cylinder draws in a charge. This results in a diluted air/fuel mixture that can cause flat spots and reduced power. This is evidenced by the fuel fog or fuel "standoff" that is seen at the carburetor's entryway.

Enlarging the manifold's plenum chamber sometimes is a solution to power losses and poor carburetion caused by intake tract pulsations and distribution problems. A larger plenum chamber increases the volume of mixture that must be drawn when the intake valve opens. This cushions the pulsations in the intake tract so they are less harsh, which can provide more accurate fuel metering and possibly an increase in flow. The larger the plenum volume is, the greater the softening effect that takes place. However, keep in mind that throttle response and low-end performance may be reduced, depending on the engine combination and the size of the plenum area. This takes place because as the pulses are softened and the signal at the venturi's spray nozzle is reduced. Also, the mixture can lean, especially at low rpm levels. Richer jetting and more fuel volume from an accelerator pump are typical characteristics for engines with a large plenum manifold.

A large plenum area is generally associated with large displacement engines and high rpm operation. Also, cylinder heads with high velocity ports work well with a large plenum chamber. Due to the elusive interrelationships of engine displacement, cam specifications, carburetor size, rpm range, head design and even rod ratio, the best way to determine whether a larger plenum is beneficial is to have different sizes of plenum manifolds built. Then test each one at the track after the rest of the engine's tuning systems are dialed in.

There are a number of methods for increasing plenum volume. The simplest method is to add a one inch spacer block between the carb and the manifold. This sometimes will reduce fuel "standoff" and help improve overall carburetion. Although the spacer block adds plenum area, it also adds to runner length and affects the timing of the intake tract's energy pulses.

The S&S Super G and D manifolds have a relatively large plenum area and they can be modified to fit carbs with a smaller throat diameter. For small carbs, this offers a relatively easy method for installing a large plenum manifold.

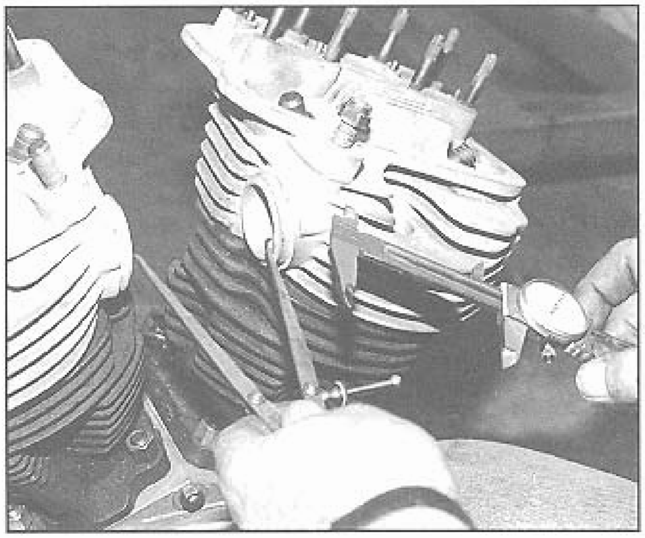
One method for mounting a small carb to a large manifold is to use a spacer that gradually tapers from the smaller carburetor diameter to the larger manifold. Trock Cycle makes a tapered adapter.

Another method is to align the bottom of the carb's throat even with the bottom of the manifold's throat, instead of centered. This minimizes eddying airflow where the carb's throat abruptly drops off to the larger diameter manifold. Eddying is turbulence in the airstream that reduces flow and the kinetic energy of the air/fuel charge. It also causes fuel to separate from the air and puddle on the manifold's floor.

Some head porters build custom plenum manifolds by applying aluminum weld to the manifold's exterior and recontouring its interior passages. Johnstone Products offers a large plenum billet aluminum manifold that fits various style carburetors.

Manifold Fitting

The intake manifold must be the proper width to provide an air tight seal. Since the Big Twin's cylinders are set at a 45 degree angle, increasing or decreasing cylinder length as little as .050-inch requires a change in the manifold's width. Stroker engines most likely require extra long cylinders.

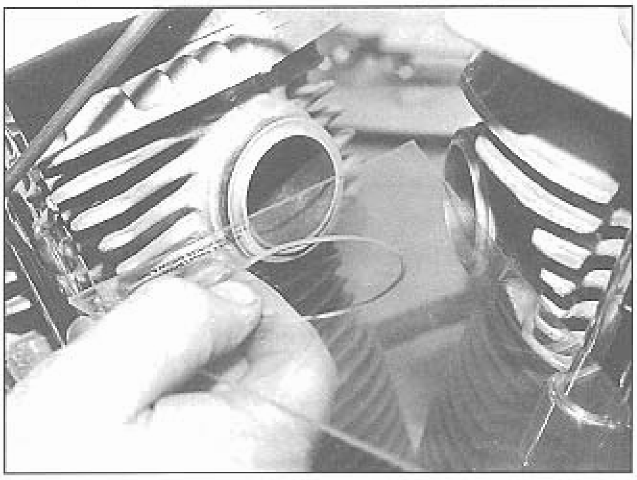


Long cylinders require a wide manifold. The correct manifold width can be determined by either measuring cylinder length or by measuring the distance between the intake ports on a mocked up engine. The picture shows the correct way to measure manifold width. Photo courtesy of S&S Cycle.

S&S makes Evolution and Shovelhead manifolds extra wide for long cylinder lengths (in cylinder increments of .050-inch). Cylinder length is measured from the cylinder's base gasket surface to head gasket surface. Don't forget to include the height of any stroker plates in the measurement. Be aware that machining the cylinder head on the head gasket surface reduces the effective cylinder length and requires a narrower manifold for a given cylinder length.

Aftermarket manifolds and 1990 and later Evolution manifolds are designed for rigid mounting, which eliminates the leak prone rubber compliance fittings on 1984 to 1989 Evolutions. These manifolds include a front and rear mounting flange that cannot be interchanged. Shovelhead manifolds are available in either O-ring (early) or rubberband (late) design.

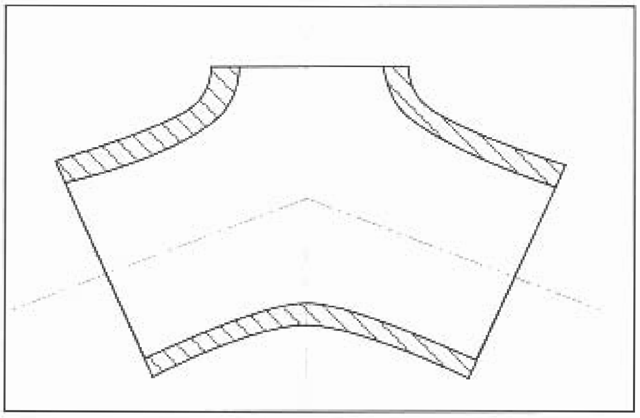
It's worth spending the necessary time to fit the manifold properly to the head ports because any gap between the manifold and cylinder head port will create turbulence that reduces airflow and separates the air/fuel mixture. During engine assembly, the cylinder heads must be positioned to match the angle of the intake manifold. The correct angle for a Shovelhead's intake ports is 60 degrees, while for the Evolution it is 30 degrees. Before tightening down the head and/or base bolts, you should use either a 30 or 60 degree triangle to verify the alignment of the intake ports. Turn the heads and cylinders so the triangle touches each port in two places simultaneous, then tighten down everything.



The manifold must fit squarely against the intake ports. Use a 30/60 degree triangle to check the angle before tightening the heads down. Turn the heads so each port simultaneously touches the triangle in two places. The correct angle is 30 degrees for the Evolution and 60 degrees for the Shovelhead. Photo courtesy of S&S Cycle.

Manifold Modifications

All stock intake manifolds can be improved by reworking their interior. This is particularly true for Shovelhead manifolds that have sharp interior corners. Some high-performance aftermarket manifolds include a finished interior while others require reworking to various degrees. Manifolds with a "downswept" design provide additional gas tank clearance; however, they generally do not flow as well as a straight manifold design.



If you're having the heads ported, get the intake manifold reworked at the same time. If you're installing a new carburetor, finish the manifold's interior with gentle radiuses and remove all machine and casting marks. Also, make sure there is no ridge where the manifold connects to the carb and intake ports.

Since the manifold is an extension of the intake ports, the manifold and ports should be reworked and flowed together as a set. If you're having the cylinder heads ported, send the manifold with the heads to the head porter. High-dollar pro stock performance heads frequently require the head porter to fabricate a special manifold to accommodate large diameter, raised ports. In this case, check with your head porter before investing any money or time modifying a manifold.

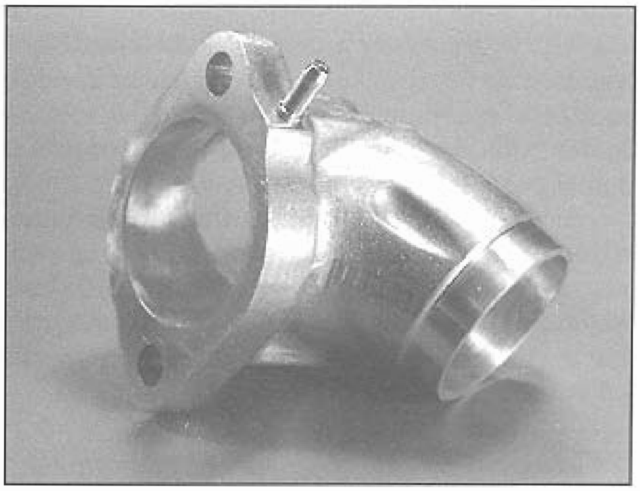
Some manifolds only fit between the cylinder heads one way, while others can be turned upside-down. Before reworking the manifold or sending it to the head porter, mark its top so it is always installed in the same position.

The main objective for reworking the manifold is to maximize airflow, reduce turbulence and minimize air/fuel separation. This requires the elimination of sharp turns and severe changes in cross-sectional area. If your ignition system does not have a vacuum retard but the manifold has a vacuum fitting, remove the fitting and plug its hole.

The manifold's two 45 degree short-side radiuses are the major restrictions in the manifold and should be reworked to help the air turn the corner at high speed between the carb and the port. Remember that air velocity is a function of an intake runner's cross-sectional area, but the velocity is not uniform across the entire area. Velocity is usually slowest on a turn's short-side radius and highest on its long-side. Smoothly blending this turn to a large, gently curving radius will help the airflow turn the corner and produce a more uniform distribution of flow pressure. This results in reduced air/fuel separation and increased net flow. If you're reworking a stock Shovelhead or early Evolution manifold, there usually is not enough material at the corner to get a large radius and the results may be limited.

The manifold's backside radius splits the two runners and it also should be shaped with a large gentle radius. Aftermarket performance manifolds come with this radius fairly well defined. However, early model Evolution and all Shovelhead manifolds need work in this area.

The manifold's interior should have no severe changes in cross sectional area. It also should have gently curving radiuses and all casting marks removed. The point where the



The interior shape of the intake manifold is just as important as the carburetor's flow capability. For maximum performance, have your manifold ported and make sure it fits properly. Also remember that the longer the cylinders, the wider the manifold must be.

manifold runners and the ports meet should be matched for size. In no instance should the manifold be larger than the port. Be sure you spend the necessary time to match the manifold to the ports. There usually is little excess material on the manifold's walls for increasing the plenum area. Therefore, if you intend to run a large plenum, you generally need to start with either an S&S G or D manifold or fabricate your own.

To help fuel vaporization, the manifold's interior should not be polished. Instead, finish it with a 60-grit (maybe as coarse as a 40-grit) cartridge roll. Polished runners allow fuel to puddle similar to how a drop of water puddles on a flat plate of glass. The 60-grit finish creates a thin layer of turbulence that helps redistribute the fuel into the airstream. The flow loss attributed to the additional wall friction is no more than about one percent and it is more than offset by the gains derived from better fuel atomization.

Manifold Clamps

Shovelheads from 1966 to 1978 and some 1979 models use O-rings to seal the manifold to the cylinder head. Some 1979 Shovelheads and all 1980 through 1985 Shovels use a rubberband for sealing. For good sealing, replace the flimsy stock intake clamps with heavy duty stainless steel aircraft-style clamps. Evolutions use a special design O-ring and cast mounting flange for sealing.

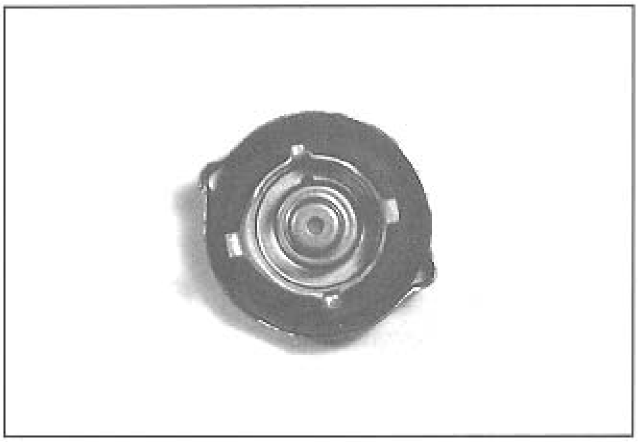
For large heavily modified Shovelhead style manifolds, automotive radiator hose of the correct size can be cut into sections for sealing bands and used with automotive clamps for sealing the manifold to the heads.

Intake manifold leaks can cause havoc on carburetion. Check for leaks (only in a well-ventilated area) by spraying WD-40 on all three adjoining surfaces while the engine is idling. An air leak is indicated by any change in engine speed.

FUEL DELIVERY SYSTEM

The Big Twin's fuel delivery system encompasses everything from the gas tank to the carburetor fuel bowl. It includes the gas cap, fuel valve, fuel line, filter, fittings, and the needle and seat assembly. The system is gravity fed and is responsible for maintaining a constant fuel level in the fuel bowl. The fuel level has to be maintained during normal street riding conditions or racing conditions, which may include violent launches, hard acceleration, harsh braking and tight cornering.

Sufficient fuel flow is required to maintain the correct air/fuel ratio and this requires proper venting of the fuel tank. The tank is vented through the gas cap and if the vent isn't large enough, the tank can pressurize and make the engine run too rich. It also can create a vacuum in the tank and cause the engine to run lean. The gas cap's vent system can be modified for race track use to ensure the tank's pressure is equivalent to atmospheric pressure. If your tank has two gas caps, the right side cap (located on the high side of the tank when the bike is resting on its kick stand) is always the vented cap.



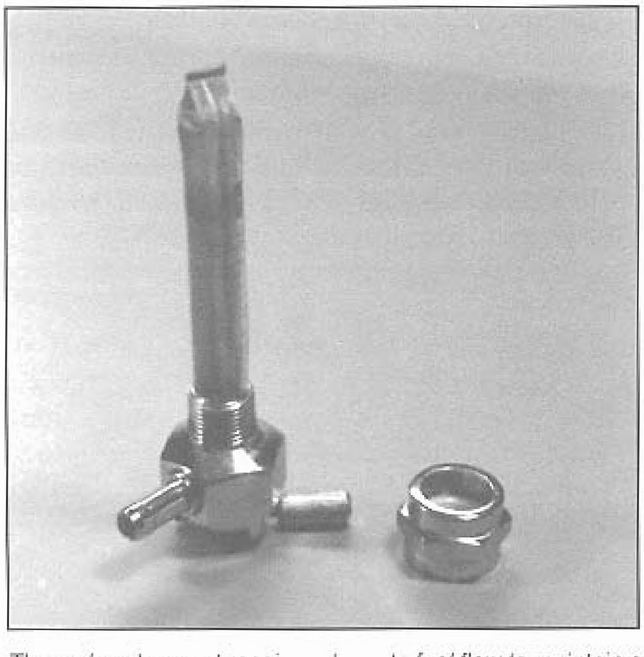
The fuel tank must be vented properly for adequate fuel flow. Notice the additional vent hole drilled near the top underside of the cam style gas cap. This modification is for race track use only.

Before modifying the cap, first check its venting capability. Wipe the cap dry and blow into it from its bottom side to determine how much air resistance the vent has. The FL gas cap is a cam design. Late model FX style gas caps are a screw-in design. The cam style cap is modified for race track use by drilling a .125-inch hole into its underside as shown in the photograph. On the other hand, when modifying the screw style cap, notice that the plastic boss in the middle of its underside includes four symmetrically located circles. This cap is modified for race track use by drilling a .120-inch hole (4 total) into each of the four small circles. Don't drill very deep or the cap will be unusable. When finished, clean the cap and again blow into it to compare the difference from stock.

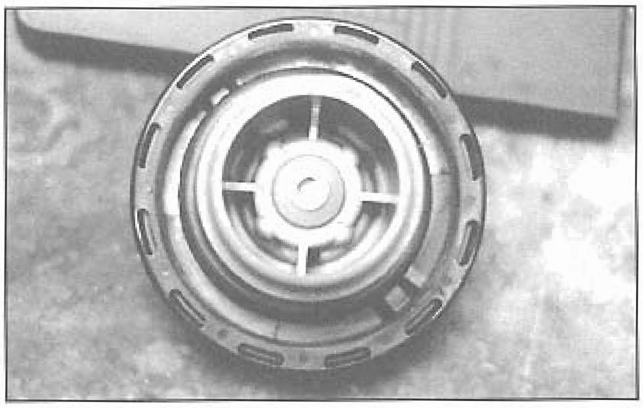
For race only bikes, a 90 degree 1/4-inch I.D. brass hose fitting can be installed into the top of the cap. Then one end of a vent line can be attached to the fitting and the other end secured to the front fork.

The stock fuel petcock (shutoff valve) provides acceptable fuel delivery for near stock engines, but it is marginal at best for modified engines and is not recommended for any highly modified engines. Although there are a number of aftermarket petcocks available, the Pingel shutoff valve is race proven, maximizes fuel flow and provides a customized appearance. Be sure to order the correct model and adapter nut.

Controlling the amount of debris that enters the carburetor through the fuel is critical to performance. If dirt passes into the carb, the chance of clogging jets or obstructing the proper operation of the needle and seat is high. It is a wise precaution to filter fuel as it is poured into the fuel tank. Use a fine stainless steel mesh



The carburetor must receive adequate fuel flow to maintain a proper air/fuel ratio. A high-flow fuel shutoff valve is recommended for any modified engine. Shown is the race proven Pingel shutoff valve. Be sure to order the correct model valve and adapter nut.



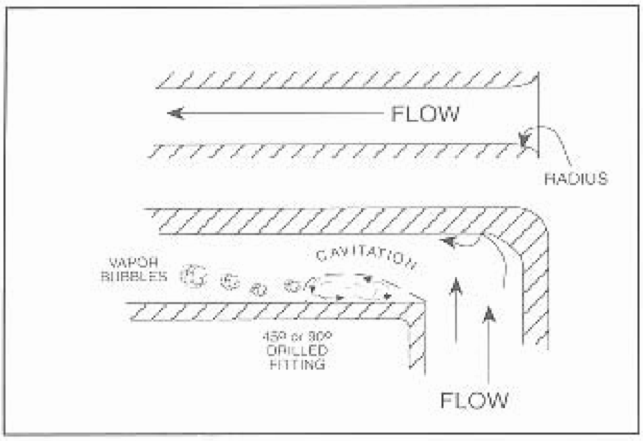
Notice the four circles that surround the centered boss on this screw style gas cap. Each circle area can be drilled for additional gas tank venting. This modification is for race track use only.

filter or a leather chamois, but do not use cloth since lint from its "clean side" can enter the system.

Engines run only on the race track can be run without an in-line filter as long as the fuel is stored in a clean container and it is filtered through fine stainless steel mesh as it is poured into the fuel tank. Street engines, however, should run a large-capacity in-line filter. Pingel offers a three inch long, 40 micron (25.4 microns equal one thousandth of an inch) filter, which is fine enough for good filtering, yet large enough in volume to hold a significant amount of debris without clogging. This filter uses a cleanable stainless steel mesh element that is about three times as fine as the one included on Pingel's fuel shutoff valve.

The fuel line should be routed so there are no kinks or sharp bends. Smooth curves impose the minimum restriction. Heated fuel is less dense and changes the optimum carburetor jetting. Keep the lines from touching the cylinders or heads. The plastic loom that covers the stock fuel line is a good insulator. Take note that a 5/16-inch I.D. line carries 57 percent more fuel volume than a 1/4-inch line and a 3/8-inch line carries 125 percent more volume.

If 45 or 90 degree fittings are needed for connecting pieces of fuel lines, use bent tube fittings instead of machined (drilled) fittings, especially if the fuel system is under pressure. Machined fittings have sharp corners and rough edges that can cause fuel cavitation. Ninety degree fittings are not recommended for use



The sharp corner of a machined fitting causes eddying flow and cavitation, which reduces flow. Use tube fittings instead of machine fittings, especially if the fuel system is under pressure and radius all inside edges. The same rules also apply to the engine's oil system.

anywhere in the fuel system. Also, any machining lips on the entry or exit of any fitting should be radiused for maximum flow.

CARBURETOR MODIFICATION TIPS

Many riders use carburetors other than Mikuni or S&S. Therefore, depending on your carburetor's design (butterfly or slide type), the following modifications may be helpful for improved airflow and fuel management.

- Choke Mechanism
  When a butterfly style choke assembly resides in a carb's throat, it creates turbulence and reduces airflow. Removing the butterfly and shaft and plugging the shaft's two holes in the carb's body will enhance airflow. You may need to add a fuel "tickler" or enrichment device for cold weather starting. A tickler is a small hand controlled plunger the mounts to a hole drilled through the fuel bowl. When depressed, it contacts the float, which opens the needle and seat and allows fuel to enter the carb's throat through the emulsion tube.

- Venturi
  Some carburetors have enough metal so its venturi can be bored larger or its entryway tapered bored to a different angle for increased airflow. If you bore a butterfly style carb, it will require a larger diameter throttle plate. All things being equal, flow increases with the square of the diameter.

- Throat Area
  For carburetors with a cast body, remove any casting flash in the venturi and throat area using a Dremel tool or die grinder.

- Radius Entryway
  The carb's entry should be radiused to direct the turbulent incoming air into the venturi in a laminar fashion for increased airflow. A velocity stack not only smoothes the flow into the carb's throat, but it also changes the intake tract's tuned length.

- Butterfly Shaft
  To increase airflow, the butterfly shaft can be thinned down in the area of the carb's throat and the throttle plate's mounting screws can be ground smaller and counter sunk. Be sure you install the screws with Loctite and stake lock them. The 1990 and later stock Keihin CV carb offers a good example of a streamlined shaft. Have patience because this modification takes time to do correctly.

- CV Piston
  The vacuum operated piston of a CV carb can be modified for increased airflow by gently radiusing the piston's front bottom edge.

- Fuel Bowl Volume
  The fuel bowl of some carburetors is too small to meet fuel demands under high acceleration conditions. Install either a larger fuel bowl or a spacer to increase the bowl's capacity.

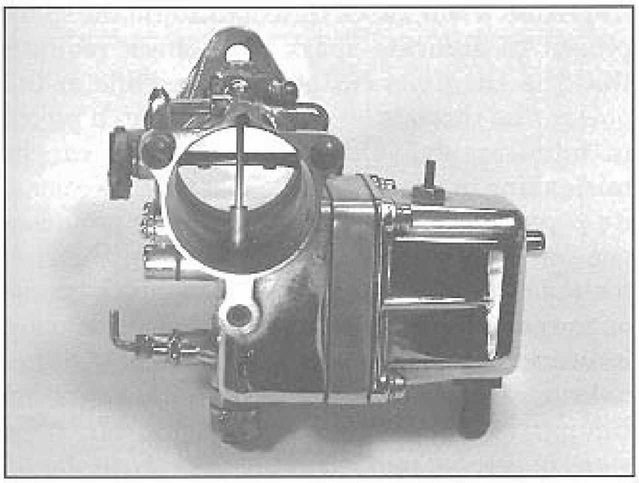
- Needle and Seat
  Install a larger diameter needle and seat assembly to fill the fuel bowl quicker. This will keep the bowl from running dry under hard acceleration. Note that Viton tipped needles offer the best sealing because they have a high tolerance for dirt.

- Fuel Level
  Install a clear plastic fuel level sight plug in the side of the fuel bowl for quick fuel level verification. A clear hose connected to fittings mounted on both the top and bottom of the fuel bowl will accomplish the same thing.

- Air/Fuel Orifices
  The quantity, size and position of the idle holes in the carb's throat can be changed to help improve throttle response. Likewise, the number, size and position of the emulsion tube's air holes will change fuel metering. Increasing the number of holes in the emulsion tube leans the mixture. Holes placed at the top of the tube affect mixture at the lower rpm range, while holes placed at the tube's bottom affect mixture at high rpm. Additionally, an adjustable main air bleed can be added for more flexible tuning.

- Butterfly Air Hole
  Rough idle problems and low speed flat spots sometimes can be helped by drilling a .060 to .125-inch hole through the butterfly plate. Drill the hole down from the top of the butterfly a distance equal to one quarter of its diameter. The hole allows additional air and fuel to enter the engine while allowing the throttle butterfly to remain more nearly closed. Start with a small hole and work your way up. The hole can be soldered closed if not needed.

- Spacer Block
  Add a one or two inch spacer block between the carburetor and manifold. This increases the plenum area, besides increasing the intake runner's tuned length. In general, a longer runner length increases low-rpm torque by moving the point of maximum torque to a lower rpm. Conversely, a shorter runner increases high rpm torque by moving the torque peak to a higher rpm.



Notice the thinned butterfly shaft and throttle plate screws on this bored out Linkert race carb.

TUNING CONSIDERATIONS

Preparation

Tuning a carburetor requires patience, knowledge of the carburetor's tuning circuits and an understanding of how the carb responds to adjustments. Up to this point, tuning procedures for the Mikuni HS40 and S&S Super carburetors were discussed. What follows is a more global approach to carburetor tuning.

Carburetion is only one of a number of major components that need tuning for the engine to achieve maximum performance. The air/fuel ratio must be adjusted before the ignition, gearing and exhaust system can be tuned properly. This is because the engine must be able to accelerate smoothly without missing or cutting out and to ensure the mixture is not excessively lean, which could damage the engine. Also, some tuning elements, such as ignition timing, need the carburetor correctly tuned because different air/fuel ratios burn at different speeds during combustion. As a result, the optimum ignition advance for a slightly rich mixture is different than the optimum advance for a slightly lean mixture.

The precision of carburetion tuning is directly affected by the mechanical condition of the engine. This means the engine must be a good sealing pump without leaking valves or rings. A weak ignition system causes a rough idle and will act as a governor, thereby limiting the engine's rpm. Check that the ignition timing is set to a reasonable baseline position. A high output coil and top quality spark plug wires can significantly help throttle response and high rpm power.

The ignition's mechanical advance mechanism, if you're running one, must work smoothly and cannot have weak springs. One symptom of a worn advance mechanism is that the engine will not immediately drop back to a normal idle rpm (about 1000 rpm). Instead, it will linger around 1600 to 2000 rpm and then suddenly drop to idle when the mechanical advance mechanism falls from the advance position. Some carburetors do not work well with the Evolution's stock electronic ignition and flat spots may persist because of the advance curve.

When running hydraulic lifters, make sure they are adjusted properly. Solid lifters should have the correct valve lash clearance. Also, cam timing that is too radical for the engine combination can make carburetor tuning difficult.

Straight pipes are easily tuned for the race track because they only need to run at high rpm. However, trying to tune them over a broad range, such as 2000 to 6500 rpm, can be difficult. Also, very large diameter exhaust headers can make carburetor tuning difficult. Whenever a free flowing exhaust system is installed, be sure to enrich the jetting.

Adding a deflector shield to a velocity stack's entry can be helpful when high velocity side winds and turbulent wind buffeting are present. The deflector will increase the intake tract's air pressure when at high speed. The float bowl should be vented to the same pressure as the intake tract. In this case, richer jetting is normally required.

Making a Run

Regardless of whether you are tuning a carburetor on a stock street engine or a highly modified big inch stroker, the tuning procedures are the same. Start by making sure the carb's jets, adjustable settings and float level are at a reasonable baseline for the engine combination. Also, check that there are no fuel leaks, the throttle linkage works smoothly and the throttle butterfly or slide fully open and close. The gas tank must be properly vented and the fuel line must be free flowing. Now the fine tuning can begin and it is up to you to get the most out of the combination. Remember, the performance difference between one engine and any similar engine is all in the tuning.

Make sure the engine is thoroughly warmed up. Tuning a carburetor always starts by first adjusting the idle mixture, next the low speed or pilot system and then progressing through each jetting circuit until the high speed circuit is complete. A good method for jetting is to determine the jet that causes distinct signs of richness, then lean down slightly. Review the procedures outlined under the Mikuni HS40 and S&S Super carburetors for reference. These procedures are good examples for tuning either a slide or butterfly style carburetor.

Always pay attention to what the engine is doing. The engine should accelerate smoothly, yet hard with a strong top-end. Think about where on the track or at what throttle position or rpm a problem occurs. Does the engine break up, blubber, misfire, or stumble and where does it take place? Does the exhaust sound crisp or muffled? The exhaust should sound crisp, but too crisp of exhaust note may indicate a lean condition. A rich mixture usually sounds muffled or blubbery and makes the engine accelerate sluggish and lazy. Look for a good transition from the low speed to the high speed circuits. An ignition caused misfire will usually be steady; however, a misfire caused by carburetion will usually be sudden and intermittent and usually affects only part of the rpm range. For example, a lean main jet will usually cause the engine to pop or "break up" at the top of each gear.

Remember that the idle and low speed circuits respond to manifold vacuum and the high speed circuits respond to airflow. Problems encountered in the low or midrange may require a larger air bleed jet to move the effective range of the high speed jetting circuit. Pilot jets, intermediate jets or the jet needle's position may require a change. In general, an engine gives the quickest acceleration when the low and midrange air/fuel ratio is slightly rich. Don't forget the accelerator pump's delivery may need to be increased or decreased and it's sometimes best to tune with the pump turned off. Installing a high flowing air cleaner or a velocity stack in place of an air cleaner normally requires enriched jetting.

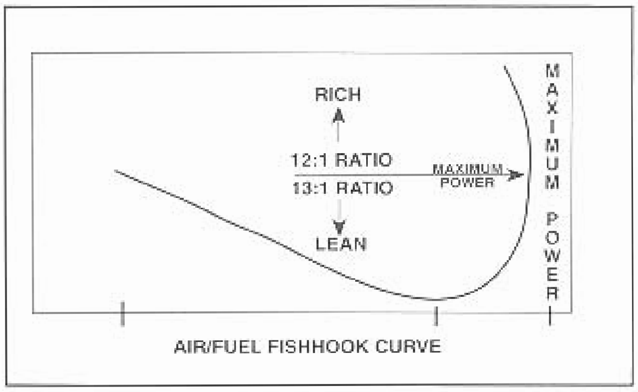
Key Indicators

MPH is one of the key indicators for achieving maximum power and optimum high speed jetting. After a run, check the condition of the spark plugs. An accurate spark plug check requires that the engine is cut off cleanly while at full power, the throttle closed and the clutch pulled in immediately. A spark plug's color can be misleading in a quarter mile test and the color of its porcelain and electrodes can only provide a general idea of the air/fuel mixture. Signs of melted cement, cracked porcelain, black specks on the porcelain's nose and melted electrodes are indications of detonation or extreme heat due to a lean air/fuel mixture or improper ignition timing. The plug's porcelain should not show signs of glazing and the center electrode should not be turning blue. These conditions, along with a ground electrode turning green are signs of too much heat. Black specs or small shiny black and purple colored balls attached to the porcelain are indications of detonation. If your plugs show any of these signs, add fuel or back off the timing. Be sure to correct the problem before making any additional runs.

Some racers look at the top of the piston dome or check the inside of the exhaust port for color. Use a small inspection light to view the piston dome through the spark plug hole. A backup check for determining a rich or lean condition is to check the underside of the piston domes when the engine is torn apart. The dome's underside will be light brown when the air/fuel mixture is about right. If the dome shows no color, its cylinder is running cold and too rich. When dark brown, the cylinder is too lean and you need to correct it.

Unless the plugs indicate a rich mixture, continue adding fuel to the high speed circuit until mph drops off. Then back down one jet size. If you enrich the high speed and nothing happens, keep enriching it repeatedly until the mph drops. Sometimes one jet change will not reflect any difference in mph.

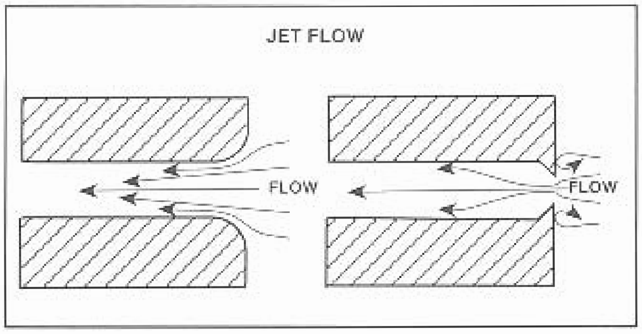
MPH will also drop when the engine gets too lean. However, power drops quicker when the mixture becomes lean than when it becomes rich. This is referred to as the air/fuel fishhook curve. For this reason and because of potential engine damage, you are better off slightly rich than lean. Remember that a lean mixture is more detonation prone than a rich mixture.



The chemically correct air/fuel ratio is about 15:1. However, maximum power is achieved between 12:1 and 13:1. Notice that power drops more quickly as the mixture becomes lean. However, an engine is never tuned for a specific air/fuel ratio, only for maximum power. Initially tune the carburetion for obvious signs of richness, then slightly lean the mixture.

Jet Considerations

A jet must be accurately machined and its orifice finished reamed for precise fuel flow. Also, the radius leading to the jet's orifice must be constant from jet to jet for accurate tuning. A jet with a radiused orifice will flow up to 20 percent greater than a jet with a straight through hole. This is because the radius smoothes the flow, which makes it easier for the fuel molecules to move through the hole.



A radiused orifice has up to a 20 percent larger flow area than one with a sharp edge. You can be fooled during carburetion tuning when using jets with different shapes of orifices.

Some jets are size rated while others are flow rated. The number stamped on a *size rated* jet indicates the diameter of its hole. Doubling the hole size of a size rated jet quadruples its flow volume. This is because flow volume is proportional to the hole's diameter squared. This means a small increase in jet size gives a large increase in fuel flow. For size rated jets, use a calculator or an orifice area conversion chart to determine the amount of jet change (jet flow area) required for a given air density reading.

The number stamped on a *flow rated* jet usually indicates the jet's flow rate in cubic centimeters per minute. This means the jet's flow rate is proportional, on a percentage basis, to the number stamped on it. Jet changes are easier to calculate with flow rated jets.

If the engine does not respond well to jetting changes, chances are that some other tuning variable is radically out of bounds or something is severely wrong with the engine combination. Factors beyond carburetion, such as poor combustion chamber design or low compression may limit the degree to which the engine's carburetion may be tuned.

Fuel Considerations

When tuning, always use the same brand of fuel and octane rating that you normally use. If using racing fuel, choose an octane rating that suppresses detonation for your compression ratio and engine combination. As the octane rating increases, the fuel's resistance to detonation will also increase. But keep in mind that excess octane is not helpful and remember that detonation is the major cause of piston, rod and bearing failures. Also, different racing fuels may require a different ignition advance setting for optimum power because different fuels often burn at different speeds.

Fuels vary in density and composition and the variations can have a major affect on tuning. An engine performs best at only one air/fuel ratio and the ratio is based on the *weight* of the air and fuel. Since carburetors meter fuel by *volume*, a jet change must be made when the specific gravity of the fuel changes to keep the *weight* of the fuel being mixed with air the same. The denser the fuel is, the smaller the jet you will need for a given set of conditions.

Changing from one brand of fuel to another may require jetting changes for maximum power. Even two batches of the same brand and grade of fuel can change in density and require jetting changes. The specific gravity rating of racing fuels usually ranges between .66 and .74 while aviation gasoline is normally .68 to .72. Specific gravity indicates the *weight* of fuel on a percentage basis when it is compared to water. When changing brands of fuel, ask the manufacturer for their fuel's rating so you can compare one fuel to another. As ambient temperature increases, the specific gravity of fuel decreases. All fuels are rated at 60 degrees Fahrenheit. In close, heads-up racing, measuring the specific gravity of fuel at the track can be a big help.

Specific gravity test kits include a hydrometer bulb and are available from Kinsler Fuel Injection and automotive performance shops. As a low cost alternative, a battery hydrometer or antifreeze tester can be used to test fuel for specific gravity.

Weather Effects

Once the engine's carburetion is dialed in for a given altitude, temperature and humidity, jet changes can be calculated for different altitudes and ambient weather conditions against the original baseline jetting. If possible, use an air density

Use an air density gauge to correlate your baseline jetting to a given atmospheric condition. The gauge simplifies jetting changes required for different altitudes and temperatures. Be sure to always use the same gauge, since there is no guarantee every gauge is calibrated identical.

gauge to determine high speed jetting changes from the baseline value and always use the same gauge because different gauges may be calibrated differently.

For every 1000 foot increase in altitude, engine power drops about three percent due to thinner air. Also, for each 11 degree rise in temperature, power drops one percent for the same reason. These values translate into about a 1.0 percent change in E.T. and mph for each 1000 foot change in altitude, and a 0.3 percent change in E.T. and mph for each 10 degree change in temperature.

CARBURETOR TUNING GUIDE

CONDITION	EFFECT ON MIXTURE	CHANGE REQUIRED
Warmer Air	Richer	Leaner
Colder Air	Leaner	Richer
Dry Air	Leaner	Richer
Lower Altitude	Leaner	Richer
Higher Altitude	Richer	Leaner

Table 4.6

When tuning, only make one change at a time because multiple changes introduce too many variables and confuse the tuning process. Also, record all tuning changes made, including any change in performance and the current whether conditions. This information can be referenced during future tuning sessions and it should be an invaluable time saver.

For every run down the track, have an objective to accomplish and concentrate on that objective before, during and after the run.

INTAKE TRACT TUNING

A cylinder's volumetric efficiency can be increased by taking advantage of the acoustic energy pulses within the intake tract. As the intake valve closes, a pulse is sent through the intake tract. As the pulse hits the atmosphere at the open end of the intake tract, the pulse bounces back toward the intake valve. By adjusting the length of the intake tract so the return pulse arrives at the valve when it opens, cylinder fill can be increased.

The tract length is measured from the backside of a closed intake valve to the air inlet entrance. This point is either where the air cleaner bolts to the carb or the end of a radius ring inside the air cleaner. It also can be the radiused end of a velocity stack. On the drag strip, a Big Twin engine is normally turning between 4500 and 7000 rpm. However, depending on the engine combination and gearing, the rpm may be as low as 4000 or as high as 8000. The tuned intake length would normally be set for the middle of the rpm band. For an engine turning 4500 to 6500 rpm, this would be 5500 rpm.

The intake tract can be tuned to several different pulses, but each time the pulse changes direction it loses energy. This means the strongest usable pulse is the second, but it also requires the longest tract length and this length may be impractical to use. Also, the calculated tuned length should be considered only as a starting point. Testing at the race track will determine the best length.

Each of the three following formulas is designed to take advantage of a different intake tract pulse Use the one that gives the most practical intake length for your bike. The constants used in the formulas consider the speed of sound, intake tract temperature and the amount of time the intake valve is open.

2nd pulse:

$$= \frac{1100 \times \frac{1}{2} \textit{Intake cam duration} \times .96}{\textit{rpm}}$$

3rd pulse:

$$= \frac{1100 \times \frac{1}{2} \textit{Intake cam duration} \times .705}{\textit{rpm}}$$

4th pulse:

$$= \frac{1100 \times \frac{1}{2} \textit{Intake cam duration} \times .538}{\textit{rpm}}$$

Example:
Optimize an Evolution engine's intake tract length for 5500 rpm and 250 degrees intake valve duration. Use the third pulse for the calculation.

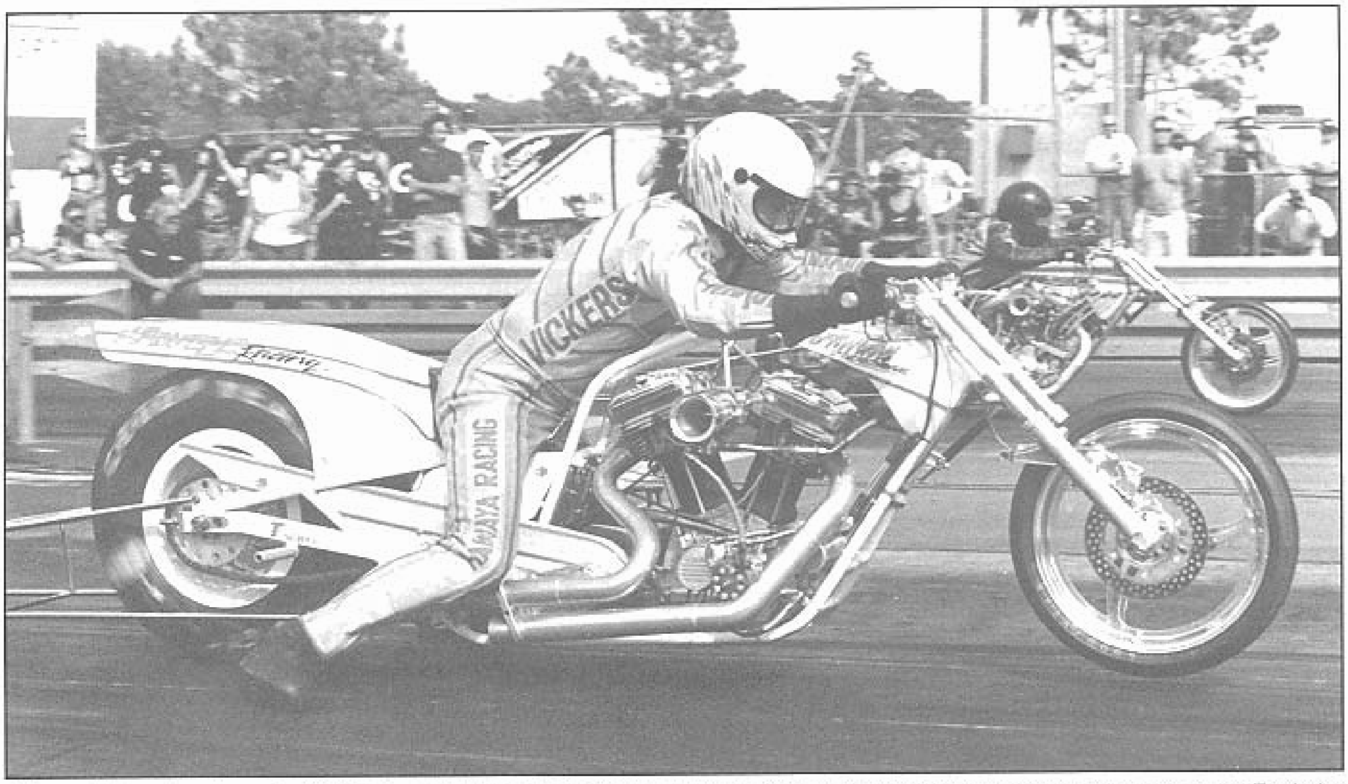
$$= \frac{1100 \times 125 \times .705}{5500}$$

$$= \frac{96937.5}{5500}$$

= 17.625" tuned intake tract length

The intake tract length of a stock 1990 and later Evolution engine is about 9.5-inches and it's about the same with an S&S Super E or G carb. Adding a 4-inch velocity stack increases the tract's length to about 13.5-inches. The S&S B and D model carbs have a long body, which increases the length about 1.5-inches. With a Super B or D carb, a 4-inch velocity stack and a 1-inch spacer block, the intake length calculates to 16-inches. This is still 1.625-inch short of the calculated tuned intake length. Adding a second 1-inch spacer will help. Another option is to calculate length using the 4th pulse. This results in a shorter 13.45-inch length, which is easier to achieve. ❖

S&S Super B carburetor on Chris Barbarrie's beautifully detailed FX. Notice the adjustable air bleed jet on the side of the carb. Chris runs in an ECRA Eliminator class on an 11.30 E.T. index. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.



Johnny Vickers has turned a 7.26 E.T. on the Pro Fuel, high gear only, fuel injected, Hawaya Racing drag bike. This 140 ci Evolution Big Twin includes a sophisticated racing fuel injection system often used in place of carburetion on nitro burning engines and 4-cam crankcases. The 4-cam cases create a hybrid engine design that combines a Sportster style cam arrangement with Big Twin cylinder heads. Four-cam crankcases are a current trend in high dollar motors because they offer superior valvetrain geometry and cam degreeing capability over the traditional single cam Big Twin crankcase. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

Chapter 5

Cam and Valvetrain

Got To Have Timing

The camshaft critically influences the amount of air and fuel the engine breathes.

In fact, it is considered the brain and personality of the four-cycle engine because it controls the timing and duration of all major events. As a result, the camshaft is the primary component for tuning the engine's torque and horsepower curves. It also directly affects the engine's induction and exhaust systems, besides the optimum compression ratio.

Due to emission requirements, the stock Big Twin has less than optimum cam timing and this severely restricts its breathing capability in the higher rpm ranges. Replacing the camshaft should be one of the first four performance upgrades for a Big Twin engine — the others being the carburetor, exhaust and ignition. A performance cam will add a few horsepower down low in the rpm band and 10 or more horsepower in the 6000 rpm range to a relatively stock non-California Big Twin engine. It will add even more power when the engine is highly modified. However, it must be matched to the engine combination for maximum power.

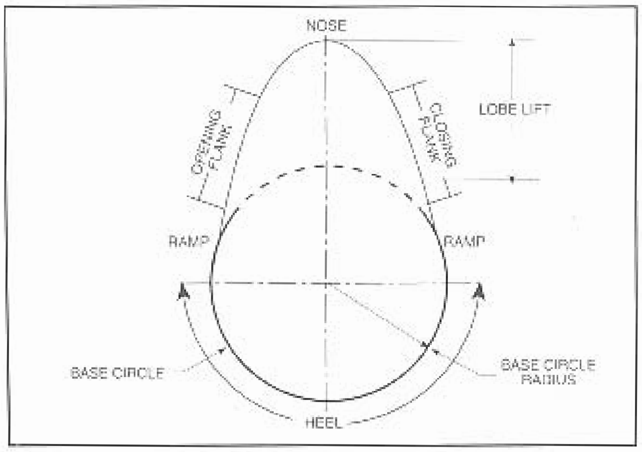
There is no magic involved in choosing the best cam for an engine combination. Although there are numerous cam grinds that would make a reasonably good match for any given engine, the only way to know for sure which cam is best is to test each one. Since cam design is not a precise science, it is impossible to guarantee without prior testing that one cam is best for a given engine combination because too many variables are involved.

For maximum performance, the cam must be matched not only to the engine's displacement and rpm band, but also to its head's airflow characteristics, bore-to-stroke relationship, induction system, exhaust system and compression ratio. Even the engine's rod-to-stroke ratio plays a part. Choosing a functional cam for a given engine combination is not too difficult; however, identifying the optimum cam requires some knowledge. Because there are many variables involved, winning racers spend a great amount of time testing custom ground camshafts to determine what works and what doesn't with their engine combination.

Lift, duration and the timing of events are the three major considerations used in camshaft design. The size, shape and position of the cam lobes determine how high, how long and when the valves open and close. Selecting the best combination of lift, duration and timing amounts to a compromise between low-speed torque and high speed horsepower. Yet many Big Twin owners buy a cam without understanding the factors involved and this results in disappointing engine performance. Major factors and elements you should understand when selecting and installing a cam are covered below.

LIFT

Airflow through the cylinder head is directly controlled by the height of the valve's lift. Up to a point, the higher the valve lifts, the greater the flow. At low lift, when the valve is only slightly off its seat, airflow is restricted between the valve and its seat. When the valve is raised, the airstream forms a cone-shaped path around the circumference of the valve's head. If the shape of the cone is disturbed by any obstruction, such as the shape of the combustion chamber, airflow through the valve's opening will drop.



Cam lobe lift is measured from the base circle radius to the high point of the lobe. This measurement is equal to lifter lift, not valve lift. To calculate valve lift, multiply lobe lift by the rocker arm ratio.

The amount of flow is controlled not only by how high the valve is lifted, but also by how quickly it lifts. Ideally a valve would lift instantaneously to maximum lift, but this is impossible. The quicker a valve is lifted, the greater amount of time it spends at higher lifts. In cam terminology, this increases the area under the valve lift curve and results in increased flow past the valve. The negative effect of lifting the valve quickly is that the valve requires higher valve spring pressure to ensure the valve follows the profile of the cam lobe. High spring pressures increase the wear of the valve, seat and guide, therefore a compromise is made between the speed at which the valve lifts and the amount of tolerable wear.

In general terms, higher valve lift helps generate torque and horsepower, but the speed at which the valve is raised to full lift is also a factor. Increasing lift is a good way to increase power without significantly reducing low speed performance. But keep in mind that some cylinder heads stop flowing more air as lift is continually increased. In this situation, the heads respond more to duration than lift. From a reliability standpoint, there needs to be some relationship between lift and duration since only increasing lift without an accompanying increase in duration can produce high valve acceleration rates, which can increase valvetrain wear and reduce reliability. Cams with very high lift and relatively mild duration are typically used to maximize low and midrange power or used with low compression engines.

The optimum valve lift is generally determined by the cylinder head design. This is one reason a cam should be matched not only to the displacement of the engine and its application, but also to the cylinder head. When a valve lifts 25 percent of its diameter, the area of its orifice is equal to the area of its diameter. For this reason, the ideal valve lift is usually some ratio of the valve's diameter. Stock engines usually have a maximum lift of about .25 of the intake valve diameter while racing engines are usually about .30 to .35 of valve diameter. Up to about .15 of valve diameter, airflow is primarily controlled by the valve and seat area, but at higher lifts it is usually limited by the capacity of the port and sometimes by the shape of the valve or combustion chamber.

The stock Shovelhead valve lift is .390-inch, while for the Evolution it is .495-inch for 1988-1991 year models (L-cam) and .472-inch for all other years. For the Evolution, this results in a lift slightly greater than .25 of its intake valve diameter. On the other hand, the Shovelhead lift is only about .20 of its intake valve diameter.

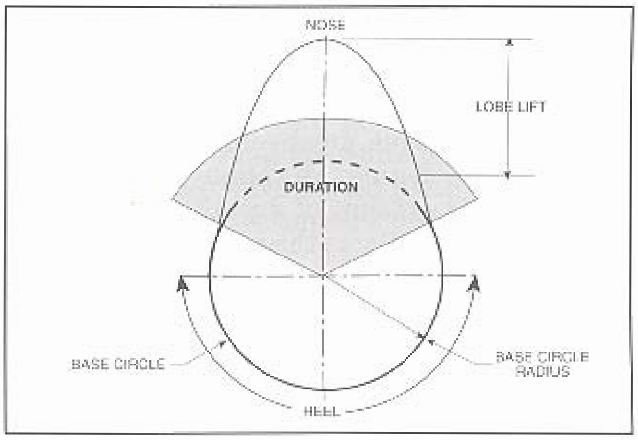
Valve lift is calculated by multiplying cam lift by the engine's rocker arm ratio. One method for determining optimum valve lift is to flow test the port at different valve lifts. This will indicate at what lift flow stops increasing or even starts decreasing. Some cams are designed to lift the valve beyond the point of maximum flow and some lift even greater than .35 of the valve's diameter. Lifting the valve quicker places more area under the cam's lift curve. But lifting the valve quicker sometimes requires that it is lifted beyond the point of maximum flow. This results in gaining flow at high lift because the valve dwells longer at maximum lift. Flow at lower lifts is also gained because the point for a given lift is reached earlier in the valve's cycle.

Due to its cylinder head design, it is much easier to install a high lift cam in an Evolution engine than a Shovelhead. As a result, it is common for an Evolution cam to have .050 to .100-inch more lift than a comparable Shovelhead cam. Where a typical high lift Shovelhead cam raises the valve between .475 and .575-inches, a comparable Evolution cam lifts between .530 and .640-inches with some lifting higher than .750-inch. For a bolt-in cam, maximum lift is limited to about .500-inch for an Evolution engine and about .450-inch lift for a Shovelhead, depending on the year of the engine and profile of the cam.

The term "lobe lift" sometimes is mentioned when discussing cams. This term refers to the lift of the cam's lobe and in most cases it is different then valve lift. Valve lift divided by the rocker arm ratio equals lobe lift. For example, dividing the Evolution's .472-inch valve lift by its 1.6:1 rocker arm ratio results in a .295-inch lobe lift. Take note that the Shovelhead's rocker arm ratio is 1.5:1. Also, the Knucklehead engine has a rocker arm ratio of 1:1, so its valve lift and lobe lift are identical.

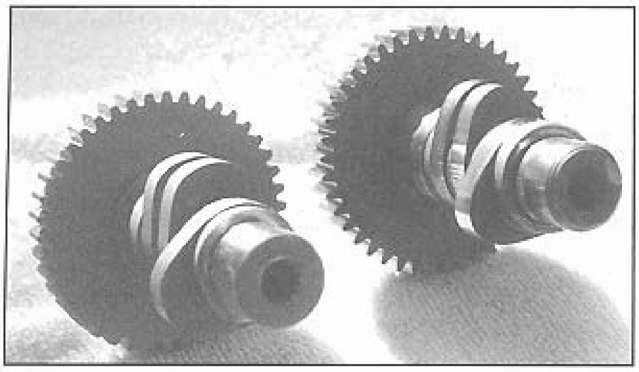
DURATION

Duration is the amount of time computed in crankshaft degrees the cam holds the valve off its seat. The greater the duration is, the longer the valve is held open. Up to a point, the greater the cam's duration, the greater the engine's breathing ability will be. However, increased power is not guaranteed by additional duration unless the induction and exhaust systems can make use of the duration and move more air/fuel mixture through the engine. This requires fine tuning the induction and exhaust systems to a particular cam for maximum results.

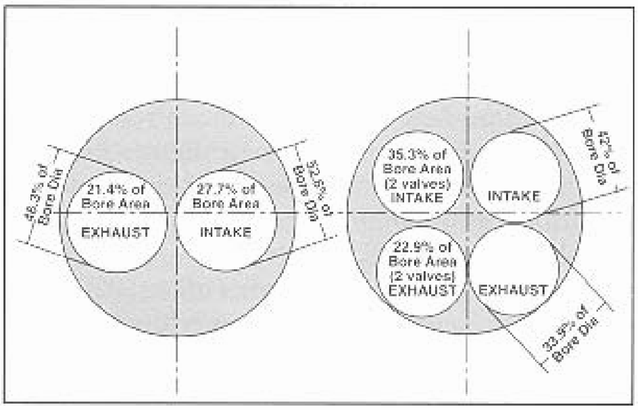


The valve is closed when the lifter is traveling along the cam's base circle surface. The cam's ramps start the valve's opening and finish its closing. The time from which the valve leaves its seat to the time it again is seated results in the cam lobe's duration. Duration is normally measured at a point where the lifter is not positioned on the lobe's ramp.

In general, duration is a function of engine displacement and rpm. As engine displacement is increased, cam duration usually can be increased. For a given engine displacement, increasing duration moves peak power to a higher rpm range, resulting in decreased low-rpm power and torque. What is gained on one end is lost on the other. This is because more duration increases cylinder filling at high rpm. Conversely, the longer the valves remain off their seats, the less time there is to maintain cylinder pressure at low rpm and low-end power is generally reduced. Therefore, duration can be used to determine where in the rpm band the engine will make peak power.



Notice the difference in lobe profile between the stock Evolution L cam (right) and an Andrews EV3.



The cylinder bore diameter limits the size of valves more in a two-valve head than a four-valve design. Since four-valve heads expose flow area quicker than two-valve designs, less valve duration is required. The two valves in a stock Evo head are equal to 49.1 percent of the stock bore area; 27.7 percent for the intake valve area and 21.4 for exhaust. For unmodified four-valve heads, the total valve area is 58.2 percent; 35.3 percent for the intake valve area and 22.9 for exhaust. Four-valve heads generally perform best in large bore, very high rpm engines.

Depending on the year of the engine and the state it is sold in, the Evolution's intake duration varies between 208 and 222 degrees and its exhaust duration between 202 and 236 degrees. Engines sold in California have about 178 degrees duration (C-cam) on both the intake and exhaust. Stock Shovelhead duration is 220 degrees for the front cylinder intake, 232 degrees for the rear cylinder intake and 244 degrees for both exhausts.

As a general rule, any Evolution cam with up to about 240 degrees duration is considered having a mild amount of duration, 241 to 265 degrees duration places the cam into the moderate class and one with more than 265 degrees duration falls into the "maximum effort" category. A Shovelhead engine has lower velocity ports than an Evolution, so add about 10 degrees duration to each cam category for equivalent values.

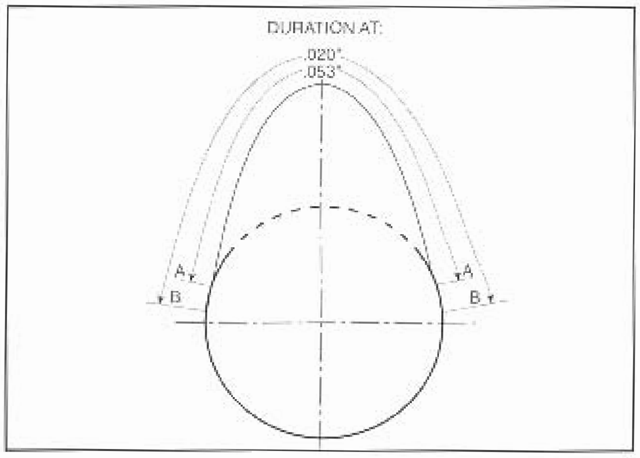
Cam duration for four-valve heads is an exception to the above rules. Since four-valve heads expose flow area quicker than two-valve designs, less duration is required. This is because flow through a valve's opening is a function of the valve's circumference and two small valves have greater circumference than one large one. Using a cam designed for two-valve heads in a four-valve head engine produces a narrow high rpm power band with little power down low.

Long duration means the valves are off their seats for a longer time, which can result in lower cylinder pressure and reduced power at low speeds. This is especially true if the cam has a large amount of overlap. Raising the engine's mechanical compression ratio in conjunction with increasing duration will help maintain cylinder pressure at low rpm and minimize potential power losses.

The de facto measuring standard for Big Twin cam duration is at .053-inch lifter lift (not valve lift) off the cam's base circle. This means that duration is measured in crankshaft degrees (not cam degrees) from the point where the *lifter* rises .053-inch to the point where it drops back down to .053-inch lift. Some manufacturers measure duration at .020-inch lifter lift. The reason .053-inch is used as the measuring point is because it offers more accurate and consistent readings than measuring at a lower lift.

To reduce shock to the valvetrain, cam lobes are ground with extremely gradual opening and closing clearance "ramps" that remove the play in the valvetrain before valve opening. This causes the lifter to move very slowly when on the ramps and makes it very difficult to measure accurately the exact point in crankshaft rotation where the lifter moves. Although some opening and closing ramps are longer than others, checking at .053-inch moves the lifter past the slow moving ramp and on to the faster moving flank of the cam for more consistent and accurate readings.

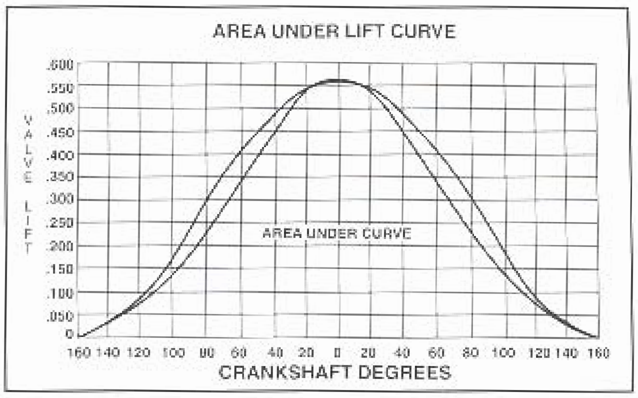
Tom Sifton started the de facto Harley cam measuring standard of .053-inch by adding .013-inch lift, which was generated by the slow moving ramp, to an arbitrary value of .040-inch (approximately one millimeter). As a comparison, the de facto measuring standard for automotive cams is .050-inch lifter lift.



The duration of most Harley cams is measured at .053-inch lifter lift. Duration will always be greater when measured at a lower lift such as .020-inch. This is because lift generated by the lobe's opening and closing ramps is included. When comparing the duration of two different cams, be sure to use figures calculated at the same lifter lift.

Readings taken at less than .053-inch lifter lift will reflect a greater amount of duration and an earlier opening and a later closing point. Therefore, for accuracy you must always compare cam specifications measured at the same lifter lift. The only accurate way to compare two cams measured at different lifter lifts is to remeasure one at the other's standard. However, if you want to compare the duration of a cam measured at .053-inch to one measured at .020-inch, add 40 degrees duration to the one measured at .053-inch. This will give an approximate comparison.

Even measuring two different cams at the same lifter lift does not tell everything about the cams. For example, it is possible that both cams open and close the valve at the same crankshaft timing, however, one cam may have steeper flanks and a broader nose, thereby lifting the valve quicker and keeping it open at maximum lift longer than the other. This means the opening and closing rates are another critical design consideration for the cam manufacturer. There is no way to determine the cam's opening and closing rates by looking at the advertised specifications, although in some cases the amount of TDC lift may give a general indication of lift rates. The only sure way to know how two cams compare is to graph their lobe profiles at 20 degree intervals and then compare the results. This will show how much area each cam has under the lift curve. In general, more powerful cams have more area under the lift curve.

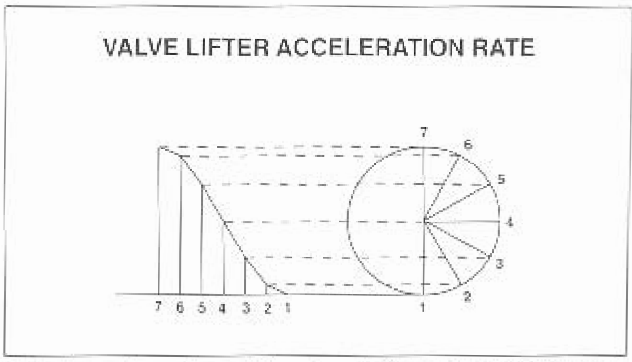


The only valid way to compare two cam profiles is to measure their lift at different degrees of flywheel rotation and plot the results on a graph. These two cams have the same duration and lift at .053", but one has more area under the curve because it lifts quicker.

Opening and closing rates are dictated not only by the amount of tolerable valvetrain wear, but also by the rod-to-stroke ratio. Engines with a high rod-to-stroke ratio typically have a lazier piston acceleration rate away from TDC than those with a lower ratio. The speed at which the piston moves away from TDC is an important factor in determining the cam's profile. In most cases, the intake valve should reach at least half of its total lift by the time the piston reaches its maximum acceleration rate away from TDC (about 74 to 76 degrees ATDC for the Big Twin). It is at this point during the intake cycle that the strongest signal is delivered to the intake port.

SINGLE AND DUAL-PATTERN

A single-pattern cam design has an identical amount of duration and lift for the intake and exhaust valves. Conversely, a dual-pattern cam has a different amount of duration or lift for the



This illustration shows lifter rise relative to camshaft rotation. The angle of the curve illustrates how quickly the cam lifts the valve. The steeper the angle, the faster the lift rate. Notice that the fastest acceleration occurs during the middle of the lift curve.

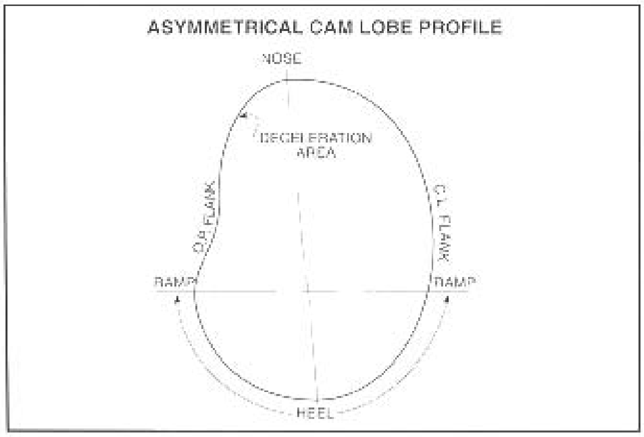
intake and exhaust sides. Some companies favor single-pattern cams while others favor dual-pattern designs.

A dual-pattern cam is sometimes favored by engine builders when there is unequal breathing between the intake and exhaust ports. On the other hand, single-pattern proponents believe that a poor flowing exhaust port can be adequately scavenged with a single-pattern design because the exhaust is purged by positive piston motion.

The flow differences between stock and ported heads should be taken into account when choosing a cam. Some stock heads, like the Evolution's have a poor flow ratio between the intake and exhaust ports. At .500-inch lift, its exhaust port flow is less than 80 percent of the intake. You will frequently see dual-pattern Evolution cams having greater exhaust duration than intake duration. With ported heads the exhaust port normally flows between 80 and 90 percent of the intake and sometimes as high as 95 percent. Then when the carb, intake manifold and exhaust header are installed, the induction and exhaust sides should flow similar amounts.

SYMMETRICAL AND ASYMMETRICAL PROFILES

A symmetrical cam has identical lobe profiles on the opening and closing sides (flanks). Conversely, an asymmetrical cam has a different lobe profile on each side. Big Twin cams usually



An asymmetrical cam lobe profile has a different shape for its opening and closing sides. The opening flank starts accelerating the lifter. The deceleration area slows the lifter down so it stays in contact with the lobe as it goes over the nose. The closing flank must close the valve without bouncing it on the seat.

are asymmetrical because they open the valve quickly for increased flow, but close it slowly to reduce noise and wear. This requires the lobe to have one shape for its valve opening side and another for the closing side. An asymmetrical lobe profile increases cylinder filling and engine torque, besides allowing a higher operating rpm.

The more radical the lobe's profile, the higher the valve spring pressure must be to keep the lifter following the cam lobe. If the shape of the lobe changes too quickly, inertia will cause the lifter to separate from the lobe at high rpm, causing inaccurate timing, valve float and possible engine destruction.

Increased valve spring pressure can lead to rapid valvetrain wear, especially at low rpm. High pressures generate greater internal friction, which leads to increased temperatures and reduced horsepower. However, it is critical to run sufficient valve spring pressure, but not excessive pressure. Different cams require different on-seat and maximum lift spring pressure. Check with your cam manufacturer for his recommendations. If you are in doubt about how much pressure to run, it is always better to run too much pressure than too little.

OVERLAP

Overlap is the time measured in crankshaft degrees that the intake and exhaust valves are open simultaneously. This is when the piston is near TDC and the intake valve is just starting to open and the exhaust valve is about to close. The amount of overlap and where it occurs will have a major impact on the engine's ability to make power. Some phenomena that occur during overlap are beneficial, while some are harmful.

Since valves cannot open and close instantly, the overlap period provides the time to open or close a valve gradually. If matched properly with the exhaust system, the overlap period can help evacuate exhaust gases when the piston is near TDC. At high rpm, the inertia of moving gases can be used to help draw fresh fuel mixture into the cylinder during overlap, which greatly increases volumetric efficiency. However, at low speeds the inertia effect is too low to be helpful and exhaust reversion (exhaust backflow into the combustion chamber) contaminates the fresh fuel charge and reduces power. Also, at low speeds exhaust pressure pulses can travel through the combustion chamber into the intake tract and upset carburetion. The benefits of the overlap period are very rpm-specific and are maximized when combined with a tuned exhaust.

As duration increases, overlap will also increase, all other things being equal. When the overlap period increases, the valves are off their seats longer, which means that a higher rpm is required to establish sufficient cylinder pressure. This is why a cam with a long overlap period performs better at higher rpm, but reduces power in the lower rpm range. Tuned straight pipes can benefit significantly from a long overlap period.

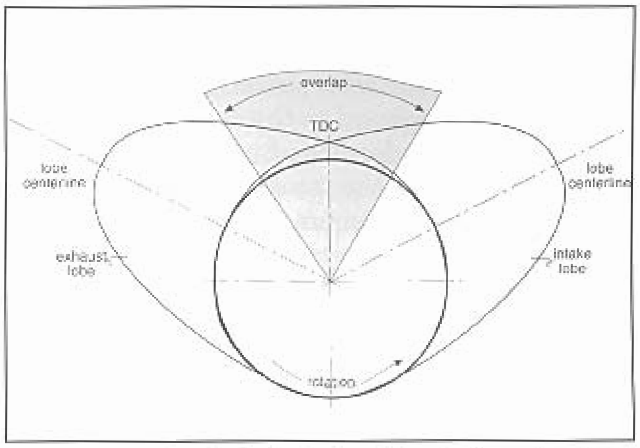
Besides increasing duration, overlap also can be increased by moving the cam's intake and exhaust lobes closer together. On the other hand, moving the lobes farther apart reduces the amount of overlap. The relationship between the intake and exhaust lobes for the same cylinder is referred to as lobe separation angle and it will be discussed later.

As overlap is increased, the engine's mechanical compression should also be increased to retain low rpm cylinder pressure and low-end torque. In general, the shorter the overlap period for a given engine, the lower the rpm at which the engine will develop maximum torque.

The amount of overlap can be determined by adding the intake valve opening timing (before TDC) to the exhaust valve closing timing (after TDC). For example, lets assume a cam has an intake opening of 31 degrees before TDC and an exhaust closing of 27 degrees after TDC. Its overlap period would be 58 degrees.

VALVE TIMING

Lift, duration and overlap are not the only critical cam design factors that affect an engine's power level. The timing of the valve events, which are the points at which the intake and exhaust valves open and close (in *crankshaft* degrees) are also extremely critical in determining not only the engine's power, but also its power curve. The performance differences between two cams can only be proven by cut-and-try testing on the track or dyno. However, the cam's opening and closing points and its duration can offer clues about its performance characteristics.



The position of the intake lobe in relation to the exhaust lobe determines valve overlap. Overlap is the time when the piston is near TDC and the exhaust valve is about to close and the intake valve is starting to open. The period when both valves are off their seats is valve overlap.

The optimum camshaft for a given engine is determined by a myriad of factors such as port airflow, displacement, stroke, rod length, rpm and other factors. But the bottom line is how much cylinder pressure the cam allows the engine to make at a given rpm. Changing the timing of the cam's opening and closing events can significantly change the power making characteristics of the engine. However, there is always a tradeoff between high rpm horsepower and low-speed torque.

Increasing duration moves the points at which peak torque and horsepower occur to a higher rpm level. As a result, low speed power suffers. Extended duration causes the valves to open earlier and close later. This increases cylinder filling on the intake side and helps scavenging of combustion gases on the exhaust side. An early opening intake valve can help cylinder filling at high rpm, but it also can waste fuel and reduce power if much of the mixture is sucked out of the combustion chamber and through the exhaust port during valve overlap. Additionally, a late closing exhaust valve can cause over scavenging by dumping fresh fuel mixture out the exhaust port.

Of the four opening and closing valve events, the *intake closing* is the most important followed in importance by the *exhaust opening*. These events are most critical because they determine the time the cylinder is sealed (both valves closed) and cylinder pressure can build on the piston.

The intake valve closing starts the change from the engine's intake cycle to compression cycle. At this point, the piston has passed BDC and is moving upward in the cylinder. Also, the intake charge has developed a significant amount of inertia. Holding the intake valve open as long as possible at high rpm makes maximum use of the charge's inertia and increases cylinder fill for additional cylinder pressure and more power. The inertia that generates the additional cylinder filling is called "ram effect." However, this phenomenon must be balanced against the moment when the upward piston movement creates enough cylinder pressure to push the fresh fuel charge back into the intake port. This reduces cylinder filling at low rpm and hurts torque and power

Although closing the valve late takes advantage of the intake charge's high velocity and built up inertia, it only works well at high rpm. At low speeds, the intake velocity is too low for ram effect. Therefore, the best time to close the intake valve varies according to engine rpm. The later the intake valve closes, the higher the engine must be revved to develop peak torque. On the other hand, closing the intake valve early increases cylinder pressure and reduces the amount of fuel charge forced back into the intake port at low engine speeds, thereby increasing low rpm torque.

Keep in mind that the later the intake valve is closed, the more the engine's mechanical compression ratio must be increased to offset cylinder pressure losses at low rpm. When comparing two cams with similar specifications, take note of the intake valve's closing point. All other things being equal, the later it closes, the higher in the rpm band peak torque and horsepower will be developed. Heavy touring bikes that are frequently lugged at low speeds don't respond well to a late closing intake valve or long duration.

Opening the intake valve early extends the induction period, but it can cause some problems. Opening the intake too early allows exhaust pressure, which still exists in the cylinder to push the fresh fuel charge back into the intake port, thereby diluting the charge. This can upset carburetion, slow the combustion process and reduce combustion pressure. Opening it too late can lean the cylinder and reduce power.

The timing of the exhaust valve *opening* is second in importance of the four timing events because it determines the amount of *blowdown* effect on cylinder exhausting. Blowdown helps clear the cylinder of exhaust gases through combustion pressure rather than depending on piston motion and pipe scavenging. The earlier the exhaust valve opens, the greater the blowdown effect is. The key is not to open the exhaust valve too early, otherwise power (combustion pressure) is lost because usable pressure is tossed out the exhaust due to excessive blowdown.

At high rpm, more power is generally gained by opening the exhaust valve early because blowdown will have a greater effect in purging exhaust gases. At low rpm, however, more benefits are gained by delaying the exhaust valve opening to retain combustion pressures longer than opening it earlier for additional exhausting. Generally speaking, the earlier the exhaust valve opens, the higher up the rpm band peak power occurs. An early exhaust opening also tends to produce a louder exhaust sound. The later the exhaust valve opens, the stronger low-end performance will be.

Closing the exhaust valve too early will trap some of the burned combustion gases in the combustion chamber while closing it too late will scavenge an excessive amount of fresh intake charge out of the chamber and through the exhaust system.

LOBE CENTERLINE

Cam lobe centerline is another factor in camshaft design. The lobe centerline is an imaginary line that simultaneously passes through the point of maximum lift on the lobe's nose (tip) and the camshaft's center of rotation (axis). The point at where the actual centerline of the *intake* lobe occurs in relationship to TDC position of the piston is defined in degrees of *crankshaft* rotation *after* top dead center (ATDC).

The intake lobe centerline is calculated by dividing the intake duration by 2, then subtracting the intake valve *opening*. For example, lets assume a cam has 266 degrees intake duration and the intake valve opens at 31 degrees BTDC and closes at 55 degrees ABDC. This would result in an intake lobe centerline of 102 degrees: 266/2=133, 133-31=102.

The exhaust lobe centerline is calculated by dividing the exhaust duration by 2, then subtracting the exhaust valve *closing*. Lets assume the cam's exhaust lobe has 276 degrees duration and the exhaust valve opens at 64 degrees BBDC and closes at 32 degrees ATDC. The exhaust lobe centerline calculation differs from the intake calculation only in that the valve's closing (instead of opening) is subtracted in the calculation. In this example, the exhaust lobe's centerline is 106 degrees: 276/2=138, 138-32=106. Sometimes the lobe centerline is identical for the intake and exhaust lobes, while at other times it is different. This will be discussed in more detail later.

If the intake lobe's centerline is increased (for example, from 102 to 104 degrees), the cam is considered retarded. This means *all* cam events take place later in the engine's cycle because all lobes are fixed on the camshaft. Retarding the cam causes the intake valve to open and close later, which reduces cylinder pressure at low speeds. Consequently, low speed torque is reduced. However, cylinder pressure at high speeds is normally increased due to the valve's later closing and this increases top-end power. On the other hand, decreasing the intake lobe centerline (for example from 102 to 100 degrees) advances the cam, which increases torque lower in the rpm range, but reduces top-end power. Keep in mind, however, that advancing or retarding the cam advances or retards both the intake and exhaust and both in the same amount. You cannot alter only one event.

As you can see, retarding or advancing the cam can rock the engine's power curve either up or down. This technique is used by knowledgeable tuners to fine tune an engine combination. Assuming the cam is a relatively close match for the engine combination, 4 degrees normally is the maximum amount a cam is advanced or retarded. If advancing or retarding the cam up to four degrees helps performance, it indicates that the cam may not be optimum for the engine combination. If adjusting cam timing more than 4 degrees shows a performance increase, the cam is not properly matched to the engine combination and one with different specifications should be installed. Take note that advancing or retarding the cam 4 degrees will usually move the engine's power band 200 to 300 rpm in either direction. Also, advancing the cam will give more lift near TDC so you need to check valve-to-piston clearance.

The stock Big Twin cam includes the camshaft, a press fit drive gear and a keyway to help locate the gear on the shaft. The drive gear can be repositioned on its shaft to either advance or retard the cam's timing events. Do not attempt to advance or retard a cam by installing it one gear tooth from its normal alignment marks because this will alter the cam timing events 17 degrees, which is too much advance or retard. Instead, the drive gear must be removed from the camshaft and reinstalled at a slightly different position. Before attempting to remove the gear, be sure to mark it where the keyway is cut into the shaft. Crane Cams offers a series of cams that include three keyways. The keyways allow the cam to be installed either in its normal position, advanced 4 degrees or retarded 4 degrees.

When building a maximum effort engine, cam timing should be checked ("degree" the cam) to ensure the valves open and close at the correct time. Due to machining tolerances, it is sometimes impossible to have all valve events correctly timed. Since intake closing is the most important timing event, the cam should be "degreed" to this event. This may require repositioning the cam drive gear. When finished, some engine builders TIG weld the drive gear to the shaft to ensure the gear will not slip.

LOBE SEPARATION ANGLE (LSA)

A major specification that racers note when selecting a camshaft is lobe separation angle (LSA). Lobe separation angle (also known as lobe displacement angle) is the distance measured in *camshaft* degrees between the centerline of the intake lobe and the centerline of the exhaust lobe for the same cylinder. Lobe separation angle is related to lobe centerline and both may be the same value, but they are not the same because they refer to different reference points. Unlike lobe centerline, LSA is ground into a cam and cannot be changed without regrinding the cam.

LSA is one of the few occasions where cam specifications are specified in cam degrees instead of crankshaft degrees. Cam degrees are different from crankshaft degrees since the cam turns at half the speed of the crank. This results in twice as many crankshaft degrees for a given number of cam degrees.

Lobe separation angle has a direct relationship on overlap. Two camshafts having the same duration and lift figures can be ground with different lobe separation angles, which results in different amounts of overlap. In past times, valve overlap was a frequently used term. Although it is still used, it now is often replaced by the term lobe separation angle. LSA now is used because it gives a truer perspective of the lobe's relationship when duration is changed.

For a given amount of duration, as the lobe center angle is decreased (tighter), the following takes place:

- the intake valve opens earlier
- the exhaust valve closes later
- overlap increases
- the time the intake and exhaust valves are closed simultaneously increases
- cylinder pressure increases

Conversely, as LSA is increased (wider), the following takes place:

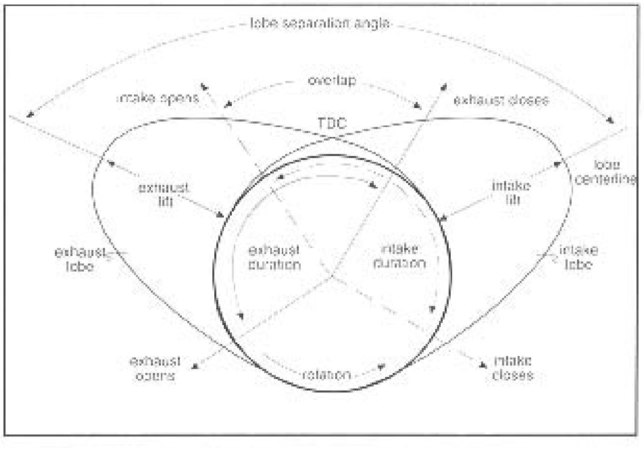
- the intake valve opens later
- the exhaust valve closes earlier
- overlap decreases
- the time the intake and exhaust valves are closed simultaneously decreases
- cylinder pressure decreases

Also, for a given amount of duration, overlap changes when LSA changes; or you can say that for a given LSA, overlap changes when duration changes.

Generally speaking, a given engine combination performs best when the cam is ground within a narrow range of lobe separation. Within this range, an engine will produce greater peak power with one LSA value and a broader power curve with another. Knowledgeable racers fine tune the engine's power curve by first determining the best lobe separation angle then trying different amounts of duration and lift .

For example, lets assume we have two cams that have 252 degrees duration, identical intake and exhaust lobe profiles and the same lift. Cam A is ground with a 102 degree lobe separation angle and has opening and closing specifications of 28-44 for the intake and 52-20 for the exhaust. Cam B is ground with a 108 degree lobe separation angle and has an intake opening and closing of 22-50 and exhaust of 58-14. The tighter lobe separation of cam A indicates it has greater overlap and its intake valve opens and closes earlier while its exhaust valve opens and closes later than that of cam B. This means cam A is likely to have more low and midrange torque due to its earlier closing intake valve, but a rougher idle and lower intake manifold vacuum due to its higher overlap. Cam B, however, will probably make more top-end power and has smoother idle characteristics. The difference in the performance characteristics is not due to lift or duration because these factors remain the same. Instead, it is attributed to the difference in the timing of the opening and closing events.

All this theory may sound great, but what conclusions can we draw from it? The effect of changing lobe separation angle varies depending on numerous factors. By narrowing the lobe separation angle with no change in duration, the intake closes earlier and the exhaust opens later. This improves power at low and midrange be-



Lobe separation angle (LSA) is measured from the centerline of the cam's intake lobe to the centerline of the exhaust lobe for a given cylinder. LSA is one of the few instances where cam specs are given in cam degrees instead of crankshaft degrees. LSA will determine the cam's valve overlap period for a given duration.

cause cylinder pressure is increased, but there is a limit. Narrowing lobe separation too much can increase cylinder pressure to the point of causing low rpm detonation or significant degradation of top-end horsepower. Furthermore, the narrowing of the LSA also increases valve overlap and the longer the duration, the greater the overlap will be for a given LSA. This results in a rougher idle, which isn't necessarily bad on a Big Twin, but it will reduce intake manifold vacuum. The low manifold vacuum may cause the stock electronic ignition to excessively retard the engine's spark advance. Removing the V.O.E.S eliminates the problem.

With the exception of engines sold in California, stock Big Twin cams are ground with an LSA of either 108 or 112 degrees. Their wide LSA is designed to reduce emissions and smooth the engine's idle. Narrowing the LSA increases torque in the low to mid rpm range; however, duration also plays a part. The longer the duration, the wider the lobe separation must be to retain a reasonable idle. So there tends to be some relationship between lobe separation angle and duration. When you look at a wide range of cams, you will see that cams with more duration generally have a wider LSA. This is done so overlap does not become excessive.

A highly tuned Big Twin engine will typically run best with a cam that has a tighter lobe separation angle than stock. This is because as an engine breathes better, a tighter LSA is required to produce maximum power.

When comparing a group of cams that potentially match your engine combination, LSA can give clues how each cam may differ in performance. At least it gives a reference point. Remember, the only sure method for determining a cam's performance is to run the engine on the track.

For a given duration, a tighter (smaller value) LSA generally results in higher low and midrange torque, slightly less top-end power, a more peaked torque curve and a rougher idle. Also, when a tight lobe separation angle is combined with a long duration, it produces a relatively high amount of overlap, which results in a narrowed power band. Conversely, a wider LSA usually increases top-end power, sacrifices some midrange power, has a flatter torque curve and provides a smoother idle. Overall, a wide LSA tends to be more forgiving when other cam specifications are not properly matched to a specific engine combination.

For a given LSA, a short and long duration cam will have a similar torque curve and peak torque rpm. Also, within certain limits, an engine will have a broader power band with one LSA and a more peaked band with another. Additionally, as duration gets progressively shorter, a smaller LSA (within limits) can be tolerated without incurring excessive overlap.

Within limits, changing lobe separation angle a given number of degrees has about twice the effect as advancing or retarding the cam an equivalent number of degrees.

CALCULATING LSA

Calculating lobe separation angle first requires the cam's intake and exhaust lobe centerlines to be calculated. If the intake and exhaust centerlines are identical, then the cam's lobe separation angle is identical to the lobe centerlines. However, if the lobe centerlines are different, one additional calculation must be performed since LSA is half way between the intake and exhaust lobe centerline values. Therefore, to determine lobe separation angle add the cam's intake and exhaust centerline values together and divide by two.

LOBE SEPARATION ANGLE CHARACTERISTICS

CONDITION	NARROW SEPARATION ANGLE	WIDE SEPARATION ANGLE
Intake Event	Starts & Ends Earlier	Starts & Ends Later
Exhaust Event	Starts & Ends Later	Starts & Ends Earlier
Valves Closed Simultaneously	Increased Time	Decreased Time
Overlap	Increased	Decreased
Low Speed Cyl. Pressure	Higher	Lower
High Speed Cyl. Pressure	Lower	Higher
Detonation	Higher Potential	Lower Potential
Manifold Vacuum	Lower	Higher

Table 5.1

EVOLUTION CAMS STOCK ENGINES											
Year	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
84-95	Stock-C	-15/03 03/-15	168 168		.423 .423	-30	99 99	99		Hyd	Stock
84-87	Stock-V	-06/38 25/-03	212 202		.472 .472	-9	112 104	108		Hyd	Stock
88-91	Stock-L	01/37 52/02	218 234	266 280	.495 .495	3	108 115	111.5	.091 .083	Hyd	Stock
92-95	Stock-N	-02/30 31/-09	208 202	250 242	.472 .472	-11	106 110	108	.070 .049	Hyd	Stock

Table 5.2 Valve lift with 1.63:1 rocker arm ratio.

For example, suppose a cam has an intake lobe centerline of 102 degrees and an exhaust centerline of 106 degrees. The cam's lobe separation angle is then 104 degrees: 102+106=208, 208/2=104. In the automotive world, this cam would be considered ground two degrees advanced (the intake lobe reaches maximum lift two degrees earlier than the cam's LSA) because its intake lobe centerline is two degrees less than the cam's lobe separation angle. If a cam has identical lobe centerlines, the cam has "split overlap" and is ground neither advanced nor retarded.

High-performance Big Twin cams are generally ground with a lobe separation angle somewhere between 100 and 109 degrees, although most fall between 100 and 104 degrees. The cams ground with the widest lobe centers typically are long duration, maximum effort cams.

CAM SELECTION

The camshaft is the personality of any engine because no other component affects the engine's power characteristics as much as the cam. Selecting a cam is not only a science, but also an art. Cam selection based on an incomplete under-

EVOLUTION CAMS BOLT-IN*											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	EV13	15/31 45/13	226 238	270 280	.485 .495	28	98 106	102	.161 .148	Hyd	Stock
Andrews	EV3	21/37 43/15	238 238	280 280	.495 .495	36	98 104	101	.197 .159	Hyd	Stock
Andrews	EV46	25/41 49/17	246 246	283 283	.495 .495	42	98 106.5	102	.207 .197	Hyd	Stock
Bartels'	BP40	21/37 48/20	238 248		.495 .495	41	98 104	101		Hyd	Stock
Carl's Speed Shop	CM495F	19/47 50/16	246 246		.495 .495	35	104 107	105.5		Hyd	Stock
Crane	310	16/40 43/19	236 242		.490 .490	35	102 102	102	.164 .185	Hyd	Stock
Crane	H286	19/43 48/24	242 252		.490 .490	43	102 102	102	.179 .206	Hyd	Stock
Head Quarters	HQ-24	20/36 52/19	236 251		.500 .500	39	98 106.5	102.2	.174 .166	Hyd	Stock
Leineweber	E3S	44/62 70/36		286 286	.502 .502	80 at .020"	99 107	103	.215 .170	Hyd/Sld	Stock
S&S	502	28/40 50/24	248 254		.500 .500	52	96 103	99.5	.225 .221	Hyd/Sld	Stock
Screamin' Eagle	406	16/48 51/19	244 250	282 288	.480 .480	35	106 106	106		Hyd	Stock
Sifton	143-EV	20/35 46/14	235 240		.500 .500	34	97 106	101.5	.184 .160	Hyd	Stock
Sifton	145-EV	28/42 50/20	250 250		.460 .460	48	97 105	101	.200 .176	Hyd/Sld	Stock
Sifton	140-EV	30/42 55/27	252 262		.450 .450	57	96 104	100	.222 .178	Hyd	Stock

Table 5.3 *Pistons must have valve reliefs. Valve lift with 1.63:1 rocker arm ratio.

EVOLUTION CAMS MODIFIED 80 CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	EV35	21/37 52/20	238 252	280 298	.495 .530	41	98 106	102	.197 .190	Hyd	Yes
Andrews	EV51	28/44 54/22	252 256	286 290	.510 .510	50	98 106	102	.233 .195	Hyd	Stock
Carl's Speed Shop	CM5	19/47 50/16	246 246		.580 .580	35	104 107	105.5	.185 .176	Hyd	Yes
Crane	H296	24/48 57/25	252 262		.490 .500	49	102 106	104	.200 .206	Hyd	Stock
Head Quarters	HQ-25	18/38 42/14	236 236		.550 .550	32	100 104	102	.166 .150	Hyd	Yes
Head Quarters	HQ-23	19/47 52/24	246 256		.600 .530	43	104 104	104	.172 .191	Hyd	Yes
Leineweber	E5S	39/56 69/26		275 275	.544 .544	65 at .020"	98.5 111.5	105	.225 .132	Hyd/Sld	Yes
Red Shift	575	25/54 63/18	259 261		.575 .575	43	104.5 112.5	108.5	.225 .162	Hyd	Yes
S&S	561	30/47 55/26	257 261		.560 .560	56	98.5 104.5	101.5	.240 .203	Hyd/Sld	Yes
Screamin' Eagle	400	26.5/50.5 55.5/31.5	257 267	304 314	.500 .500	58	102 102	102		Hyd	Yes
Screamin' Eagle	433	23/47 56/24	250 260	286 299	.530 .530	47	102 106	104		Solid	Yes
Sifton	144-EV	27/46 56/22	253 258		.490 .490	49	99.5 107	103	.216 .166	Hyd	Yes
Sifton	141-EV	29/41 58.5/26	250 264		.480 .480	55	96 106	101	.232 .184	Hyd/Sld	Yes

Table 5.4 Note that bolt-in cams also can be considered for this application level. Valve lift with 1.63:1 rocker arm ratio. Refer to Appendix A for additional cam tables.

standing of the factors involved usually leads to less than optimum performance. There are many elements interrelated to the cam such as displacement, induction, exhaust, compression ratio, rpm, gearing and more — and each element leans on something else. As a result, there is an enormous amount of mystery surrounding camshafts and on the surface everything seems complicated. By understanding what affect a cam has on valve timing and engine dynamics, you will have a better knowledge of the cam's role and you should be able to make an intelligent cam decision.

Stock cams are extremely mild, especially the California version. If you're only interested in a bolt-in cam for a mild performance increase, the selection process is easier because there are many functional cams that will give a 10 horsepower boost to a relatively stock Big Twin engine. However, if you're looking for that extra one hundredth of a second on the race track, cam selection gets more difficult and time consuming.

There is no cam that will deliver maximum power from a 1000 rpm idle to maximum rpm. As a result, every cam is a compromise. However, the cam can be used to tailor the engine's valve timing for optimum power in the bike's operating range. To do this, the cam must be matched to the engine, transmission and chassis as a complete combination. The major elements that must be factored into the combination are: engine displacement, rpm range, cylinder head airflow, mechanical compression ratio, lifter type, desired reliability, total bike weight, gear ratios, fuel octane and other variables. No one person can guarantee how a cam will perform until it is run with a particular engine combination and this takes a lot of cut-and-try testing. However, there are some basic guide lines you can follow that will help in the cam selection process. Most major considerations were previously discussed, but the following is a summary.

For a maximum effort engine, all major factors previously discussed should be considered when selecting a cam. At the very least, engine displacement, rpm range and compression ratio must be considered. Then you need to pick two to five different cams and perform cut-and-try track testing for each one.

For an engine where the ultimate cam is not required, start by matching the cam to the engine displacement and desired rpm range. Street engines most frequently are run between 2000 and 4000 rpm (read lower rpm). Depending on the engine combination, drag engines generally rev between 4500 rpm on the low-end and 6500 to 8000 rpm on the high-end. Each gear shift will probably drop the engine's rpm about 2000 rpm. A heavy bike, such as an 800 pound "dresser" and bikes with very high gear ratios (low numerically) cannot handle as much cam as lighter or lower geared bikes.

Consider whether you're willing to remove the cylinder heads to replace the valve spring package and check piston-to-valve clearance. If you don't want to go through this trouble, then you need a bolt-in cam. Obviously, if you're building a complete engine this is not an issue.

Determine whether you want to run stock hydraulic lifters for low maintenance and minimal noise or higher maintenance solid lifters. For a street driven Evolution, hydraulics are recommended. For the Shovelhead, solid lifters are better for performance, although they require periodic adjustment.

Don't forget that the stock electronic ignition limits engine rpm to about 5250, so it doesn't make sense to install a cam that starts making power at 4500 rpm unless you also install an ignition that allows a higher rev limit. In fact, to get the most out of even a mild performance cam, the engine needs to rev beyond 5250 rpm.

One major mistake frequently made when selecting a cam for street use is to install too radical of cam. The result is a functional engine, but one down in power in the lower rpm range. In general, a radical cam drains more power from the low-end than it adds to the top.

The following topics summarize factors to consider when selecting a cam for a given application. Also listed in Tables 5.2 through 5.4 are Evolution cams grouped together by engine application. The tables should be used only as a general reference because, depending on the application, many cams are appropriate for other engine combinations. Refer to Appendix A "Camshaft Specifications" for cams suitable for other engine displacements including the Shovelhead.

Duration

The cam's duration should be matched to the engine displacement and rpm band. A given amount of cam duration will always appear to be milder as engine displacement increases. If in doubt, always favor a milder cam when making a selection. Long duration generally reduces low rpm cylinder pressure, thereby reducing low rpm power. Beyond a given point, increased duration and overlap helps high-end power, but it hurts low-end performance. Engines run on dirt tracks typically can handle more duration than the same engine combination run on hard surfaces.

Duration, timing events, lobe separation angle and overlap are all interrelated. Remember that an early closing intake valve favors low rpm performance while a late closing intake helps top-end power. For two cams with similar duration and lift values and all other things being equal, the cam with the earlier closing intake will generally have improved low-end power. For a given duration, the earlier the exhaust valve opens, the stronger top-end performance will be, but low speed performance will be reduced.

A combination of long duration and a narrow lobe separation angle produces relatively high overlap; consequently, the power band is narrowed. The longer the duration, the higher the required mechanical compression ratio must be for a given level of low-end performance. An excessively long duration cam that is intended to compensate for a poor flowing port will generally allow an engine to turn a high rpm, but acceleration will be poor.

Stock Evolution heads, due to their poor exhaust to intake flow ratio, generally benefit from more exhaust than intake duration (dual-pattern). With ported Evolution heads where the exhaust to intake flow ratio is more favorable, consider cams with more similar intake and exhaust duration figures. Due to its higher velocity ports, an Evolution engine generally needs about 10 degrees less intake and exhaust duration than a comparable Shovelhead engine.

Compression

When selecting a cam, be sure to consider cylinder pressure at both low and high rpm. Large amounts of duration coupled with low compression kills low speed performance. Some engine builders install a long duration cam that effectively bleeds off cylinder pressure at low rpm. The intent is to eliminate low speed detonation encountered by a street engine run on low octane gas and/or having inefficient combustion chambers. This technique works, but it kills low speed power. A better technique is to rework the inefficient combustion chambers for greater turbulence and better flame travel, then install a cam with the proper amount of duration for the engine combination. Also, consider dual spark plugs that shorten flame travel and combustion time because the potential for detonation will be reduced.

For a stock displacement street engine with a performance cam, raise the mechanical compression ratio at least one half point and create as much combustion chamber turbulence as possible to offset low-end power losses.

A long duration cam with a narrow lobe separation angle generates low cylinder pressure at low rpm and is a poor match for a low compression engine.

An engine with a low mechanical compression ratio responds favorably to shorter duration, wide lobe separation angles, quick lift rates and relatively high lift. These characteristics allow for more area under the cam's lift curve while retaining excellent low rpm cylinder pressure.

When cylinder pressure is reduced at low rpm due to long duration and/or narrow lobe separation angle, the engine will probably respond favorably to a quicker ignition advance curve.

Lift

Stock Evolution heads stop flowing at about .500-inch lift, so a bolt-in cam may be all that is needed for unmodified heads. Ported heads definitely can take advantage of lifts .550-inch or greater.

Keep in mind that the cam's lift rate will have a direct bearing on performance. Beyond a certain point, the faster the cam's lift rate, the higher the valvetrain wear will be. But performance will be increased. Rapid lift rates require higher spring pressures to maintain proper valvetrain control. The higher the total valve lift, the faster the valves and guides will wear. Roller rocker arms and the correct rocker arm geometry will minimize the rate of wear.

Cams with very high lift typically require more work to install. Due to cylinder head and port design, Evolution cams frequently have .050 to .100-inch more lift than comparable Shovelhead cams.

A custom ground cam with a faster than stock lift rate is sometimes installed when class restrictions require stock duration and lift. The faster lift rate puts more area under the "lift curve" while still retaining stock duration and lift specifications.

Lobe Separation Angle (LSA)

Remember that the optimum time for opening and closing valve events changes with engine speed. For increased low speed torque and power, the intake events should take place earlier and the exhaust events later in the cycle. On the other hand, to enhance high speed performance, the intake events need to take place later in the cycle and the exhaust events earlier.

For higher low and midrange power, consider a cam with shorter duration, higher lift and a narrower LSA. Conversely, for greater top-end power a longer duration and a wider LSA are best. For a given engine displacement, a wide lobe separation angle will help top-end power as long as there is a *sufficient* amount of duration to support the wide lobe separation. A key point to remember is that a wide LSA, when combined with an insufficient amount of duration will generally not reflect improved top-end power.

Mild Big Twin Evolution cams generally have a lobe separation angle ranging between 100 and 104 degrees while hotter cams typically fall into the 103 to 110 degree range. Mild Shovelhead cams normally have a 100 to 102 degree lobe separation angle while hotter versions generally end up in the 103 to 110 degree range.

VALVETRAIN

We know that moving as much air as possible through the engine is a key factor to making large amounts of power. We also know that selecting the correct cam for a given application can maximize the amount of air flowing through the engine and can optimize the timing of the air's movement. However, for these things to take place, the cam must accurately transmit its lift, duration and timing to the valves.

Stroker engines are usually revved 6000 to 7000 rpm while stock stroke engines are often turned 7000 to 8000 rpm. As a result, one of the major causes if not the largest cause of engine failure originates in the valvetrain. Assembling all the valvetrain components in the correct relationship to each other is critical to engine reliability. Lighter, stiffer and stronger parts are typically the norm. However, stronger parts sometimes can mean heavier parts and this can encourage valve float. With the large selection of camshafts, pushrods, valves and valve springs, reliability problems sometimes stem from mismatched or weak components. Therefore, to achieve maximum horsepower and engine reliability, the proper selection and assembly of all components located between the cam and the valve is vital.

Rigidity is crucial if the valvetrain is to transmit accurately the cam lobe's action to the valve. At high rpm, pushrods can flex and bend under extreme pressures. This causes inaccurate valve timing, uneven cylinder firing and reduced power. Sloppy or worn camshaft bearings, poor fitting lifters, inadequate pushrod strength, worn rocker arm bushings and worn valves and guides also contribute to inaccurate valve control.

Many engine builders subscribe to the "bigger is better theory" and concentrate most of their time and money on the more glamorous performance items. Yet understanding the fundamentals of the valvetrain and assembling it correctly will pay big rewards to those willing to take time to learn about the factors involved and then apply this knowledge to part selection and engine assembly.

The valvetrain is the mechanism used to convert the rotating motion of the camshaft into the motion necessary to open and close the valves. This process requires the motion to be redirected about 180 degrees, which means there is a continual starting and stopping of the components involved. As a result, extreme pressures and forces are involved to ensure the valvetrain follows the cam profile and closes the valve properly. Starting at the cam and working upward, the valvetrain components include: lifter, pushrod, rocker arm and rocker shaft. Additionally, springs are mounted on the valves with collars and retainers to apply pressure to close the valve.

LIFTERS

A lifter (sometimes referred to as a tappet) sits directly above the cam and makes direct contact with the cam's lobe. As the lifter moves up and down, it transfers its energy to the pushrod, which activates the rocker arm and finally the valve. Lifters are divided into two categories: hydraulic and solid (mechanical). All Evolution and Shovelhead engines are equipped from the factory with hydraulic roller lifters.

HYDRAULIC LIFTERS

A hydraulic lifter uses engine oil pressure to maintain zero clearance in the valvetrain. The zero clearance leads to quieter operation, less vibration and less maintenance. With some cams, the lifters are not interchangeable from hydraulics to solids. The profile of the cam's lobe determines whether a hydraulic or solid lifter is required. Some cams are compatible with either lifter type. A cam ground for a hydraulic lifter has more aggressive ramps and typically needs less duration than a comparable solid lifter cam. Be sure to check for lifter compatibility when selecting a cam because excessive valvetrain noise or serious engine damage can take place if the wrong style lifter is matched to the cam.

A hydraulic lifter consists of a lifter body, roller and hydraulic plunger. The plunger maintains zero clearance by riding on a small chamber of pressurized engine oil. The pressurized chamber works in conjunction with small bleed holes to allow the plunger to move up and down to maintain zero clearance. This design automatically compensates for cylinder length expansion due to engine heat.

The hydraulic lifter is fitted with a needle bearing roller that contacts the cam lobe. The roller allows a more aggressive cam lobe profile, which puts more area under the cam's lift curve.

Since hydraulic lifters depend on oil pressure to operate, the fit between the plunger and lifter body, and the fit between the lifter body and lifter block are important. A loose fit reduces oil pressure to the lifter, which causes the lifter to collapse slightly and then pump back up. The intermittent collapsing and pumping result in a rough running engine and lost power. The rough running is due to the changes in valve timing while the lost power is attributed to varied valve timing and loss of lift. At high rpm a hydraulic lifter can over-pump (pump-up) and cause valve float, which results in reduced power and possible engine damage. Another characteristic of hydraulic lifters is that when the engine is cold, oil pressure is high because the oil is thick. As the engine warms, the oil thins and oil pressure drops. This causes the valve timing and lift to fluctuate.

Shovelhead lifters do not have precision machine tolerances and are very susceptible to lifter collapse and pump-up. Even a closely machined set has difficulty handling much more than a stock lift cam. As a result, to get reasonable performance from a Shovelhead with a performance cam, you need to run solid lifters. However, the solid lifter has a difficult time compensating for the .030 to .050-inch growth of the Shovelhead cylinders. One option to solid lifters is a set of precision Velva-Touch hydraulic lifters. These lifters are machined to extremely precise tolerances, which eliminates problems caused by solid or stock hydraulic lifters because they maintain zero valvetrain clearance even with high cylinder growth.

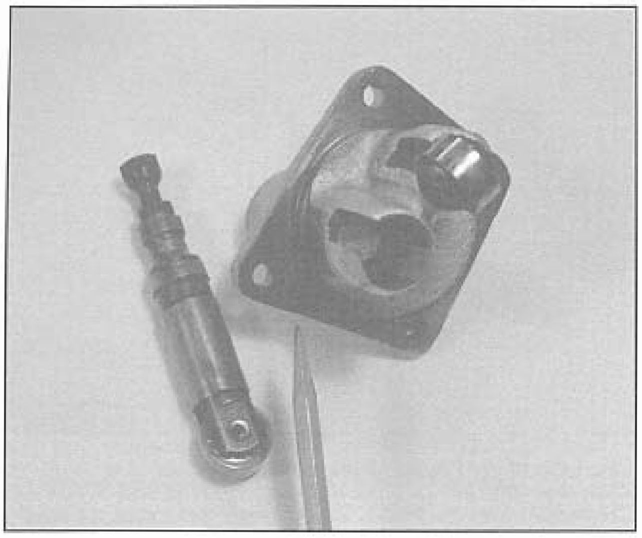
The Velva-Touch Lifter Kit eliminates hydraulic lifter pump up and maintains zero valve lash to over 7000 rpm. Machined to close tolerances, these lifters eliminate the alternate collapsing and pumping up of stock lifters and maintain accurate valve lash even during high cylinder growth. Photo courtesy of V-Thunder.

Shovelhead hydraulic lifters easily collapse due to loose tolerances and low oil pressure. S&S Shovelhead Solid Lifter Conversion Kit with non adjustable pushrods is one remedy for the problem. Photo courtesy of S&S Cycle.

The Evolution's hydraulic lifter is far superior to the Shovelhead's. Depending on the precision of its machined tolerances, it can handle up to about 6300 rpm before pumping up and causing valve float. As a result, most mild and moderate aftermarket cams for the Evolution use hydraulic lifters. On the other hand, most maximum effort Evolution cams require solid lifters because they can handle more aggressive lobe profiles and are lighter in weight.

In 1986 the stock Evolution lifter was improved by increasing the diameter of its plunger. The change allows the lifter to maintain more consistent valvetrain clearances by reducing the potential for collapsing. The plunger for the 1984 and 1985 lifters is about .042-inch smaller in diameter than the later models. The factory now offers an upgrade kit to bring the 1984 and 1985 model lifters to the latest standards. When a high lift cam is combined with the early lifters, it sometimes results in the lifters making a ticking sound, especially the front exhaust. Be sure to install the lifter upgrade kit if you have an early model Evolution engine.

Hydraulic lifters can be converted to solids by installing either a lifter conversion kit or by replacing the entire hydraulic lifter with a solid lifter. When converting lifters to solids, be sure to plug the lifter block's oil feed hole (located on the bottom side of the block's mounting base) with an 8-32 Allen head set screw 3/16-inch long. This will prevent excess oil from escaping beyond the lifter and filling the pushrod tube. Any



The pencil points to the location of an oil hole in the lifter block that needs to be blocked off when converting from hydraulics to solids. Shown is a Shovelhead lifter block.

excess oil increases drag, requires additional oil scavenging and can cause an oil leak at the pushrod tube.

S&S Products offers a Hydraulic Lifter Limited Travel Kit for the Evolution that is designed to make the hydraulic lifter work like a solid lifter at high rpm, yet like a hydraulic at lower speeds. This eliminates lifter pump up for more precise valve action without converting to a full solid lifter. Also, never run solid lifters with a stock cam and pushrods.

Remember that hydraulic lifters require sufficient oil pressure to work properly. Yet the Big Twin's roller bearing crankpin requires the engine to run with relatively low oil pressure because high pressure causes the rod bearings to skip over the crankpin, which creates flat spots. This results in a paradox because the hydraulic lifters work better with higher oil pressure, yet the engine is designed for relatively low pressure. Low oil pressure or thinned oil on a hot day can cause the lifters to collapse intermittently, thereby creating a ticking sound and subsequent loss of valve control and performance. And this condition is more prevalent when a performance cam is installed, especially with the front exhaust. A slightly higher tension oil pressure relief spring can be installed in the pump to increase oil pressure slightly while not interfering with the bearing's lubrication requirements.

SOLID LIFTERS

A solid lifter uses a roller to follow the cam lobe the same as a hydraulic lifter. However, a solid does not incorporate an absorption system to maintain operating clearance in the valvetrain. This requires valve lash (clearance) to be adjusted at engine assembly time and periodically there after. Correct adjustment for a solid lifter is when the pushrod is finger spin tight on a cold engine. This should result in a clearance between zero and .002-inch. Turning the lifter adjustment screw a distance equal to one flat changes clearance .006-inch.

Aftermarket solid lifters (Evolution and Shovelhead) are lighter in weight than hydraulic lifters. Also, some solids are available in a shorter length. The short lifters are used in conjunction with longer pushrods to help minimize the severe angle on the pushrods (especially the front exhaust), which hampers valvetrain geometry. Adapter kits are available that convert Shovelhead hydraulic lifters to solids. This offers a low cost method for converting to solids, but the converted lifters will not be as light weight as true solids.

Before installation always check the location of the lifter's roller axle to determine if it is centered on the lifter body. If the roller's axis is not centered, valve timing will be affected. The axle's position can be checked by laying the lifter body on a flat surface with the roller in a vertical position and extending off the edge of the surface. Set a dial indicator on the roller for an initial reading. Now turn the roller 180 degrees and take another reading. For maximum performance, use lifters that have a roller axis difference of no more than .005-inch.

Some high lift cams can raise the lifter high enough so it hits the pushrod tube's lower oil seal (the seal located between the tube and lifter block), thereby causing an oil leak. To eliminate any contact, use a die grinder with a fine cartridge roll to chamfer the lifter's top edge. The chamfer will also prevent the sharp top edge of the lifter from cutting into the lifter bore.

For maximum effort engines, the lifters can be lightened by grinding flat areas on each of its sides (90 degrees from the thrust faces) and by removing weight from its neck area where the adjusting screw attaches. The adjuster screw's hex-head and lock-nut can also be ground to a smaller size and the lifter's roller can be narrowed two-thirds by machining a 45 degree angle on each edge. Years ago I ground flats on lifters and narrowed the rollers and the result usually was a cracked lifter body and excessive wear on the cam lobes. Therefore, it's usually best to spend your time and efforts on other engine modifications.

If you're concerned about using light weight lifters, the aftermarket solids (not converted from hydraulic) offer the lightest weight available.

LIFTER BLOCKS

The engine's four lifters are held in place by two lifter blocks — one lifter block for the two front cylinder lifters and one for the rear cylinder's. The lifter blocks are bolted to the crankcase and require a 12-point socket to remove the hold-down bolts.

Be sure to check the lifters for proper clearance in the lifter blocks. A loose fit causes inaccurate valve timing and loss of duration and lift. Worn and scored lifters or blocks can significantly reduce power and make it impossible to properly degree a cam. Never change lifter positions. Always replace each lifter in the same bore of the lifter block. To save the high cost of replacing lifter blocks, .005 and .010-inch oversize diameter lifters are available.

For maximum effort engines, the position of the lifter bore centerline to the cam is extremely important. Ideally, the axis of all lifter bores should be perpendicular to the cam's axis. But this dimension can be slightly off without significant power loses. However, there are two other dimensions that can ruin the proper timing of cam events: the relationship of the camshaft centerline to the crankshaft centerline and the camshaft centerline to the lifter bore centerline.

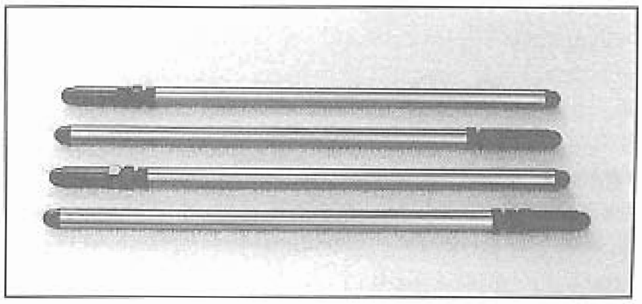
First, the Big Twin's camshaft centerline should be offset toward the rear of the engine .094-inch from the flywheel centerline. With some crankcases it may be off. Also, if a vertical line is drawn upward from the flywheel centerline, the line should evenly divide the 45 degree angle between the lifter block decks and cylinder decks into two 22.5 degree angles. The decks should be evenly spaced from and parallel to the flywheel centerline axis and exactly 45 degrees (22.5 degrees included angle) to one another. In some situations, the decks are slightly tilted off-center and are not evenly divided. Machining the crankcase cylinder decks and the lifter block decks ensures that all surfaces are at the proper angle to one another.

Another critical aspect of proper cam timing is the relationship of the lifter bore centerline to the camshaft centerline. These two centerlines must intersect one another, otherwise cam timing will be either early (lifter axis biased toward the front of the engine) or late (lifter axis biased toward the rear of the engine). Also, maximum valve lift will not be achieved. In many cases, it is difficult to determine what part is off. Possible solutions include repositioning the lifter blocks, machining the lifter bores for oversize lifters, or sleeving the lifter bore to change the lifter axis.

These procedures are very time consuming and costly and should only be considered for maximum effort engines. For other engines, be sure to use a lifter block alignment tool (listed in the Crane Cams catalog) to align the blocks during assembly.

PUSHRODS

The pushrod is a very simple part that connects the lifter to the rocker arm. However, the difference between a good pushrod and a bad one is straightness, strength, weight and length. Shovelheads are equipped with solid aluminum adjustable pushrods. The Evolution has hollow steel one piece non-adjustable pushrods. The Evolution's pushrod not only activates the rocker arm assembly, but it also passes oil to the top-end through its hollow interior.



Adjustable chrome-moly pushrods are needed to simplify Evolution cam installation and keep valve timing accurate at high rpm. Chrome-moly pushrods are heavier than aluminum, but flex less at high rpm. For maximum effort engines, consider lighter, non adjustable pushrods. Shown are Andrews adjustable chrome-moly pushrods for the Evolution.

Different length pushrods are required for each pushrod position because of the different angles leading to the rocker arms. The front exhaust pushrod is angled the greatest amount, which results in its geometry being the worst of all the valves. The more in-line the pushrod is in relationship to the lifter axis, the better the geometry. The Big Twin's poor valvetrain geometry, which is caused by the extreme angularity of its pushrods, is a major reason why some racers have resorted to a four-cam crankcase design like that of the Sportster.

The correct pushrod length must be used in each valve position to help maintain correct valvetrain geometry and adjustment. The longest pushrod goes to the front exhaust, the second longest to the rear exhaust, the next longest to the front intake and the shortest to the rear intake. Some manufacturers make the two intake pushrods the same length. Stock length pushrods will work properly as long as the cam's base circle is the same diameter as the stock cam's and stock length cylinders are used. Long pushrods are required when extra long cylinders are installed. Be sure to use long pushrod cover keepers with long pushrods.

STOCK EVOLUTION PUSHROD GUIDE		
POSITION	**LENGTH**	**COLOR**
Rear Intake	11.045"	Blue
Front Intake	11.090"	Yellow
Rear Exhaust	11.210"	Purple
Front Exhaust	11.325"	Green

Table 5.5

The stock Shovelhead pushrod is heavy and the Evolution's pushrod is impractical to use with high lift cams. Stock pushrods should be replaced with either aluminum or heat treated 4130 chrome-moly units. Aluminum pushrods are lighter while chrome-moly units have more rigidity. Aluminum flexes about three times as much as chrome-moly when under high load. When pushrod flex is combined with hydraulic lifters, lifter pump-up often occurs, resulting in loss of valve control. Reducing valvetrain weight is always a goal when building a performance engine. However, when trading off less weight for rigidity, take rigidity every time. Accurate valve timing is much more important than reduced weight. Overall, you're much better off with chrome-moly pushrods, although they are noisier than aluminum when used with solid lifters.

The larger the pushrod's outside diameter (O.D.), the thinner walled it can be for a given amount of strength. Depending on the material used and the manufacturer, aftermarket pushrods normally vary between 3/8 and 7/16-inch O.D. and between .038 and .095-inch wall thickness. The larger the pushrod's diameter, the greater the chance it will rub against another engine part. Long pushrods and high valve spring pressure require a stronger pushrod tube. As tube strength goes up, its weight also goes up. Consequently, higher spring pressures are required.

For a Shovelhead with solid lifters, the one piece non-adjustable 4130 chrome-moly pushrods are best for maximum performance. Aluminum is okay for lesser applications. For the Evolution, adjustable chrome-moly pushrods and hydraulic lifters is the way to go unless you're building a maximum effort engine. In this case, use chrome-moly one-piece (non-adjustable) pushrods and solid lifters with adjusting screws. It will take some extra work to determine the correct length for the non-adjustable pushrods, but it's worth the effort due to lighter weight and elimination of variables introduced by the adjusting mechanism. A special adjustable pushrod can be used to determine the correct length for one-piece pushrods. The precision Velva-Touch hydraulic lifters and pushrods can handle over 7000 rpm and are used in many performance applications.

For maximum performance, pushrods must be perfectly straight. If you find a bent pushrod, either the pushrod is too weak for the valve spring pressure or there is interference such as coil bind somewhere in the valvetrain. Check for any shinny wear marks on the pushrod that indicate interference. Any marks you find may be due to a lack of clearance or possible flexing. A pushrod can handle high loads along its axis, but it will bend relatively easily if it is side loaded, so remove any interference.

The best way to check a pushrod for straightness is to roll it across a perfectly flat surface such as a piece of glass. If it doesn't jump around, it should be okay. While you're at it, make sure the pushrod's hardened steel inserts are a tight press fit in the tubing.

ROCKER ARMS

The pushrod actuates the rocker arm, which causes the arm to rotate at its fulcrum or pivot point and depress the valve. The rocker arm rotates on a shaft and uses replaceable bronze bushings to maintain the correct running clearances for the shaft. The rocker arm contacts the valve stem through a radiused pad. Aftermarket rocker arms that use a roller tip instead of a radiused pad are available and some even have needle bearings in place of the bronze shaft bushings.

The valve-to-rocker arm relationship is extremely critical for maximum performance and reliability. To minimize valvetrain stress and optimize valve lift and duration, the angles of the pushrod-to-rocker arm and rocker arm-to-valve stem should be 90 degrees when the valve reaches 50 percent of its maximum lift. This relationship applies to both radiused pad and roller tipped rockers. The angle between the axis of the rocker arm (when measured from the center line of the rocker shaft to either the radiused pad or roller tip axle centerline) and the end of the valve stem should be less than 90 degrees when the valve is fully closed. At maximum lift the angle should be greater than 90 degrees. Setting up the rocker arm geometry in this manner minimizes valve side-loading and optimizes valve timing since the rocker applies pressure equally to the valve stem. It also lifts the valve to its maximum potential.

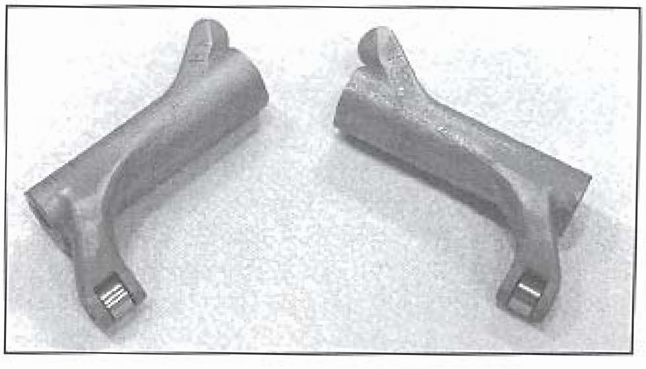
There are a few things that can be done to correct inaccurate rocker arm geometry. First, the valve stem height can be adjusted. When the valve is sunken into its seat, its stem sits higher in relationship to the rocker arm. Also, when large valves are installed, it is possible for the valve stem to sit lower than its normal position. Sometimes incorrect geometry can be corrected by installing new valve seats. At other times, valves with extra long stems are required. When installing a high lift cam, extra long valves are sometimes required for correct spring spacing between the collars.

Valve stem lash caps, which are small caps that fit over the tip of the valve stem, are sometimes used to help adjust valvetrain geometry or to protect the valve stem in severe usage applications. When installed, they essentially increase valve stem height. The caps are typically precision machined from hard 8620 steel alloy and heat treated to survive the high loads from the rocker arm.

Nevertheless, it is critical that all valve stem heights are equal and set within the prescribed limits for optimum valvetrain geometry. Unequal valve stem heights result in unequal filling and scavenging between cylinders.

Rocker arm geometry can also affect where the rocker arm's radiused pad or roller tip contacts the valve stem. The rocker arm's point of contact with the valve stem moves back and forth as the valve moves up and down. At zero lift, the rocker arms contact point should be slightly toward the rocker side of the valve stem. This will allow the rocker to sweep back and forth equally against the valve stem, reducing side loading and wear. The radius pad on solid tipped rockers can be refinished (with the correct radius) to adjust the valve stem contact point. Valvetrain geometry should be checked at the same time the heads and valves are setup. Also, some professional head porters offer special services that go beyond the basic valvetrain geometry checking.

A number of companies offer roller tipped rocker arms. The roller tip reduces valve stem tip wear, galling, friction and heat. It also provides greater valve stem tip contact. Some roller



Roller rocker arms decrease friction and reduce valve and guide wear. They are suitable for street and race engines and are especially helpful when used with very high lift cams. Shown are Baisley Pro-street roller rockers. Photo courtesy of Great Lakes Cycle.

tipped rockers include shaft needle bearings in place of bronze bushings to further reduce friction. Racers running dry top-ends frequently use needle bearings instead of bushings.

For maximum effort Evolution engines, the rocker arm's width can be shortened to help improve valvetrain geometry. This allows the top of the pushrod to be moved toward the left (clutch side) as viewed when sitting on the bike. Besides shortening the rocker arm width, this modification requires machining the rocker cover and running a dry top-end.

ROCKER ARM RATIO

The rocker arm ratio is the difference between the distance the valve moves and the distance the pushrod moves. The ratio is accomplished by offsetting the rocker arm axis toward the pushrod side of the arm. The Evolution has a rocker arm ratio of 1.6:1, while the Shovelhead is 1.43:1. Changing the rocker arm ratio determines how far and how fast the valves open and close. Remember, the faster the valve is lifted, the more area under the lift curve.

The approximate gross valve lift is calculated by multiplying the camshaft lobe lift by the rocker arm ratio. For example, if the rocker arm has a 1:43:1 ratio and the cam's lobe lift is .309-inch, then the gross valve lift is .442-inch (1.43 x .309 = .442). With a 1.6:1 rocker ratio and the same lobe lift, the gross valve lift is about .495-inch. However, with deflection in the valvetrain caused by pushrod flex and rocker arm bending, the net valve lift is reduced.

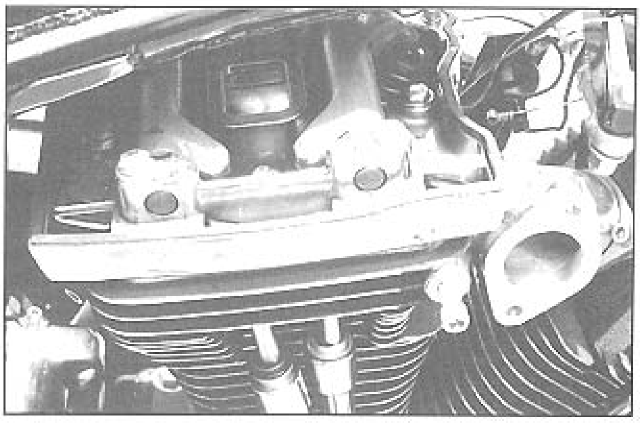
A higher rocker arm ratio will lift the valve faster and farther for a given camshaft profile. A larger ratio will make a cam appear more radical to the engine while a smaller ratio makes it appear milder. Besides lift, the rocker arm ratio also affects cam duration. The greater the rocker arm ratio, the longer the duration will be when measured at the valve. For example, since a 1.6:1 rocker arm ratio lifts the valve faster, it will reach .053-inch lift earlier (in less crankshaft degrees) than a 1.43:1 ratio rocker arm will. It will also close the valve later than a lower ratio. As a result, increasing the rocker arm ratio increases camshaft duration at the valve because the time (as measured in crankshaft degrees) the valve is open beyond .053-inch increases.

Keep in mind, however, that high ratio rocker arms are not always the better setup. For example, it is possible to over scavenge a cylinder, thereby tossing horsepower out the exhaust system with the same result as if you ran too radical of cam for the engine combination.

Currently, high ratio rocker arms are available on a custom basis from some professional head porters and soon should be available from others as a standard item.

ROCKER ARM INSTALLATION

Besides proper valvetrain geometry, there are a number of other factors that should be considered when installing rocker arms. Some racers have spent a great amount of time and energy lightening rocker arms and the result frequently was broken rockers. However, even if the rockers don't break due to excessive lightening, they can flex more than normal because of reduced strength. The result is loss of valve timing and horsepower. Consequently, an engine builder's time is usually better spent modifying other engine components. Polishing rocker arms to eliminate any stress risers is a good idea, but if you feel compelled to lighten them, only remove material from the sides of the arms and not from the top and bottom. In this situation, vertical mass is more important than horizontal mass.



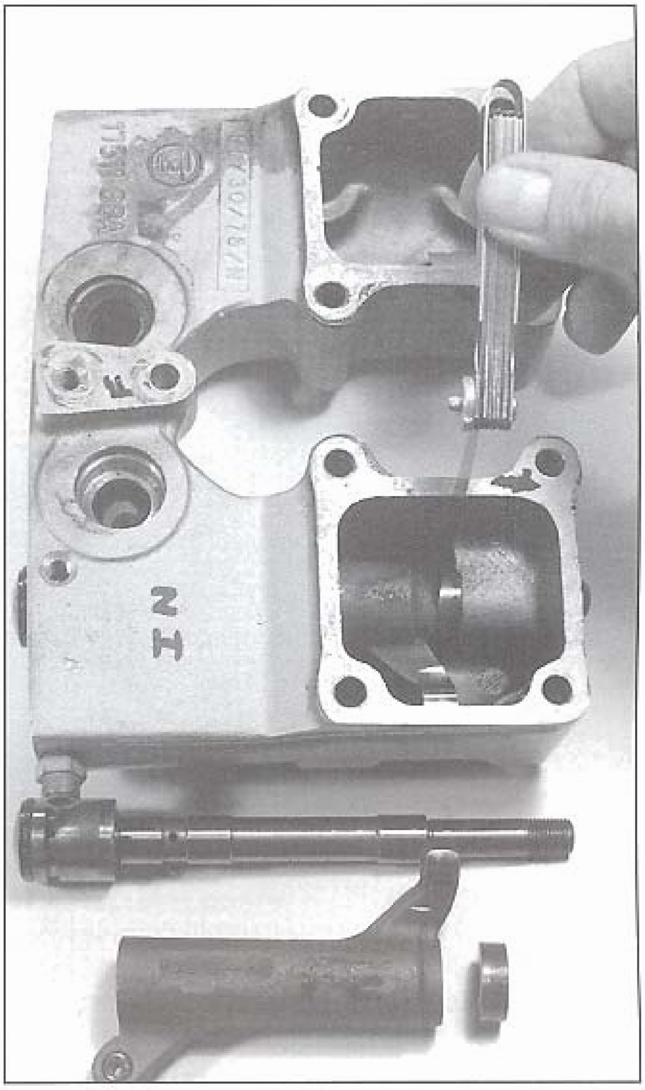
With few exceptions, cam installation in an Evolution engine requires removal of the top rocker covers and rocker arms. Numerous methods have been tried to install an Evolution cam and in the end it's easiest to just remove the rockers. Removing the bolt near the pushrod side of the rocker arm frees the rocker shaft.

When installing a performance cam, valves, valve springs and collars, be sure to check for rocker arm interference. There are two major areas to check: between the top of the rocker arm (valve side) and the rocker cover; between the underside of the rocker arm (valve side) and the outside edge of the top valve spring collar when the valve is in a closed position. Use clay or machinists dye to check each area for clearance. Make sure both of these areas have a minimum of 1/16-inch clearance, otherwise valve lash cannot be set properly and the potential for engine damage is high.

Shovelhead engines use a shim to control the rocker arm's side-to-side clearance. The factory recommended clearance is .004 to .025-inch. The valvetrain geometry remains more stable if the side clearance is set at .004-inch. Assemble the rocker arms without any lubricant, torque the shaft nuts to factory specifications and check the clearance with a feeler gauge. Install any necessary shims on the rocker arm shaft before placing the rocker arm on the shaft. When finished, make sure the rocker arm rotates freely on the shaft.

With an Evolution engine, check the hole in the rocker cover that the rocker shaft fits through. Make sure the hole is round, not elongated and no sign of cracking exists. Billet aluminum rocker shaft supports, which mount on the rocker cover's pushrod side, are available to eliminate flexing, cracking and wear of the stock cast rocker support.

Oil flowing to a Shovelhead's top-end can be regulated to minimize unneeded oil. The top-end is supplied oil through a thin external oil line tube that passes oil from the crankcase to the rear cylinder rocker cover. The oil line has fittings at both ends that connect it to the crankcase and rocker cover. The hole in one of the fittings can be soldered closed and drilled for a .040 to .050-inch hole to limit the oil supply.



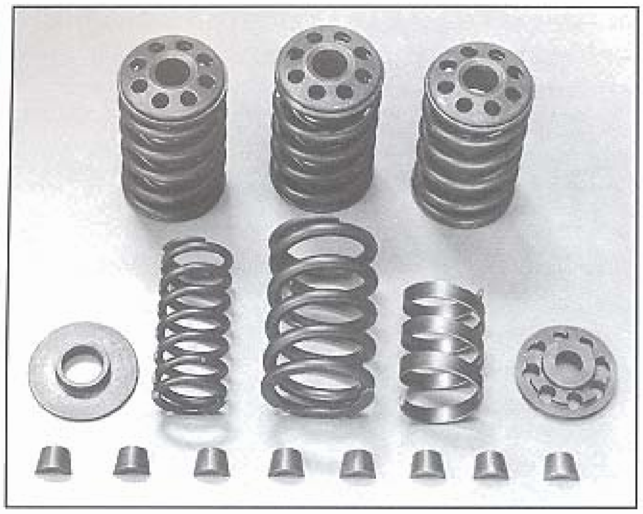
You can help the Shovelhead's valvetrain geometry by setting up the rocker arms with .004-inch end play. Use a feeler gauge to measure the clearance between the spacer on the valve side of the arm and the rocker. Alter the clearance by changing the width of the spacer or adding shims to the pushrod side of the rocker arm shaft. Note that the rocker arm pads also have been radiused for better valvetrain geometry.

VALVE SPRINGS

Most performance cams lift the valves higher and accelerate them quicker than a stock cam. This requires higher valve spring pressure to eliminate valve "float." It also requires longer spring travel to prevent spring coil bind. Increasing spring pressure increases stress, heat and fatigue and can cause the spring to lose tension quickly. Therefore, a quality spring with sufficient pressure, along with strong lightweight collars is the combination to look for when selecting spring sets.

Valve float is the condition where the valve loses contact with the valvetrain due to high inertia and it stays open when it should be closing. Overweight valvetrain components, low valve spring pressure and high rpm are the major causes of valve float.

Valve spring pressure is normally checked when the valve is seated and when it is at



A performance engine needs positive valve action to reach its maximum potential. Performance spring kits provide increased pressure for positive valve action, greater installed height for high lift cams and light components to minimize valve float at high rpm. Shown is the S&S .630" lift triple spring kit with drilled chrome-moly collars. The flat wire spring serves as a harmonic dampener to reduce valve float. Photo courtesy of S&S Cycle.

maximum lift. Although high spring pressure at maximum valve lift catches everyone's attention, keep in mind it is seat pressure that closes the valve. A mild performance cam requires about 140 to 160 pounds spring pressure with the valve on the seat and 300 to 330 pounds when it is at full lift. Moderate performance cams generally need about 180 pounds on the seat and 350 to 400 pounds wide open. Maximum effort cams usually have 200 pounds or more spring seat pressure and 450 to 600 pounds or more at maximum lift. For comparison, stock Evolution spring pressure is about 125 pounds on the seat and 280 to 320 pounds wide open. Shovelheads have slightly less pressure.

Stock Evolution springs can handle up to about .500-inch valve lift. If you're frequently going to rev a mild bolt-in Evolution cam beyond 6000 rpm, you should consider a set of aftermarket performance springs and collars. You won't need the extra pressure or spacing they offer, but you can benefit from their ability to retain pressure longer under hard use. If you use stock springs, take note that the 1990 and later Evolution springs seem to retain pressure longer than earlier versions.

Stock Shovelhead springs are limited to about .440-inch valve lift. Because the stock springs have limited travel, performance springs are usually installed and bolt-in cams are less frequently used in the Shovel than the Evolution. Nevertheless, be sure to match the correct springs to the cam.

The shape of the cam lobe, weight of the valvetrain and the maximum engine rpm determine the required spring pressure. Running excessive spring pressure increases friction and heat and reduces horsepower. However, from a reliability standpoint, too little spring pressure is worse than too much. Keep in mind that spring pressure will drop after even a short run-in, so watch it closely on serious race engines. It's not unusual to lose 10 or 15 pounds of spring pressure after the initial run-in. This means that if you're shooting for 170 pounds seat pressure, you may need 185 pounds at installation time.

The only way to know for sure what a spring's exact pressure is at a given installed height is to check it with a valve spring tester. Any spring more than five percent under specifications should be replaced. Also, check each used spring for proper free length by placing it on a flat surface next to a new one. Any used spring that measures 1/16-inch shorter than a new one is probably fatigued and usually down in pressure.

Additionally, a spring must sit perpendicular against the spring collar, otherwise excessive wear and heat will be generated. Inspect both ends of every spring with a machinist square for correct squareness.

Some racers pre-condition new springs before installation. The procedure involves placing a spring in a soft-jaw vise and gently compressing it several times without coil binding it. But be careful, you can easily ruin a spring if you're not careful.

Various valve spring kits are available that satisfy required spring pressures and simplify cam installation. Most of the kits use round wire dual interference fit springs, although some include a third spring. The interference fit between the dual springs is designed to reduce harmonics and control spring surge. The third spring is made of either round or flat wire and it will add a slight amount of pressure. If it is made from flat wire, it is designed primarily to dampen natural spring frequencies. This can reduce spring surge, which quickly fatigues a spring. Better springs are frequently made from chrome silicon wire.

Top and bottom spring collars are available in aluminum, chrome-moly and titanium. The aluminum collars are the lightest, the chrome-moly collars the strongest and the titanium collars offer the best combination of strength and light weight. Some valve collars are machined for additional spring travel, which can ease installation. Additionally, bottom collars are designed to fit either "shoulder" or "shoulderless" valve guides. Be sure to match them properly to the valve guide.

Machined heat treated chrome-moly valve keys cost more than plain steel stamped units, but they're the safest way to hold together a high revving engine and are recommended for use on all engines.

Nicks and scratches on a spring cause stress risers and can lead to spring failure. Spring life can be improved by very carefully using a die grinder with a fine cartridge roll to deburr any irregularities and to chamfer sharp edges on the spring's flat surfaces. More expensive spring kits include magnafluxed, bead blasted and thermo-coated springs. The thermo-coating reduces friction generated heat and increases heat dissipation.

VALVETRAIN SETUP TIPS

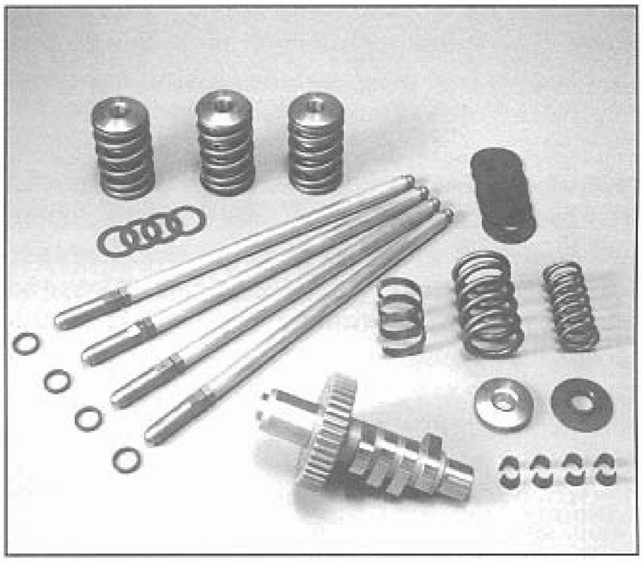
There are many factors to consider when setting up an engine's valvetrain. Most factors have been discussed. However, it probably would be helpful to summarize them.

The rigidity of each component and the angular relationship of the components are critical to achieving the ultimate performance from the engine's valvetrain.

The most important job the valvetrain has is to transmit the action accurately from the cam lobe to the valve. For this to take place, the valvetrain and surrounding engine components must be as rigid as possible. The engine's cylinders and heads also must be as rigid as possible for the valvetrain to work to its potential. Chrome-moly cylinder base studs help keep the Evolution's top-end rigid and heavy cylinder base flanges do the same for the Shovelhead. A heavy duty top motor mount also helps.

It is better to have a heavy and rigid valvetrain than a light and flexible one. Chrome-moly pushrods are much more rigid than aluminum, although much heavier. Aluminum pushrods are fine for lower rpm use, but chrome-moly should be used for serious racing.

Although keeping weight to a minimum is always important, it is more important to save weight on the valve side of the rocker arm than the pushrod side. This means light weight valves and spring collars help maintain accurate valve timing and increase the usable rpm limit more than light weight parts on the pushrod side of the rocker arm. Remember that light weight parts are acceptable as long as they do not flex and can stand up to high load usage.



Kits are available that not only include a cam, but also all necessary valvetrain components for easy cam installation. The four small washers at the bottom ends of the pushrods convert Evolution hydraulic lifters to solid action when the engine is at high rpm. Shown is an S&S Evolution cam kit with titanium spring collars and Hydraulic Lifter Limited Travel kit. Photo courtesy of S&S Cycle.

Don't run more valve spring pressure than is necessary for the engine combination and application. However, it is better to have too much spring pressure than too little. For long valve spring life, keep the springs as cool as possible and on a race engine check spring pressure often. Reduce friction as much as possible because friction eats up horsepower and increases heat.

Rocker geometry is critical for maximum performance. Make sure all valve stem heights are equal and set at the proper height. Also, the rocker arm's pad that contacts the valve stem must have the correct radius for maximum valve action and minimum side loading. Use the correct length pushrods. Long cylinders usually require extra long pushrods.

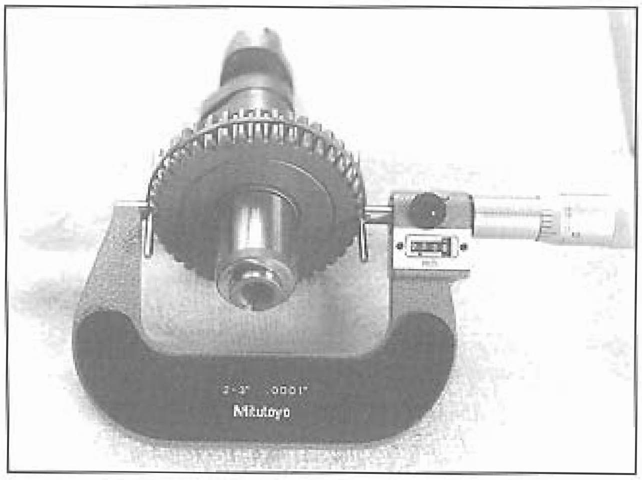
Be sure to check the valve-to-valve, valve-to-piston, rocker arm-to-spring collar and rocker arm-to-rocker cover clearances. Also make sure nothing interferes with the pushrod to cause side loading.

CAM INSTALLATION

The amount of work needed to install a Big Twin cam depends on factors such as: whether the engine is removed from the frame, year and model of bike, and the camshaft. Refer to the factory service manual and the camshaft's installation instructions for step-by-step procedures. The following information covers factors that should be considered for maximum performance and ease of installation.

Cam installation is easiest during an engine rebuild or when the engine is out of the frame. In this case, installation requires spring spacing, piston-to-valve and valve-to-valve clearance checking, besides checking gear clearances and end-plays. For maximum effort engines, installation also should include degreeing the cam and ensuring the valvetrain geometry is correct, although these procedures are beneficial to any engine.

When the engine remains in the bike, additional issues must be considered. A bolt-in cam means just that — it installs without requiring major modifications. However, some cams come closer to this definition than others. Depending on the application, some bolt-in cams require piston-to-valve clearance checked. Clearance must always be checked on late 1984 through early 1986 Evolution engines that do not have notched pistons. Any cam that requires valve spring spacing, different valve springs, piston notching or crankcase machining is not a true bolt-in.



A loose fitting cam drive gear has excessive backlash and causes a "clicking" sound. A tight fitting gear whines and can result in gear damage. The gear's size can be determined by laying two pins on the gear and measuring the distance over the pins. Different size gears are available.

Shovelhead engines normally do not require removal of the cylinder heads, rocker covers or gas tank unless spring spacing is required. On the other hand, Evolution engines usually require disassembly of the top rocker covers to remove and install the pushrods. This normally requires removal of the gas tank and on some models such as the FX, requires loosening the top and front motor mounts to allow the engine to drop down for sufficient rocker cover-to-frame clearance.

Numerous methods and various special tools have been tried to avoid lowering the engine to install a cam in an Evolution FX. However, most installations end up requiring the engine lowered. So, you might consider saving yourself time and frustration by planning from the onset to lower the engine. Additionally, the exhaust system and anything else blocking access to the cam gearcase cover must be removed. In most cases, you'll spend more time removing the gas tank and exhaust system along with lowering the engine than installing the cam.

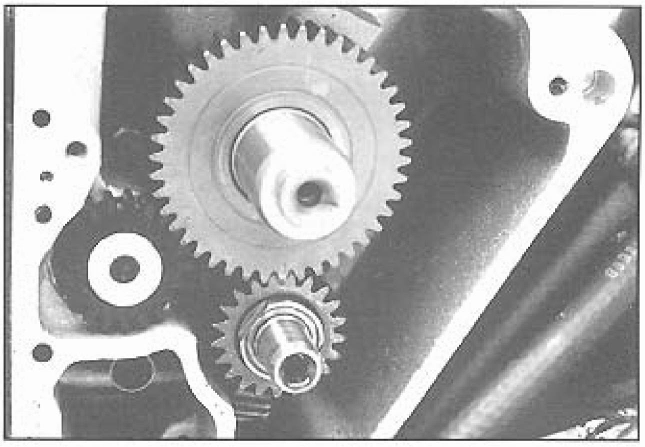
Once the old cam, lifters, lifter blocks and pushrods are removed from the engine, it's a wise idea to start the installation process by first checking the condition of a few key areas. First, inspect the area under the cam needle bearing in the right half crankcase and the bronze bushing in the cam cover for any sign of stress cracks. Look closely at the area between the crankcase cam needle bearing race and the pinion shaft race for cracks. Make sure the camshaft's fit in the bronze bushing is correct. Slop here ruins valvetrain geometry and timing. Check the needle bearing for worn or missing needles. For a free-running camshaft, it's a good idea to replace this bearing when the engine is apart. If the bronze bushing in the cam gearcase cover is replaced, be sure to align-ream it to proper size for a free-running camshaft.

The camshaft includes a drive gear that is driven by a pinion gear. The pinion gear is mounted on the flywheel's pinion shaft. A poor fit between the two gears will cause either excessive noise or excessive wear. Excessive clearance (greater than .002-inch backlash) will cause a clicking sound that may be annoying, but it won't seriously damage anything. However, it will give unpredictable cam timing. On the other hand, tight fitting gears will cause a noticeable gear whine and can result in disintegration of the gear teeth and camshaft bearings. This circulates metal debris through the oil system and leads to possible destruction of engine bearing surfaces.

The factory makes several different sizes of color-coded cam drive gears and pinion gears for the correct fit in each engine. Also, performance cam companies make different size drive and pinion gears. Take note that the pinion gear shaft nut has a left hand thread and the gear requires a special puller for removal. The Service Department Company can supply the puller.

To retain the proper gear fit when a new cam is installed, the new drive gear should be the same size as the original gear. The correct fit can be accomplished in either of two ways. First, by noting the color-coding on the original drive gear, you can order the new cam with the correct size gear. Second, the drive gear from the original cam can be removed and installed on the new cam. However, you must make sure it is properly positioned on the camshaft for correct timing.

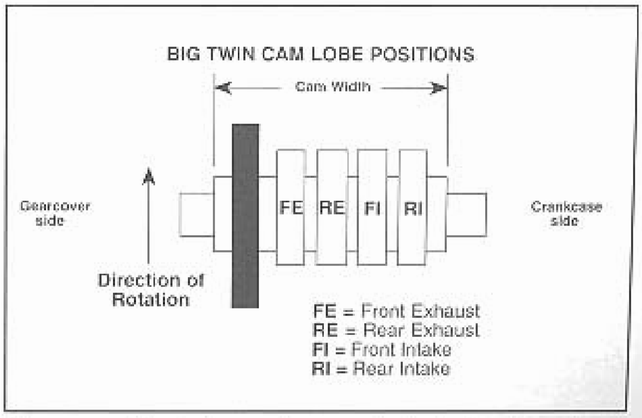
The exact size of the cam and pinion gears can be determined by laying either two .108 or .105-inch diameter pins on the gear's teeth at a 180 degree angle to one another and then measuring their outside diameter with a micrometer. Use the .108-inch diameter pins for 1990 and later engines. The 1990 and later cams are identified by two groves on the drive gear's face. The measured diameter of the cam gear then can be matched to the color-coded part numbers in the factory service manual. Take note that few engines are shipped with larger size drive gears. If you're attempting to correct a whining (tight fit) or clicking (loose fit) cam gear set, start by selecting a cam drive gear either two sizes smaller or larger than the problem gear.



The small pinion gear (bottom) mounts on the flywheel's right side main shaft and drives the large cam drive gear. Various size pinion gears are available to help obtain the proper cam gear clearance. The cam drive gear drives the breather valve. Make sure the timing marks on all three gears align before installing the cam cover. It is easiest to install the lifter assemblies after the cam and cover.

Another thing to check is the clearance between the cam's rear intake lobe and the crankcase area near the flywheel's pinion shaft. To check this clearance, place the cam along with its lock and thrust washers into the crankcase needle bearing and make sure the cam can freely rotate all 360 degrees.

For 1969 and earlier Shovelhead engines, install the camshaft and ignition drive gear into the crankcase. Then rotate the flywheels and check the clearance between the cam's front exhaust lobe (the lobe closest to the drive gear) and the ignition's drive gear. If there is less than .060-inch clearance, the ignition drive gear must be machined.

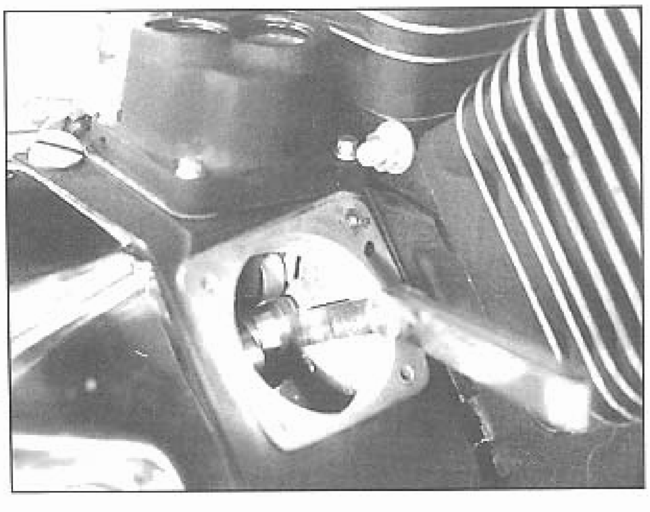


The cam rotates in the engine counterclockwise when viewed from the right side or carburetor side of the engine. The rear intake lobe is positioned next to the right crankcase half.

Camshaft side clearance also must be checked. Place the cam along with its lock and thrust washer into the crankcase needle bearing, install a cover gasket and then torque down the gearcase cover to factory specifications. Check the cam's side-play with a feeler gauge through the lifter block holes. Another method for checking the side clearance is to measure the width of the new and old camshafts with a micrometer and then compare any difference. Spacer washers with different thicknesses are available to obtain the proper clearance. Set the cam's end clearance at .010 to .015-inch for an Evolution engine and .005 to .010-inch for a Shovelhead. The engine will be functional with greater clearance, but for maximum performance you're better off minimizing all clearances in the valvetrain.

Make sure each lifter fits properly in its lifter block and that its roller turns free without binding. The rollers should not have excessive side play or very loose fitting needle bearings. Replacement rollers are available. Also, the surfaces of the lifter body and its bore in the lifter block should not be scored. Remember, that a sloppy fit will ruin valve timing and kill horsepower.

The oil hole in an Evolution's hydraulic lifter body can face in either direction. However, for a Shovelhead make sure the hole faces toward the center of the lifter block.



Check the cam's side clearance by installing the cam and bolting down the cam cover using an old compressed gasket. With the lifter assemblies removed, use a feeler gauge to check the cam's end clearance. Set the clearance for the Shovelhead engine at .005-inch and the Evolution at .010-inch. Different size shims are available.

Most high lift cams require valve spring spacing. Some require replacement springs. Be sure to follow the spring manufacturer's recommendations for spring travel and installed spring height. Depending on the cam, spring and application, spring travel must be .030 to .060-inch greater than the cam's valve lift. This will protect against spring coil bind since there will be some clearance between the spring's coils when the valve is at maximum lift.

The installed spring height determines the amount of spring travel. Installed height is the distance between the spring's bottom and top collars. Installed height can be calculated by adding the compressed spring height to the maximum lift of the cam and then adding the recommended clearance. The sum of these three values becomes the spring's installed height.

Also, make sure there is sufficient clearance between the underside of the top spring collar and the valve guide or valve guide seal. This dimension should be at least the sum of maximum valve lift plus .050-inch for clearance. The valve guide may need to be shortened to obtain the proper clearance. However, don't shorten the guide any more than is necessary.

All Evolution engines have valve guide seals and starting in late 1981 all Shovelheads were shipped with guide seals. Oil in the combustion chamber brings on detonation quickly, so for maximum valve guide oil control use a top quality Teflon type seal. Be careful not to nick the seal during valve installation. When performing a valve job on early Shovelhead engines (early 1981 and earlier), you should retrofit Teflon oil seals to the valve guides.

Valve-to-piston clearance must be checked when installing certain cams. Place clay on the piston's valve relief areas, lightly coat the valve heads with oil and torque the heads down to proper specifications. Don't forget to use the correct head gasket. Make sure the pushrods are correctly adjusted and then very gently turn the engine over two revolutions. Do not force the engine because this could bend a valve. There should be a minimum of .060-inch clearance between the valve and piston, although .080-inch is preferred (Consult with your cam manufacturer for the recommended minimum clearance).

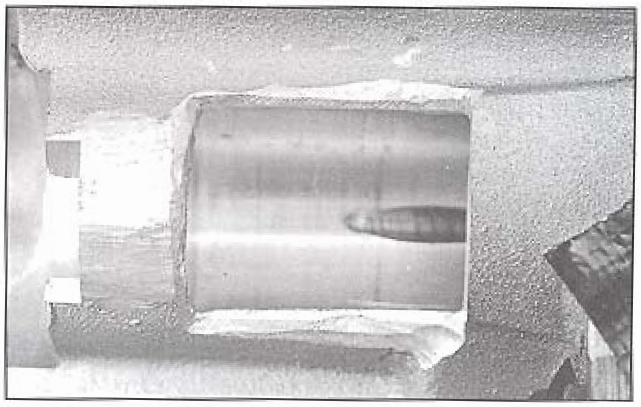
Whenever new valves, valve seats or a performance cam is installed, valve-to-valve clearance must be checked. Under some conditions the intake and exhaust valves may contact during the valve overlap period. The clearance can be checked in either of two ways. The first method requires checking the clearance between the valves when the piston is located between 5 degrees BTDC and 5 degrees ATDC. Insert a thin .050-inch diameter wire through the spark plug hole and make sure it can pass between the valves when the piston is located as described above. View the wire either through the plug hole or one of the ports.

An easier method is to measure the distance between the two valves when they are resting on their seats. A general rule is to add .050-inch to the valve's lift at TDC for minimum valve-to-valve distance. However, keep in mind that this rule may not be valid for all cams. Since valve timing and lift rates vary between cams, consult with the cam manufacturer for the proper clearance. Valve modifications are necessary if the clearance is insufficient. Sinking a valve deeper into its seat may shroud airflow and reduce power. Blending the combustion chamber in the area of the valve seat may help, but compression will be reduced.

CRANKCASE SCAVENGING

If you're doing a total engine rebuild and the engine is completely apart, there are additional modifications you can make to the cam timing chest and gearcase cover. First, inspect the area near the cam needle bearing, which is located in the right half crankcase. Look closely between the needle bearing race and the pinion shaft race for cracks. Check the needle bearing for worn or missing needles. For a free-running camshaft, it's a good idea to replace this bearing when the engine is apart. Also, check the bronze bushing in the cam gear cover for any sign of stress cracks. Make sure the camshaft turns smoothly in the bushing and that it has the correct clearance. Slop here ruins valvetrain geometry and timing. When installing a new bushing, be sure to align-ream it for a true axis.

Much horsepower is lost due to churning oil so unneeded oil must be directed to the pump for



To relieve crankcase pressure and reduce the potential for engine oiling, the breather cavity must be modified to proper specifications. This is a time consuming operation, but well worth the effort. Photo courtesy of S&S Cycle.

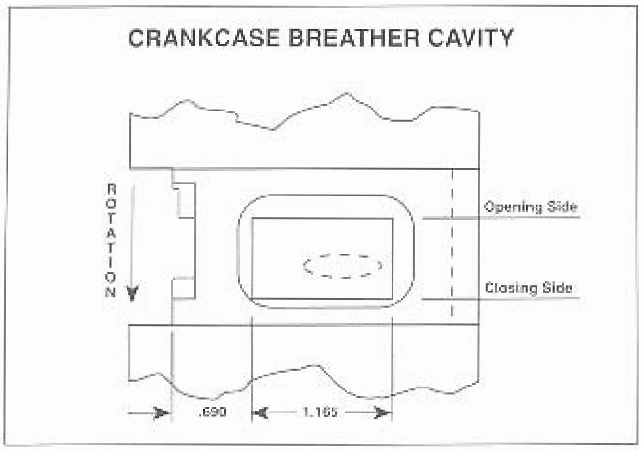
scavenging as quickly as possible. Polishing the floor of the timing chest with a cartridge roll helps speed the flow of oil to the pump for pick up.

Oil pump breather timing is another source of potential horsepower gains. Horsepower loss due to oil drag on the flywheels is high and quickly scavenging the oil from the crankcase cavity is important for minimizing drag.

At the rear side of the cam timing chest there is a breather opening roughly cast into the crankcase. Below the opening, a rotary breather valve revolves inside the breather cavity. The opening and closing timing of the valve relative to the flywheels' position allows oil to be scavenged from the crankcase and releases power robbing crankcase pressure at the optimum time.

Early model (mid 1970s and earlier) crankcases have an elliptical shaped breather opening while later models come with a rectangle shaped opening. The rear edge of the opening determines when the breather opens and the front edge when it closes. Its total area determines the volume of oil and amount of pressure it can relieve in a given time period. For maximum performance, the size of the opening and the position of its opening and closing edges must be modified for maximum performance.

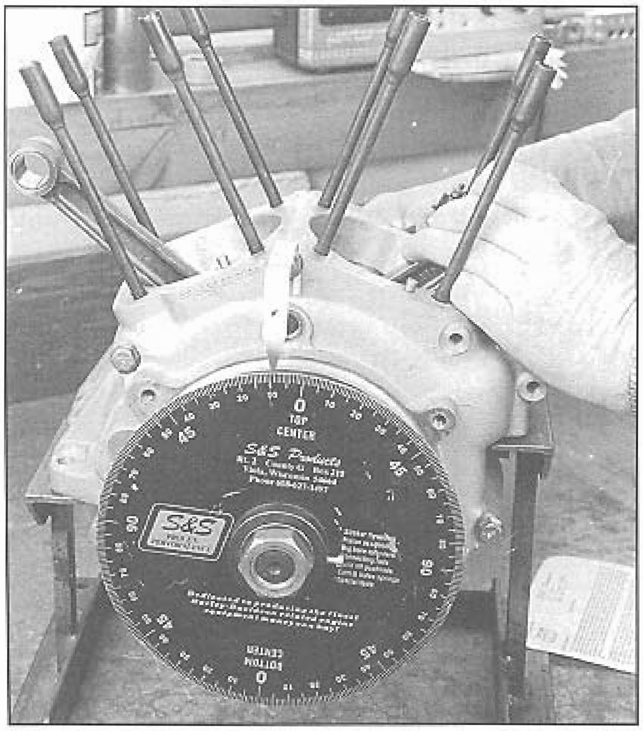
The first objective when modifying the breather is to ensure the width of the opening in the crankcase cavity is equal to the one in the rotary breather. The opening should measure 1.165-inches in length (length runs parallel to the camshaft). The breather should be modified to open between 10 degrees



To maximize crankcase oil scavenging, the breather valve can be timed to open and close at the optimum crankshaft position. The elliptical hole in the crankcase must be modified to achieve optimum timing besides increased volume. Illustration courtesy of S&S Cycle.

BTDC and 10 degrees ATDC and close between 55 and 75 degrees ABDC at the *front* cylinder.

The breather timing procedure requires installation of a degree wheel on the flywheel sprocket shaft and identifying TDC for the *front* cylinder's piston. Use a .002-inch shim (feeler gauge) in the breather opening to identify the



Degreeing the breather valve requires mounting a degree wheel on the flywheel's sprocket shaft, locating TDC for the front piston and checking when the breather valve opens and closes. Modify the breather cavity in small increments and double check your work. Photo courtesy of S&S Cycle.

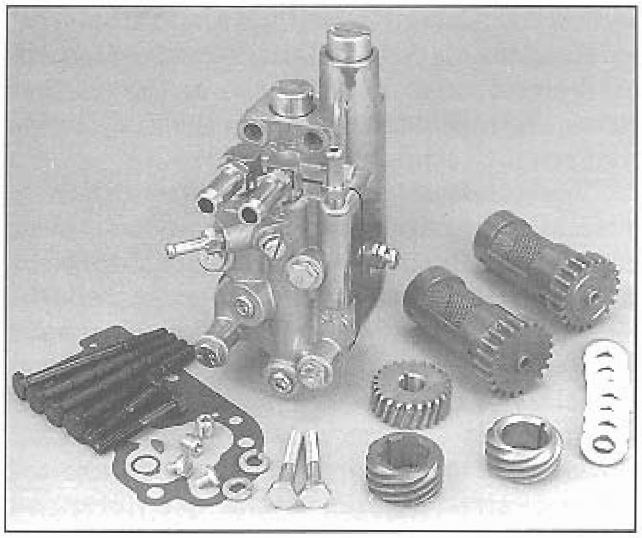
exact opening and closing point. Grind the opening and closing edges of the breather opening so the timing falls within preferred limits. Be careful because excessive removal of material from the opening may seriously inhibit the breather's operation and cause loss of engine oil.

Custom Chrome makes replacement steel breather valves that increase the total time the breather is open without requiring grinding of the crankcase breather opening. The valves are available in three different window screen sizes and easily allow improvement in crankcase scavenging when an engine is not completely disassembled. For maximum efficiency, however, the breather opening should also be enlarged for increased flow area.

Starting in 1983, the breather valve material was changed from steel to plastic. Plastic valves are abrasive and can score the breather cavity sealing surface. Also, the plastic is relatively weak and bends out-of-round during operation. Both conditions reduce the breather's sealing capability, which reduces crankcase scavenging. This can cause engine smoking and excessive oil loss.

An additional procedure for maximizing breather efficiency is to modify or remove the screen in the breather valve. However, keep in mind the screen is designed to protect the engine by stopping foreign material from passing through the oil pump and lubrication system. Consequently, this modification is done at your own risk. As an alternative, some racers remove the clip that holds the breather screen (early models only) and then spot weld the screen in place. Additionally, all holes in the screen can be drilled larger for increased flow.

It's important not only to scavenge oil quickly from the crankcase, but also to limit the amount of oil in the crankcase. After a hard drag run, the crankcase should only have about four to six ounces of oil in it. To check this, install a drain plug on the bottom of the crankcase so the oil can be quickly drained. Any more than six ounces in the crankcase means you should reduce the volume flowing to the flywheel assembly and/or top-end by installing an oil restrictor. Oil restrictors can be placed in the end of the



Most stock oil pumps can be improved upon. Shown is an S&S oil pump kit including steel breather valves. Steel valves retain their shape and increase crankcase scavenging. Note the perforated screen in the breather valve. This kit increases crankcase scavenging and improves the operation of hydraulic lifters. Photo courtesy of S&S Cycle.

flywheel's pinion shaft and the external oil line leading to a Shovelhead's top-end. Also, the top-end oil return lines can be rerouted to dump into the cam gearcase area instead of the crankcase. For the race track, consider running a dry top-end.

To relieve crankcase pressure, the gearcase must be properly vented to the atmosphere. Use at least a 1/2-inch I.D. hose line with an XR-750 breather separator mounted in-line or a K&N vent filter mounted at the line's open end.

VALVE ADJUSTMENT (SOLIDS)

Solid lifters require adjustment to ensure the valves completely close and for maximum power. They are initially adjusted during engine assembly and then readjusted after engine break-in and about every 2000 miles thereafter.

The clearance in the valvetrain is usually referred to as valve lash. The clearance is measured when the engine is cold and the lifter is resting on the backside or heel of the cam lobe. Some valve lash is required in the valvetrain to ensure that the valve can fully close. The recommended amount of valve lash is determined by the cam, the expansion characteristics of the valvetrain and other engine components — most notably the cylinders and heads.

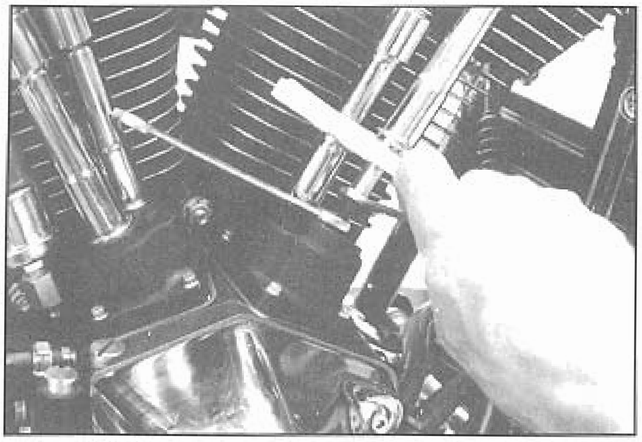
For normal operation, solid lifters are adjusted so the valvetrain has the least possible clearance when the engine is cold. As the engine warms and parts expand, the clearance can easily increase to .040-inch or more. Check with your cam manufacturer for the recommended clearance for your cam and pushrod type (aluminum or chrome-moly).

The engine must be cold when valve lash is set. Start by removing the spark plugs and the pushrod cover keepers. Remember to make sure the valve is fully closed because the clearance must be set when the lifter is positioned on the heel of the cam. The easiest way to ensure the lifter is properly positioned on the cam is to turn the engine forward until the same lifter for the other cylinder is at its highest position. Use a rubberband that has a paper clip on each end to hold the pushrod cover in the raised position when viewing the lifter. The intake valves are closest to the carburetor. If the transmission does not have a kick starter, turn the engine over by placing the transmission in high gear, raise the bike's rear wheel off the ground and rotate the rear tire by hand.

If you're installing new pushrods, notice that the pushrods are different lengths. Depending on the manufacturer, there may be two, three or four different lengths. The longest one always goes to the front exhaust and the second longest to the rear exhaust. The shortest pushrod goes to the rear intake.

Loosen the lifter adjusting screw lock nut and then turn the adjusting screw until the pushrod is finger spin free without any trace of binding. Next, tighten the lock nut and again check the pushrod for spin free motion. When done, rotate the engine so the lifter that was just adjusted is at its highest position. Now adjust the same pushrod in the other cylinder. Follow this procedure for all four pushrods. If any pushrod is adjusted too tight, you risk burning a valve because the valve will not completely close. Turning the adjusting screw a distance of one flat changes valve clearance about .006-inch.

The valve lash setting affects the lift and timing characteristics of the cam. Therefore, valve lash can be used as a tuning aid by changing the breathing characteristics of the engine. Breathing will determine how the engine develops cylinder pressure and ultimately its power



Use a paper clip attached to a rubberband or a clothes pin to hold the pushrod cover out of the way. Adjust the lifter when it is resting at its lowest point (on the cam's base circle). A lifter is at its lowest position when the like lifter for the opposite cylinder is at its highest position. Take note of the 12-point lifter block bolts. Lifter blocks have a tendency to work loose, causing considerable engine damage. Use Loctite on the bolt threads and periodically check them for tightness. Also, use a small amount of sealer under the bolt's head to eliminate oil seeping out of the bolt hole.

output. Increasing or decreasing the valve lash for the intake valves, exhaust valves, or both, can help maximize the performance of your camshaft. It can also help you determine whether the engine could benefit from different valve timing.

Increasing valve lash causes the valve to open later and close earlier. This reduces the valve's duration and the valve's total lift by the amount of additional clearance multiplied by the rocker arm ratio. However, it does not change the point at which maximum valve lift occurs. Reducing valve lash reverses the process. The valve opens earlier, closes later and total valve lift is increased by the amount the valve clearance is reduced multiplied by the rocker arm ratio.

These changes not only alter the amount of cam duration and lift, but also valve overlap. Together, these changes affect the way the engine breathes, which affects cylinder pressure. Consequently, the engine's performance characteristics are affected.

Altering valve lash settings has different effects on the engine than advancing or retarding the cam. When the cam is advanced, both the intake and exhaust valves will open and close earlier and the point of maximum lift will be reached earlier. Remember, altering valve lash keeps maximum valve lift at its normal point while only changing the opening and closing points of the valve. Additionally, valve lash can be changed only on the intake or the exhaust valves. Advancing or retarding the cam always changes the setting for both valves.

Valve lash adjustment can be used to help maximize the engine's performance for a given day and specific track conditions. It also can be used to help determine whether the engine might perform better with more or less cam. Since solid lifter valve lash for the Big Twin is set to about zero clearance when the engine is cold, you can not reduce it without risking a burned valve. However, by increasing it on both the intake and exhaust valves, low rpm power will be increased because valve duration and overlap are decreased. This can be helpful for tracks having high traction, short eight mile tracks, heavy bikes or bikes running high gearing (low numerically). Conversely, when valve lash is reduced, duration and overlap increase, which results in improved high rpm breathing and power. For a given engine combination, reducing valve lash can help on slick tracks, long tracks, where the tire's traction is too low for the engine's horsepower and chassis setup, and transmissions with too low (high numerically) of gearing.

Changing valve lash can also indicate whether the engine will perform better with a different cam. For example, if the engine always performs best with increased valve lash, less duration should be beneficial. Additionally, by changing only the intake or exhaust valve clearances, you can determine whether the engine benefits from a breathing change on only one side. Start making adjustments by increasing the valve lash in .006-inch increments up to about .018-inch (3 adjusting screw flats) and record the results for future reference.

Keep in mind that changing valve lash settings for tuning purposes is not possible with hydraulic lifters since they do not operate with any valvetrain clearance. However, with solid lifters valve lash changes can help dial in the engine and chassis combination to the track conditions and help determine the optimum cam.

VALVE ADJUSTMENT (HYDRAULICS)

For a completely stock Evolution valvetrain, the adjustment of hydraulic lifters is very simple because there is really no adjustment required. However, when a performance cam is installed in an Evolution, the adjustment is more involved than adjusting solids. In the case of a Shovelhead engine, the same principles apply.

Lets start by discussing the adjustment of hydraulic lifters for a bone stock Evolution. When the cam is removed from the engine, hydraulic lifter adjustment amounts to installing the camshaft, lifters and lifter blocks, dropping in the one piece pushrods and bolting down the rocker arms. If we assume that all valve stem heights are correct, the above procedures will result in the correct amount of hydraulic preload and an accurately adjusted valvetrain. However, when a performance cam is installed, its base circle is frequently a different diameter than the stock cam's and adjustable length pushrods are normally used. This changes the relationship of the components, which requires new procedures for accurate adjustment.

Two terms are frequently mentioned when working with hydraulic lifters: preload and bleed-down. To adjust lifters correctly, both terms must be understood. With the Evolution, the plunger in the hydraulic lifter has a maximum travel of close to 1/4-inch. A spring keeps the plunger at its highest position. For the lifter to work properly, the length of the pushrod must be adjusted so that the plunger is partially depressed. Then, when the engine is running, oil pressure to the lifter will keep valvetrain clearance at zero. The distance the pushrod extends into the lifter and depresses the lifter plunger is the amount of preload. The question now becomes: What is the correct amount of preload? If there is not enough preload, the lifter will be noisy and make a ticking sound. Additionally, maximum valve lift can be reduced .050-inch or more. If there is too much preload, the valve will be kept off its seat and this will result in a burned valve and lost performance.

Bleed-down is the time where oil trapped inside the lifter body bleeds out of the lifter and allows the hydraulic plunger to rest at its normalized position. Bleed-down can take one to two minutes or more. An important point to remember when adjusting hydraulic lifters is never to turn the engine over when the lifter is bleeding-down because you could bend a valve.

When adjustable pushrods are used, at least three methods exist for adjusting hydraulic lifters. These methods are very similar to those used to adjust automotive hydraulic lifters. The first method is the least accurate, but it is the quickest and easiest of the three methods.

For the first method, start by adjusting the pushrods to their shortest length and then install each one into its proper valve position according to its length. Remember, the longest pushrod goes to the front exhaust. Adjust each pushrod until all free play is removed. Next lengthen the pushrod by turning the adjusting screw 4 to 5-turns (24 to 30 flats) and tighten the lock nut. Wait a couple of minutes to make sure the lifter has bled-down, then repeat the procedure for the next pushrod.

From a performance standpoint, the second method is more accurate than the first, but it takes longer to do. This method requires removal of the spark plugs and the engine turned forward as described under "Valve Adjustment (Solids)." For accuracy, the valve must be fully closed and its lifter positioned on the heel of the cam. Start by rotating the engine forward until a pushrod is at its highest position, then adjust the same pushrod for the other cylinder. The intake valves are closest to the carburetor. Use the adjustment screw to lengthen the pushrod until there is sufficient pressure so the pushrod cannot be turned with the thumb and index finger. Wait at least two minutes for the lifter to bleed-down, then lengthen the pushrod again until it will not turn. Again let the lifter bleed-down. Keep repeating the tightening and bleed-down procedure until the lifter will not bleed-down anymore. The lifter is now bottomed out. Be careful never to turn over the engine with a lifter bottomed out. Now shorten the pushrod by reversing the adjustment screw one half to one turn and tighten the adjusting screw lock nut. When finished, you should be able to spin the pushrod easily with your fingers. If it does not spin free, the lifter has too much preload and you must repeat the adjustment steps again.

When done, rotate the engine so the lifter that was just adjusted is at its highest position. Now adjust its like lifter in the other cylinder. Follow this procedure for all four lifters. When all lifters are adjusted, turn the engine a few additional revolutions and check each lifter again. First, make sure the lifter is on the heel of the cam and its valve is fully closed, then check that the pushrod will spin free. If any lifter is adjusted too tight, you risk burning a valve because the valve will not completely close.

The third adjustment method is the most accurate, but access to the valve spring top collar is required. Therefore, this method is normally only used for the Evolution. Start by removing the rocker covers and mount a dial indicator on the valve spring collar of a closed valve. Turn the engine until the like valve for the other cylinder is at its highest position. Adjust the pushrod until all free play is removed. Next, lengthen the pushrod by turning the adjusting screw 4 to 4-1/2 turns (24 to 27 flats) and tighten the lock nut. Wait a couple of minutes for the lifter to bleed-down, then turn the engine forward and read the maximum lift registered on the dial gauge. The lift should be within .010-inch of the camshaft specifications. If the reading is within .010-inch, turn the engine over until the valve is completely closed and check that the pushrod spins free. If it spins free, the lifter is correctly adjusted so go on to the like valve for the other cylinder.

However, if the reading is more than .010-inch less than maximum valve lift, fully close the valve and re-adjust the dial indicator to zero. Lengthen the pushrod a small amount and wait for the lifter to bleed-down. The indicator will move off zero, then return to zero after the lifter bleeds-down. Turn the engine forward until maximum lift is achieved and read the dial gauge. Repeat these procedures until the dial gauge reads within .010-inch of the cam's specified maximum valve lift or the lifter bottoms out.

When a lifter bottoms out it cannot bleed-down anymore and the dial gauge will not return to zero. In this case, shorten the pushrod by reversing the lifter adjusting screw one half to one turn and make sure the pushrod spins free when the valve is fully closed. If it doesn't, keep reversing the lifter adjusting screw in 1/2 turn increments until the lifter spins free with the valve closed. Be sure to wait at least two minutes between each adjustment.

Follow these procedures for all four lifters, then gently rotate the engine a few times and check each pushrod one more time with its valve closed to ensure the pushrod spins free.

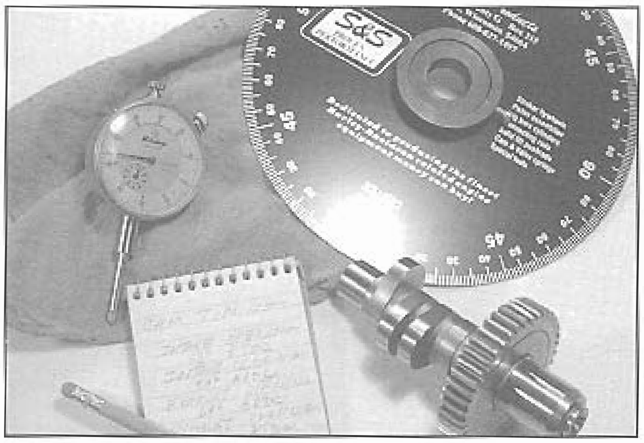
CAM DEGREEING

Difference in cam timing is a major reason why stock production engines vary in horsepower output. When a cam is installed there is no guarantee the engine's valve timing matches the cam's specification sheet. Due to varying machined tolerances between the flywheels, camshaft and valvetrain, it is difficult to achieve accurate valve timing. However, degreeing a cam will verify the engine's valve timing. Then, if the timing is not accurate, changes can be made. For maximum effort engines, degreeing the cam is necessary to achieve maximum power. For mild performance engines degreeing is not necessary, but it definitely doesn't hurt.

When cam timing differs from specifications, the normal procedure is to remove the camshaft drive gear using a small hydraulic press and replace the gear on the shaft at a slightly different position. This assumes that the rest of the valvetrain geometry is not being altered. Even if you have no intention of changing the cam's timing, degreeing the cam will at least tell you that your engine combination performed at a given level with a given cam timing. If you're a serious racer, this information can be valuable in the future when selecting a different cam or tuning the engine.

Since all four cam lobes are on one shaft, the lobe center angle for a Big Twin cam cannot be changed unless the cam is reground to different specifications. This means you cannot use LSA when degreeing a cam. Since the most important valve timing event is intake closing, this event should be used when degreeing the cam.

Currently, all Big Twin cam specifications are taken at either .020 or .053-inch of *cam* lift. It is best to degree the cam at the .053-inch timing specification unless it is not supplied. Note that cam timing specifications are given for cam lift, not valve lift. This means you must measure the lift at the cam lifter, not the valve. If you want to take a reading at the valve, you need to multiply either .020 or .053-inch by the rocker arm ratio to arrive at an equivalent valve lift. As an example, .053 multiplied by 1.6 gives .085-inch valve lift. Now, readings can be taken at .085-inch valve lift instead of .053-inch lifter lift. However, keep in mind that valvetrain deflection will cause inaccurate readings at the valve. One solution is to use low tension valve springs during checking procedures.



Cam timing is one major reason similar engines differ in power. There is no guarantee that when a cam is installed its timing reflects the timing specs. The only way to know for sure is to degree the cam and then make any necessary adjustments. This is a requirement for maximum effort engines and it is often the difference between winning and losing.

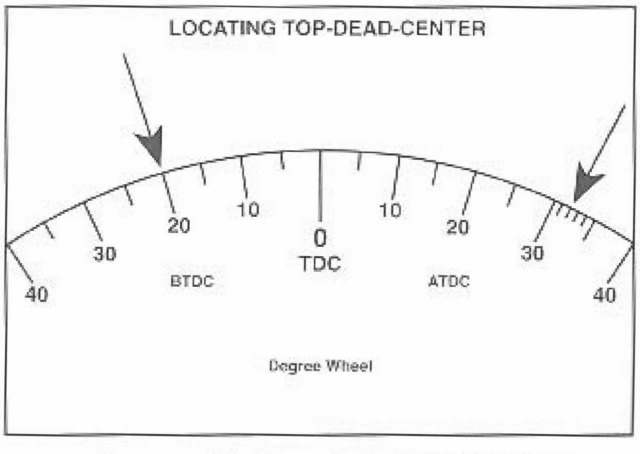
Special springs are made specifically for this purpose.

To degree a cam, a few special tools are needed besides the tools required to remove and install the cam. First, you need a large diameter degree wheel and an adapter to mount it to the flywheel sprocket shaft (located on the left side of the engine). Crane Cams and S&S Products both make degree wheel kits. These kits are also useful for checking crankcase breather timing and ignition timing. Additionally, you need a pointer to show the degree wheel's position and a TDC locator. There are a number of methods for locating TDC, but a shaft that screws into the spark plug hole eliminates the need to remove the cylinder heads. Also needed is a dial indicator with a total travel of no less than cam lift or valve lift (depending on where you are checking) and a clamp set to hold the dial gauge rigid.

Start by mounting the degree wheel on the flywheel's sprocket shaft. This requires removing the primary chain, sprocket and clutch if the engine is in the bike's frame. The pointer should be mounted to the center crankcase through bolt or some other suitable location.

Next, locate TDC. It can be located in either of two ways: by using a TDC locating tool inserted into the spark plug hole (the positive piston stop method) or by mounting a dial gauge on the piston dome. The best way to determine TDC with the positive piston stop method is to define two flywheel positions that are an equal distance from TDC — one position is BTDC while the other is ATDC. TDC will be centered between these two positions. Start by rotating the flywheels forward to bring the front cylinder piston about one half inch from TDC. Screw the TDC locator into the spark plug hole until it gently touches the piston and set the degree wheel pointer to 20 degrees. Next, gently rotate the engine almost a full turn backwards until the piston touches the TDC locator and note the degree wheel's position. TDC is located halfway between the two degree wheel positions.

For example, we know that the degree wheel registered 20 degrees when the piston stopped at BTDC. Lets assume the degree wheel was positioned at 32 degrees for the ATDC reading. By subtracting the smaller number from the larger number (32-20=12) we get a remainder of 12 degrees. Now, if we divide 12 in half (12/2=6) we get 6 degrees. Next, move the degree wheel 6 degrees in the direction of the smaller number and repeat the checking procedures. This time both degree wheel readings should be 26 degrees. When both readings are identical, the degree wheel is correctly positioned on the flywheel sprocket shaft.



The positive stop method is one technique for determining top-dead-center (TDC). After installing the TDC locating tool (piston stop) in the spark plug hole, slowly rotate the flywheels forward until the front cylinder piston is about 1/2-inch below TDC. Adjust the piston stop until it gently touches the piston and set the degree wheel pointer to 20 degrees before-top-dead-center (BTDC). Now slowly rotate the flywheels backwards until the piston again touches the piston stop. In this example we read 32 degrees after-top-dead-center (ATDC). Now subtract the smaller number from the larger number and divide the remainder by 2 (32-20=12/2=6). Next, move the degree wheel 6 degrees in the direction of the smaller number and repeat the checking procedures. Accurately locating TDC is crucial to the accuracy of all subsequent cam timing procedures.

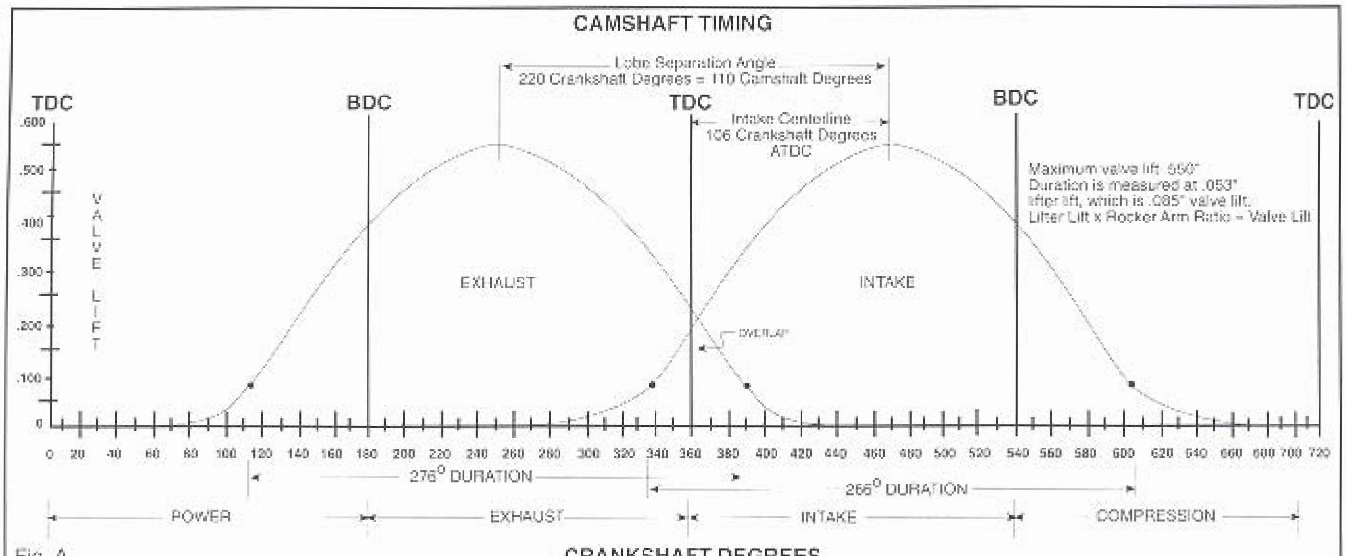


Fig. A

A cam can be "degreed in" and installed in either the advanced, straight up or retarded position. The cam's intake centerline is the point of maximum lift of the intake lobe in relation to the piston position from TDC. Advancing the cam moves the intake lobe's point of maximum lift closer to the piston's TDC position. This example uses a cam with 266 degrees intake duration, 276 degrees exhaust duration, 110 degrees lobe separation angle and .550" lift. Figure A shows the cam's intake centerline installed at 106 degrees ATDC. This means the cam is installed 4 degrees advanced since the position of the intake centerline is 4 degrees less than the cam's 110 LSA. Figure B shows the same cam, but it is installed 4 degrees more advanced because the intake centerline is now positioned only 102 degrees ATDC. Notice how all cam timing events now take place 4 degrees earlier. By advancing the intake opening and closing events, cylinder pressure is increased at low rpm and bottom end power is improved. The exhaust valve also opens earlier, which allows combustion blowdown to have a greater effect in purging exhaust gases. If the exhaust valve opens too early, combustion pressure is wasted out the exhaust and engine efficiency is reduced. For the Big Twin engine, it is recommended to use the intake valve closing timing when degreeing the cam. The intake valve closing can be advanced or retarded up to 4 degrees. If the cam requires more than a 4 degree change for acceptable performance, a cam with different specifications should be installed.

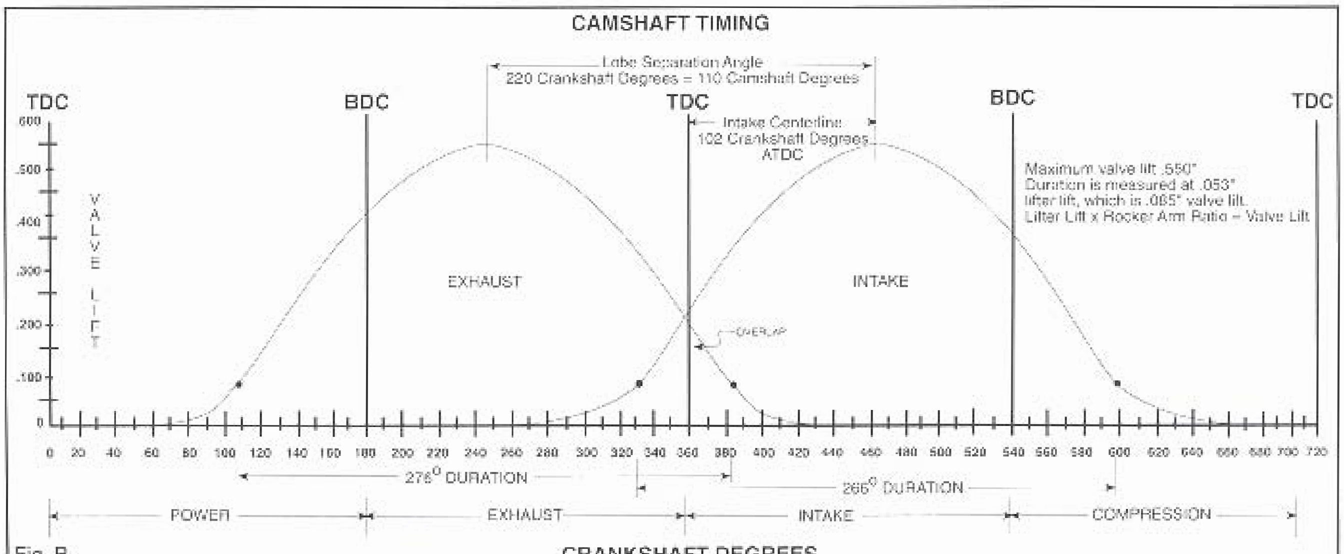
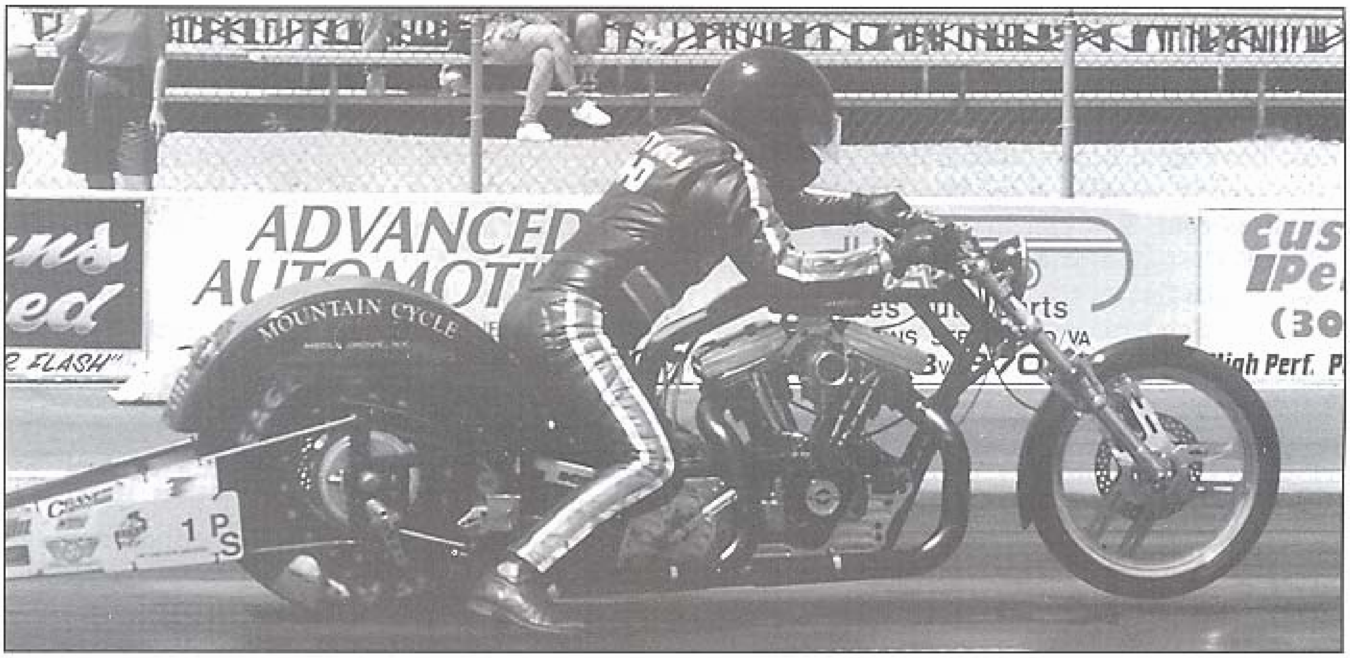


Fig. B

Follow the same procedures when locating TDC with a dial gauge.

The dial indicator now needs to be positioned on the front cylinder intake lifter. Rotate the flywheels forward until the lifter is on the heel of the cam, which places it at its lowest position in the lifter block bore. Position the dial indicator axis in line with the lifter axis and zero the indicator. Remember, we want to check when the intake valve closes. Slowly rotate the flywheels forward until maximum lifter lift is reached, then keep turning so the lifter starts descending. Always



High rpm, light weight dragsters can make use of a radical cam profile because they're usually revving between 4500 and 7500 rpm. Street riders need to balance top-end power against low speed torque. Shown is Tony Mattioli on his Pro Stock Evo Big Twin. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

keep turning the engine forward and never backup because gear lash will cause an incorrect reading. Keep turning the flywheels slowly while watching the dial indicator and immediately stop when the indicator registers .053-inch (or .020-inch, depending on the value you are checking at). Next, check the reading on the degree wheel. The degree wheel reading now indicates the timing (in crankshaft degrees) at which the intake valve closes. You should repeat the procedures to verify the accuracy of the results.

For maximum effort engines, nothing short of "spot-on" timing is acceptable. However, in the case of a street engine the choice is yours, but plus or minus two degrees from the cam's specifications is generally acceptable. Some racers perform the same checking procedures on the rear cylinder intake valve and then compare the result to the front cylinder timing. If the two values differ, the difference can be split in half to even things out.

If you need to change the cam's timing, the drive gear must be pressed off the camshaft and then replaced at a slightly different position. Without a special jig, this may take repeated attempts because the gear sometimes slips from its starting position during the pressing process. In some cases the valvetrain geometry or crankcase tolerances must be changed to obtain the correct timing.

CAM TUNING

If you intend to experiment with different camshafts or want an idea as to which direction to go (meaning more or less cam) with a different cam, there are a number of factors to consider. However, before changing the cam, there are numerous tuning procedures that need to be sorted out.

First, you want to ensure you're getting the most from the cam in the engine. Therefore, make sure that the carburetion is closely dialed in and the ignition advance is relatively optimized for the engine combination. The transmission's gearing should allow the engine to rev to its maximum power band. If you know the engine should be revving 6500 rpm through the timing lights, but it's only pulling 5000 rpm, change the gearing before changing the cam so the engine will rev high enough for accurate testing.

The exhaust system is highly dependent on the cam's timing. Once the cam or heads are changed, the exhaust system must be retuned. So you should tune the exhaust system to the existing cam before making a cam change. If the carburetion, ignition, gearing and exhaust are fairly well optimized for the engine combination, it's time to work with cam timing.

Valve lash is the easiest cam tuning change to make at the race track. Of course, the engine must have solid lifters. Remember, solid lifter valve lash is set at a snug spin free when the engine is cold. You can hardly go any tighter without risking a burned valve — but you can go looser. Each flat on the adjuster screw is about .006-inch. Some racers increase valve lash three to six adjuster nut flats (for short time periods) to determine whether the engine responds to reduced duration. This change should hurt the top-end while helping the bottom-end. If it helps both ends, you're possibly running too much cam for the engine. For a slick track with little traction, you want to reduce bottom-end power while maximizing the top-end, so keep valve lash to a minimum. Remember, you can change only the intake or exhaust valve lash.

You can advance the cam for more low-end power or retard it for more top-end, but this change isn't quickly done. It requires removing the cam from the engine and repositioning its drive gear on the shaft. Some cams have keyways cut into the drive gear to simplify this procedure. Remember, never advance or retard a cam more than four degrees. If a four degree change doesn't improve performance sufficiently, select a different cam. Also, don't forget that valve-to-piston clearance changes when the cam is advanced or retarded so be sure to check it.

Rocker arm ratios are an effective method for tuning a cam to an engine or bike combination. However, Evolution rocker arms can be changed much easier than the Shovelheads. You can change only the intake or exhaust rocker arms to see if the engine responds to a change on only one side. Remember, increasing the rocker arm ratio increases the breathing characteristics of the engine.

Some racers have special cam grinds made for their engine combination. Once they identify a cam that works fairly well, they usually keep lift and duration the same, but change the lobe center angle to see if power increases.

These tests can tell you a lot about what cam profile your engine combination needs for maximum performance. However, remember the Big Twin engine is very sensitive to exhaust and cam combinations. So once you change the cam, you need to go back and retune the exhaust system. ❖

Chapter 6

Compression Ratio

Squeeze Play

An engine's compression ratio has a direct bearing on the amount of power it makes and the amount of fuel it uses.

For a given engine combination, the higher the compression ratio, the higher its cylinder pressure will be. Brake mean effective pressure (BMEP) is an engineering term that refers to the amount of cylinder pressure an engine has and it is cylinder pressure that controls the power output of a given engine displacement. BMEP can be increased by high flowing cylinder heads or an efficient camshaft. It also can be increased by raising the engine's mechanical compression ratio.

The role compression plays in producing power is significant if cylinder pressures can be controlled. The higher the engine's compression ratio, the greater the air/fuel mixture is compressed. Compressing the mixture raises its temperature and increases pressure. Greater pressure increases the mixtures thermal efficiency, which increases the horsepower generated from a given amount of fuel. Fuel economy also is improved because less fuel is required for a given amount of power.

As an engine's compression increases, the potential for detonation is increased. If cylinder pressure is not properly controlled during combustion, detonation can develop and may lead to serious engine failure. When running open headers (straight pipes), detonation is frequently not heard because the exhaust sound is too loud.

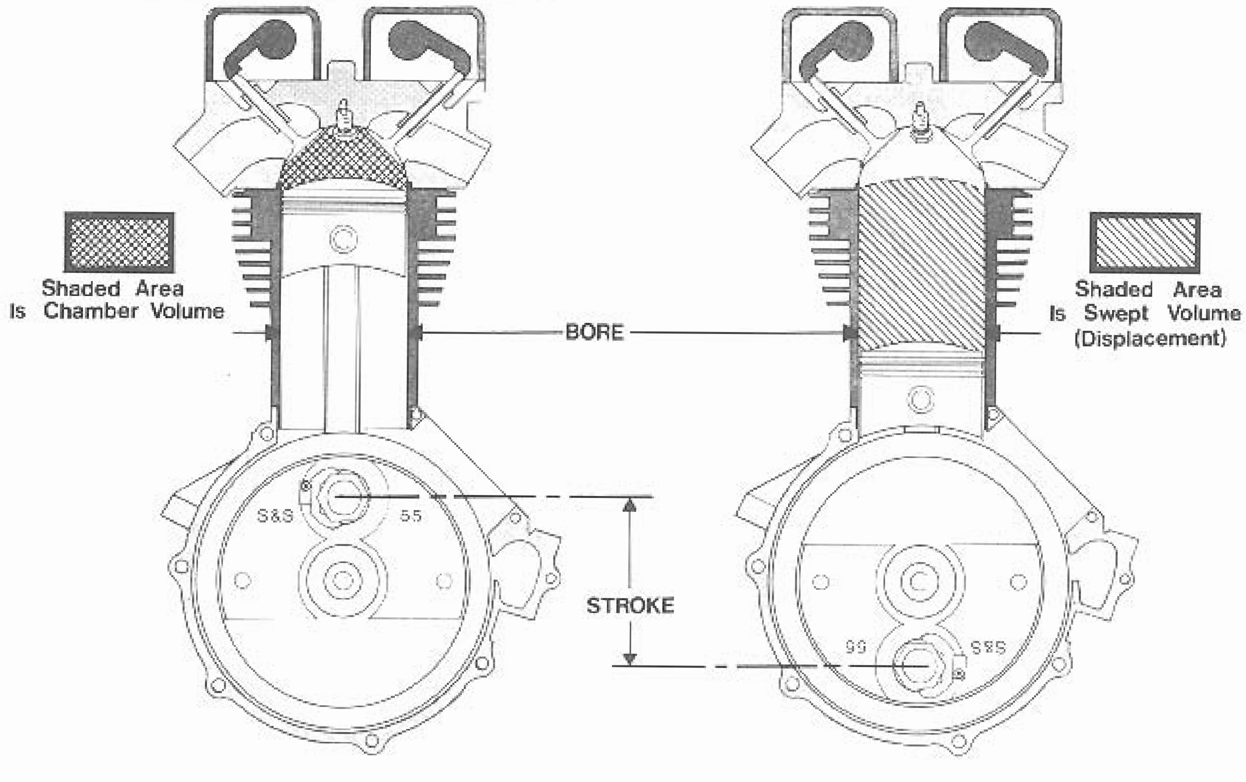
Some racers build an engine with a higher compression ratio than it is design to control, but the only tell tail symptom the rider notices when the engine is detonating is roughness or vibration. As a result, engine problems are not detected until it is too late. Consequently, numerous stress related component failures are experienced where a cause cannot be identified. Cracked heads, damaged pistons and rings, failed valves and bearings, misfiring and unexplained power losses frequently are the result of detonation. In many cases, the cause is the result of too high of compression for a given engine design.

Raising an engine's compression ratio will build more horsepower at all rpm levels as long as the efficiency of the combustion chamber is not reduced, valve flow is not reduced and detonation is not present. Keep in mind, however, that as the compression ratio is increased, there is less horsepower gain from each point of increase. For example, raising the compression ratio one point from 8.0:1 to 9.0:1 provides up to a 5 percent power increase, while raising it from 12.0:1 to 13.0:1 increases power about 2-1/2 percent. However, the percentage of increase will be greater for engines with an efficient induction system, which provides high volumetric efficiency. Long duration cams and tuned intake and exhaust systems are two methods for increasing an engine's volumetric efficiency (VE) and thereby increasing these percentages.

Since there are numerous factors that affect an engine's compression ratio, the ratio can be expressed by no less than three different methods. Understanding the different methods will provide a better understanding of compression and the relationship of the factors involved.

MECHANICAL COMPRESSION

The most common method for describing compression is referred to as mechanical compression ratio. The mechanical compression ratio is considered the industry standard method for expressing compression and it is calculated by



$$\text{Compression Ratio} = \frac{\text{Swept Volume (or Cylinder Displacement)} + \text{Chamber Volume}}{\text{Chamber Volume}}$$

Figure 6.1 *The engine's mechanical compression ratio is a function of the combustion chamber's net volume and cylinder displacement or swept volume. Illustration courtesy of S&S Cycle.*

inserting values for established volumes into mathematical formulas.

The engine's cylinder displacement (swept volume) is added to the net volume of the combustion chamber (head gasket and piston dome must be considered). The sum is then divided by the net combustion chamber volume to produce a static or mechanical compression ratio such as 8.5 to 1 (expressed as 8.5:1) or 12:1, etc. The mechanical ratio is based upon the assumption that 100 percent cylinder fill (100 percent VE) at sea-level conditions is achieved.

Although mechanical compression ratio has only a small affect on an engine's volumetric efficiency, it does promote high thermal efficiency as long as detonation is not present.

DYNAMIC COMPRESSION

Dynamic or operational compression ratio is a second way to view compression. It is the compression ratio achieved during the actual operation of the engine and it considers volumetric efficiency. You might refer to it as the "real time" compression ratio since the dynamic compression ratio of an engine increases or decreases as the engine's volumetric efficiency (VE) increases or decreases. Volumetric efficiency will increase as the engine's induction and exhaust systems are modified for improved breathing. Modifying the intake port, cam, carburetor and exhaust system are the most common methods for increasing cylinder filling.

VE is highest at the engine's torque peak since cylinder fill is highest at this rpm. However, cylinder fill is not constant, but instead wanders around at different rpm levels based upon throttle opening, engine load and other factors. For example, cylinder fill is normally reduced at rpm levels higher than the torque peak because induction system breathing is limited and there is a shorter time to fill the cylinder.

An engine that fills a cylinder to its theoretical capacity is said to have 100 percent volumetric efficiency. A stock Big Twin engine generally has 75 to 85 percent VE at maximum horsepower rpm and a slightly higher percentage at maximum torque rpm. A racing engine has a VE of 100 to 120 percent. In fact, some extremely efficient maximum effort race engines with highly tuned intake and exhaust systems can achieve over a narrow power band a VE up to 130 percent.

Mechanical compression ratio calculations are based on the assumption that the intake and exhaust valves are closed from the time the piston starts moving upward from BDC on the compression stroke to TDC. In reality, the cam keeps the intake valve open long after the piston passes BDC. This results in the valve being open while the piston is moving upward toward TDC. At low rpm, this causes some intake mixture to be pushed back out of the combustion chamber, past the intake valve and back into the intake port. This results in a theoretically smaller volume of air/fuel mixture in each cylinder. Consequently, as the piston moves upward on the compression stroke it has less volume to squeeze, which results in a lower dynamic than mechanical compression ratio.

This is the reason a higher mechanical compression ratio can be used in conjunction with a long duration cam without incurring detonation. It also is the reason an engine with a long duration cam will be down in power at low rpm unless it also includes an accompanying increase in mechanical compression ratio. The higher mechanical compression ratio compensates for the dynamic losses inherent in a cam that has a long intake duration.

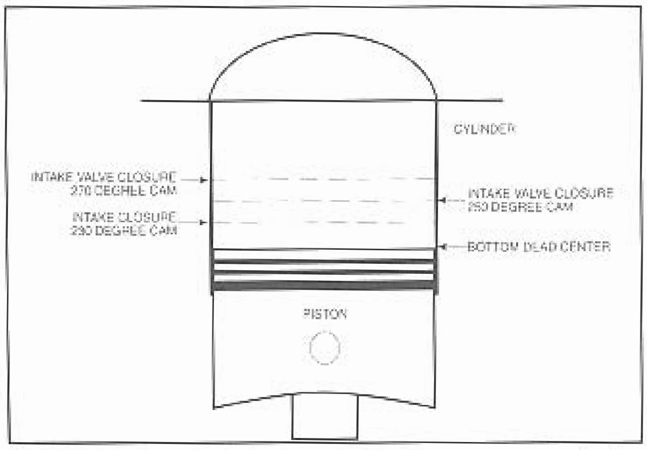
An engine's torque output starts to plummet when its volumetric efficiency drops. When volumetric efficiency and therefore torque decreases faster than rpm rises, the point of maximum horsepower is reached. As volumetric efficiency falls below 100 percent, the dynamic compression starts to drop below the calculated mechanical compression. At high rpm, a long duration intake cam can increase volumetric efficiency since the air/fuel mixture has greater force due to its higher velocity. As a result, the mixture is not pushed back out of the combustion chamber as easily. Instead, it uses inertia to ram its way past the intake valve, which packs a greater amount of air/fuel mixture into the cylinder. This increases VE at high rpm and accounts for a higher dynamic than mechanical compression ratio. Camshaft tuning is the primary method for increasing cylinder pressure or dynamic compression at *high* rpm. On the other hand, raising the mechanical compression ratio considerably improves dynamic compression at *low* rpm.

CORRECTED COMPRESSION

A third method for calculating compression ratio is referred to as "corrected compression ratio." This method is used strictly for determining approximately how much a cam's intake duration will affect compression. Although it doesn't consider all factors, it does provide a general idea what effect a particular cam will have on mechanical compression.

This method essentially uses the same formula as mechanical compression ratio, but it also takes into consideration one additional factor. As stated earlier, the mechanical compression ratio formula assumes that all valves are closed as the piston compresses the mixture. Therefore, it uses the entire cylinder displacement value within its calculation. But in reality, the intake valve remains open after the piston passes BDC. Consequently, the total or effective cylinder displacement is reduced for actual compression since compression doesn't start to take place until the intake valve closes. Remember, at this point the piston is already past BDC and positioned part way up the cylinder.

When calculating corrected compression ratio, the position of the piston relative to when the intake valve closes must be determined. The piston's position can be determined by using



Notice that as duration increases, the piston is higher up in the bore when the intake valve closes. At low rpm, a late closing intake allows some of the intake charge to be pushed back into the intake port, thereby reducing the mechanical compression ratio. The higher the piston is at intake valve closure, the higher the mechanical compression ratio should be to maintain low speed torque. A late closing intake helps top-end power at the expense of low-end torque.

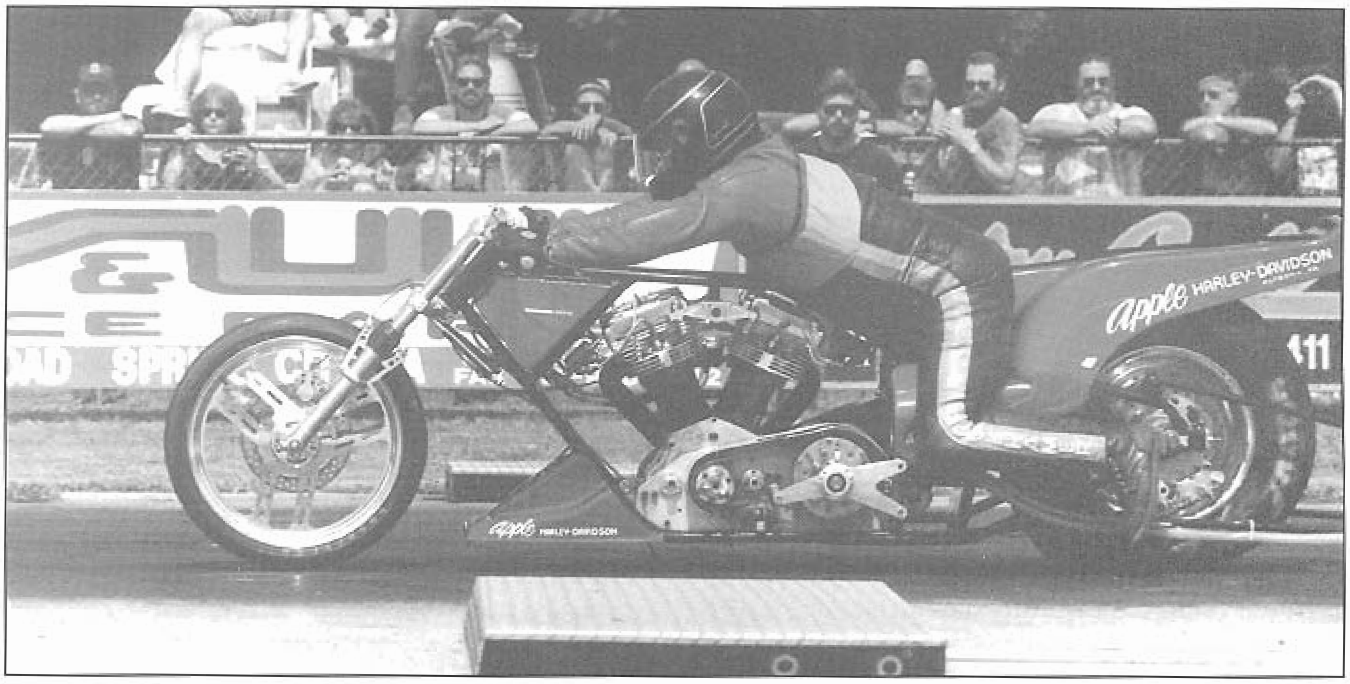
either a computer software program or by mounting a degree wheel on the flywheel's sprocket shaft and following the cam manufacturer's intake valve closing specification. The position of the piston is noted at the time the intake valve closes. Most Big Twin cam manufacturers specify cam timing at .053-inch of lifter lift; however, a few specify it at .020-inch. Regardless of where timing specifications are taken, this procedure reveals how much stroke length remains before the piston reaches TDC. This stroke length instead of the engine's full stroke length is now used to calculate cylinder displacement. The smaller value for cylinder displacement is now inserted into the corrected compression ratio formula. Essentially, if the intake valve closes earlier than the stock cam's intake, compression will increase. If it closes later, compression decreases.

To retain low rpm torque with a long duration cam, we can now determine approximately how much to raise the mechanical compression ratio to bring it back to at least the stock level.

HIGH COMPRESSION DRAWBACKS

High compression does have a number of detrimental characteristics, which should be considered when building an engine. The most obvious is the requirement for higher octane fuels. Due to increased heat, high compression engines are prone to detonation unless high octane fuel is used. The higher the octane rating of the fuel, the greater its ability to react in an orderly and controlled manner under the following conditions: compression is high, combustion chambers are hotter than normal, air temperature is extremely high, or the engine is under high load conditions.

Today, service station pump gas is usually limited to about 92-octane. When building an engine for extensive street use, it is not only inconvenient, but also costly to have to add octane boosters or race gas to pump gas to eliminate detonation. Unless the engine is built only for racing or as a "Friday night special" type engine (one that never ventures farther from home than the range of its gas tank), it is prudent to build it capable of running on 92-octane pump gas without detonation. Careful coordina-



Although Pro Fuel racers have less concern than street riders for high compression drawbacks, the engine builder and tuner must still optimize the drag engine's combustion chamber design, ignition timing and air/fuel mixture to run big numbers while eluding detonation. Tom Bookhamer on his 123 ci Pro Fuel dragster. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

tion between the combustion chamber design, intake port size and shape, and camshaft will allow the mechanical compression ratio to be as high as possible, yet still allow the engine to run on 92-octane pump gas without detonation.

High compression generates more heat and puts a greater load on the engine's rods, rod bearings and crankpin. As a result, the engine's oil temperature is higher and parts wear out quicker. Adding an oil cooler will help keep oil temperature below 220 degrees on a high output street engine. This not only will extend the oil's useful life, but also will reduce engine wear and the potential for detonation.

To make reasonable power, a Big Twin's compression ratio must be raised above the stock 8.5:1. Approximately 9:1 to 9.5:1 is the minimum for making any reasonable power. However, by raising the compression ratio two points, from 8.5:1 to 10.5:1, crankshaft and rod longevity will be reduced by up to 50 percent. On a race engine, the reduced longevity has no consequence. However, on a street engine you should consider your style of riding and your tolerance for engine maintenance when deciding what compression ratio to run.

ALTERING COMPRESSION RATIO

The mechanical compression ratio of a Big Twin engine can be altered either up or down by modifying or replacing one or more of a number of parts.

The most common method for changing an engine's compression ratio is to increase the piston's dome mass. Another common method, especially with the Evolution, is to reduce the combustion chamber volume by either (or in combination with) machining the head gasket surface or by welding and reshaping the chamber walls. Installing a different shape intake or exhaust valve, or setting the valves at a different included angle in the head is still another method for changing the compression ratio. Modifying the length of the cylinder, or changing the thickness of the head gasket or cylinder base gasket will also alter the engine's compression ratio. Finally, increasing the engine's displacement through boring and/or stroking for a given combustion chamber volume is another method.

In fact, a large increase in displacement will raise compression while the piston's dome mass is reduced in size. This makes it easy to increase compression without introducing performance drawbacks.

COMBUSTION CHAMBER DESIGN

Combustion chamber design plays an important part when designing an efficient engine and it plays an even more important part when raising the engine's compression ratio. Raising compression generates more heat from the air/fuel mixture, thereby producing more pressure and consequently more horsepower. However, greater heat can be attained not only with higher compression, but also through greater cylinder filling. Additionally, burning a higher percentage of the air/fuel mixture also increases heat. So both compression and mixture quantity are important.

During engine operation, combustion chamber design not only plays a part in the actual compression ratio, but also in the percentage of cylinder fill and the quality (condition) of the air/fuel mixture at the point of ignition. When designed properly, a combustion chamber will generate the maximum amount of thermal energy to drive the piston down instead of heating extraneous engine parts.

The rpm level at which an engine runs out of air determines its torque peak. If more air is flowed through the engine, its torque peak moves to a higher level in the rpm band. The flow characteristics of the intake and exhaust systems, the carburetor size and the cam determine the torque peak. However, the shape of the combustion chamber and piston dome, the location of the spark plug and the temperature of all areas of the chamber dictate the power *potential* that can be achieved with a particular engine combination.

Good combustion chamber design will optimize the combustion process and thermal efficiencies. As a result, the optimum amount of heat energy will be extracted from the air/fuel mixture and will be used to drive the piston down instead of heating the engine components.

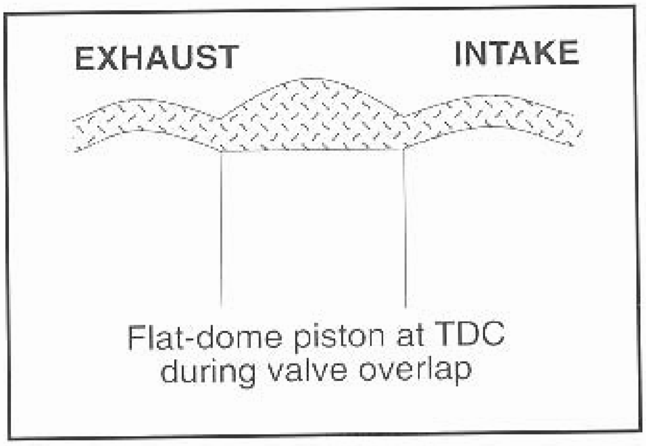
To achieve maximum power, the design of the entire combustion chamber is important. The combustion chamber consists of a roof that is defined by the cylinder head and a floor that is defined by the top of the piston. The shape of the chamber's roof and floor, along with their relationship are critical for maximum cylinder filling, encouraging full atomization of the air/fuel mixture and providing unobstructed flame travel.

An efficient chamber design has the following characteristics: it does not restrict the flow of incoming or outgoing gases; it generates a high amount of air/fuel turbulence when the piston is near TDC; it confines the air/fuel mixture in the smallest possible area near the spark plug; and it provides a short, unobstructed path for flame travel.

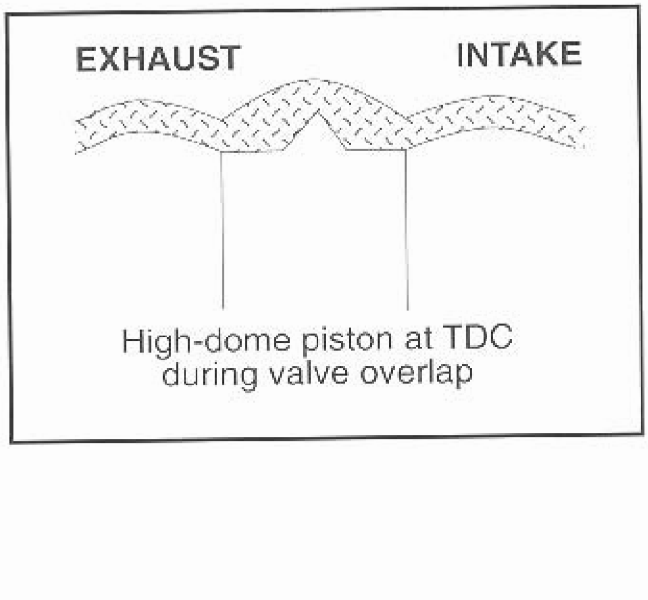
ADDING PISTON DOME

The most common method for increasing compression ratio is to add piston dome. However, if piston dome mass is added indiscriminately, cylinder filling and thermal efficiency can be reduced.

If the piston has a high dome, which gives a high mechanical compression ratio, the dome



A high dome piston can improve power and torque at low rpm, but it also can hurt power on the top-end if its dome interferes with crossflow during the valve overlap period. This is a greater concern with a Shovelhead engine because it needs a large piston dome to make high compression. Before indiscriminately adding piston dome, give thought to the overall combustion chamber design—including its roof and floor.



can interfere with the cross-flow stage (overlap period) of the camshaft. This is where both valves are opened at the same time — the intake valve is just starting to open and the exhaust valve is about to close. At this point, the piston is approximately at TDC and the air/fuel mixture is entering the chamber from the intake port. The mixture then moves across the chamber (which includes the piston dome) and out the exhaust port. If the piston dome significantly interferes with the movement of the intake charge, volumetric efficiency is dramatically reduced at *high* rpm (4500 rpm and above) with a resultant loss of torque and horsepower. However, during operation *below* 4500 rpm the torque output of an engine with a very high dome piston will be greater (in most cases) than one with a low dome piston that does not interfere with cross-flow. This condition sacrifices horsepower at high rpm in exchange for greater horsepower down low. This is due to the intake port's ability to flow enough air at low rpm for satisfactory cylinder filling.

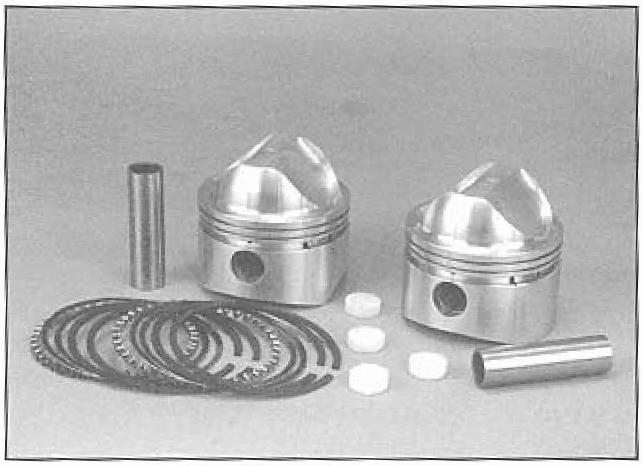
In general, engines with high piston domes that cause a high amount of cross-flow interference require a greater amount of cam overlap to produce good horsepower. This creates a problem because long overlap reduces power and throttle response at low rpm levels, besides making the engine run rough.

Big Twin pistons come with various shaped domes to accommodate different combustion chamber designs, stroke lengths and compression ratios. The flat top piston on the left is for use in an Evolution while the other three are for the Shovelhead. Photo courtesy of S&S Cycle.

The Axtell 30 degree angle top piston for the Evolution gives about 10:1 compression. It is designed for a single squish band combustion chamber and requires a 30 degree angle cut into the stock chamber's squish ledge. Photo courtesy of Axtell Sales.

Very high dome pistons can also reduce turbulence (highly desirable for good flame propagation), decrease flame speed, slow down burning and may require reduced ignition timing to avoid detonation. A high dome piston also exposes more dome to the flame front during combustion. This not only increases the heating process of the intake charge, which results in less efficient cylinder fill throughout the cylinder filling process, but it also siphons heat away from the combustion process. The result is less combustion pressure pushing on the piston.

Axtell 30 degree full dome Evolution piston gives about 10:1 compression when used with a bathtub style combustion chamber such as a STD or Branch #4 head. Photo courtesy of Axtell Sales.



Shown are S&S Shovelhead 15:1 CR racing pistons. Note the large dome required to get high compression in the Shovelhead's large hemispherical combustion chamber. The dome can be machined lower to match a custom chamber shape. Teflon buttons and spiralock clips are included to give a choice of piston pin keeper. Photo courtesy of S&S Cycle.

In general, it is best to run as low a piston dome as possible, consistent with the mechanical compression ratio objectives for the engine. The ideal piston design produces maximum BMEP by balancing a high mechanical compression ratio against the potential loss of volumetric efficiency. Remember, engine output primarily is dependent on filling the cylinder with the greatest amount of air/fuel mixture and then burning it properly. If less air/fuel mixture is drawn into the cylinder and turbulence is retarded because of poor piston design, the piston with a high mechanical compression ratio will produce significantly less power in the high rpm range than a piston that achieves maximum *effective* compression.

A high dome piston that interferes with crossflow during overlap and obstructs the flame front can still produce high levels of low and midrange torque. However the engine will be down in power at high rpm levels (above 4500 rpm). This is because the high mechanical compression ratio allows acceptable volumetric efficiency in the low to mid rpm range, but limits volumetric efficiency in the high rpm range. This results in reduced dynamic compression. Remember, dynamic compression is "real time" compression. If the engine's objective is to produce maximum power above 4500 to 5000 rpm, then it must have maximum high rpm breathing capability even if it requires a sacrifice in the mechanical compression ratio (read: "less piston dome interference").

In reality, race track testing and dynamometer engine pulls in the acceleration mode are the only valid measures of a piston's effectiveness.

TURBULENCE

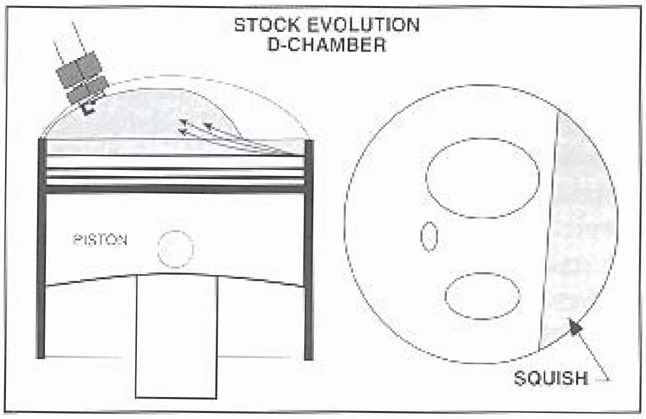
When modifying the combustion chamber or when raising compression, ensure that as much air/fuel mixture turbulence as possible is generated. This will guarantee that the greatest percentage of air/fuel mixture will be burned and thermal efficiency will be maximized. To discuss turbulence it is also necessary to elaborate on piston dome shape, chamber shape and chamber volume.

The squish band technique is one of the most common methods used to enhance air/fuel turbulence before combustion. Areas where parts of the cylinder head and piston dome come close together at TDC are referred to as squish bands (the more technically correct term is quench band). As the piston nears TDC, mixture trapped between the head squish area and piston is forced out at high velocity. This increases turbulence, which promotes better air/fuel atomization and reduces the potential for detonation.

Most turbulence is generated during the intake stroke as the inlet flow is rushing through the narrow valve opening and at the end of the compression stroke where the mixture flow is forced into the combustion chamber from the squish band area. Besides the squish band, the intake port can have a significant affect on the amount of turbulence generated. The intake port's size, its shape within approximately one inch of the valve seat and its bias to the valve's position can all influence the amount of turbulence that takes place in the cylinder.

EVOLUTION SQUISH

The D-shaped Evolution head includes a squish band located in the combustion chamber opposite the spark plug side. The squish band results from a 90 degree ledge that is lowered over a flat piston dome to create a thin horizontal squish zone. For the Evolution engine, the ideal squish band has about 0.025 inch to 0.040 inch clearance. Anything larger, diminishes the squish

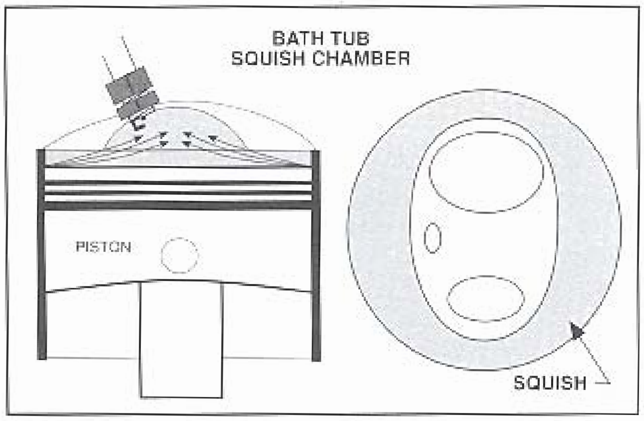


Optimizing the Evolution's squish band is crucial for making big horsepower. Get the squish band clearance to between .030 and .040-inch for a street engine and down to .025-inch for a race engine. Machine the cylinders or change gasket thickness to obtain the desired clearance.

effect. However, 0.025-inch clearance on an engine that is not disassembled frequently can allow the piston dome to contact the squish ledge due to carbon build up. Therefore, it is better to go with a minimum of 0.030-inch clearance on a street engine.

Cylinder length, cylinder base gasket thickness and cylinder head gasket thickness can be varied to control squish clearance. The head gasket surface of an Evolution cylinder head is frequently machined between 0.050 and 0.070-inch to increase the mechanical compression ratio. Keep in mind, however, that squish band clearance is not affected by this machining.

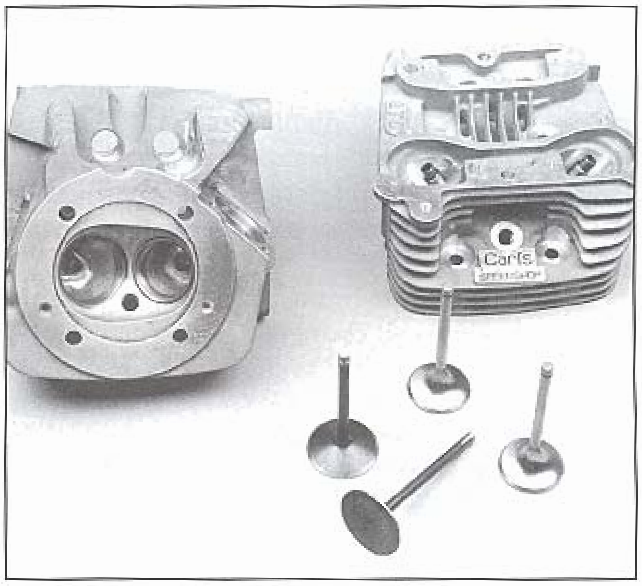
The Evolution chamber design can be improved by filling in the chamber sides through welding and then recontouring its shape similar to a bathtub. This design adds a duplicate squish



A flat top piston is usually teamed with a bathtub chamber. However, domed pistons that match the bathtub configuration are starting to appear in the marketplace.

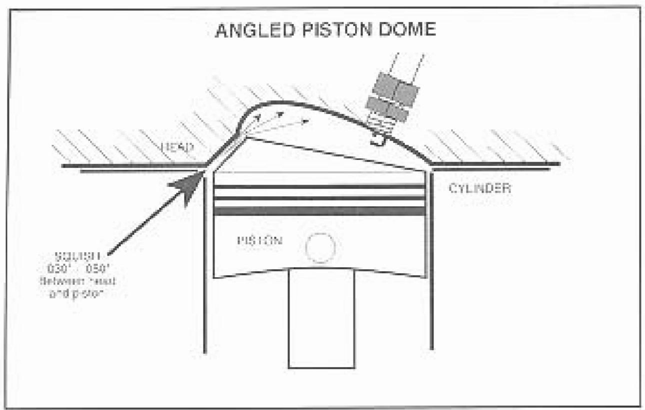
area to the opposite side of the chamber. The bathtub shape improves turbulence, increases torque and can reduce potential detonation. If done properly, it can also increase airflow by reducing valve shrouding.

The bathtub chamber is sometimes reduced in volume through the welding process or by machining 0.050 to 0.070-inch from the head gasket surface. Either modification will increase the engine's mechanical compression ratio. Bathtub chambers most frequently are used in conjunction with flat top pistons and between 0.025 to 0.040-inch squish band clearance. Arias offers a piston with a bathtub shaped dome that provides higher compression than a flat top piston.



These are STD bathtub chamber heads modified by Carl's Speed Shop. Take note of the twin squish ledges that increase turbulence and keep the chamber small. A flat dome piston is often used with bathtub chambers. For higher compression, domed pistons are available from Axtell Sales, Rivera Engineering and others. These heads are setup for large bore cylinders. Photo courtesy of Carl's Speed Shop.

Axtell sells a forged, high silicone, 30 degree angle top piston that increases compression, maintains sufficient squish and enhances cylinder breathing. This piston requires that the combustion chamber's 90 degree squish ledge be machined at a 30 degree angle to match the piston's angled dome. This results in a squish band that generates turbulence about equal to a stock Big Twin chamber, however, the 30 degree angled dome and the angled squish band force the air/fuel mixture more directly toward the



Axtell's 30 degree angled top piston raises compression to 10:1, provides a good squish band for high turbulence and retains an unobstructed flame path. This dome configuration is designed for use with a modified 90 degree squish chamber. A full 30 degree dome piston also is available for use with bathtub style chambers. The full 30 degree dome piston has an angled squish band on both sides of the bathtub chamber and provides about 10:1 compression.

spark plug. A stock displacement 30 degree piston is rated at 10:1 compression ratio when used with an 84 cc combustion chamber. Pistons for other bore and stroke combinations are available.

In summary, whenever the combustion chamber's shape is changed, regardless of the reason, it is extremely important to pay close attention to the squish area so turbulence will be maximized. It is also important to ensure that the valves are not shrouded by the chamber walls or the piston dome so airflow is not restricted.

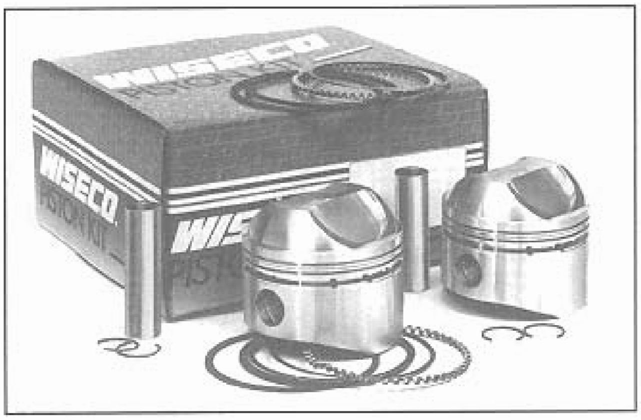
SHOVELHEAD SQUISH

The deep hemi-chambered Shovelhead (and the iron Sportster head) creates very little air/fuel turbulence because it has little or no squish band. Unfortunately, this is one of the inherent characteristics of a hemispherical chamber design. Although a properly reworked Shovel cylinder head flows extremely well, its non-turbulent, slow burn chamber typically passes a high percentage of unburned fuel out the exhaust. Consequently, mechanical efficiency and horsepower are reduced.

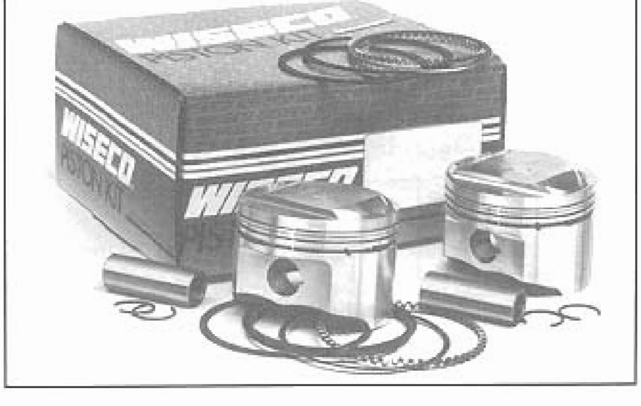
Because of poor turbulence, whenever building a Shovelhead engine it is critical to do whatever is possible to create squish area. Large bore cylinders make it easier to generate squish area on this engine. Also, the lower radius edge of the combustion chamber can be machined at an angle equal to the piston dome's angle to help create a squish band. The cylinder length then can be adjusted to provide .040 to .060-inch clearance between the piston dome and the head's newly machined squish band.

Another helpful modification is to minimize any combustion chamber nooks and crannies where air/fuel mixture is excessively cooled and goes unburned. The cylinder locating ring that raises up into the Shovel's combustion chamber can create such dead areas. Make sure there is no gap between the locating ring and the cylinder head after the head is bolted down using the correct head gasket.

Because of its inherent design (Volkswagen style locating ring), a Shovel cylinder head is generally not milled to increase compression. Instead, compression is normally increased by



Above is a 9.5:1 CR Shovelhead piston and below is a 9.5:1 Evolution piston. Note the difference in dome height required to achieve the same compression ratio. The Shovelhead requires a high piston dome that wastes combustion heat and can interfere with flame travel. Photo courtesy of Wiseco Piston.



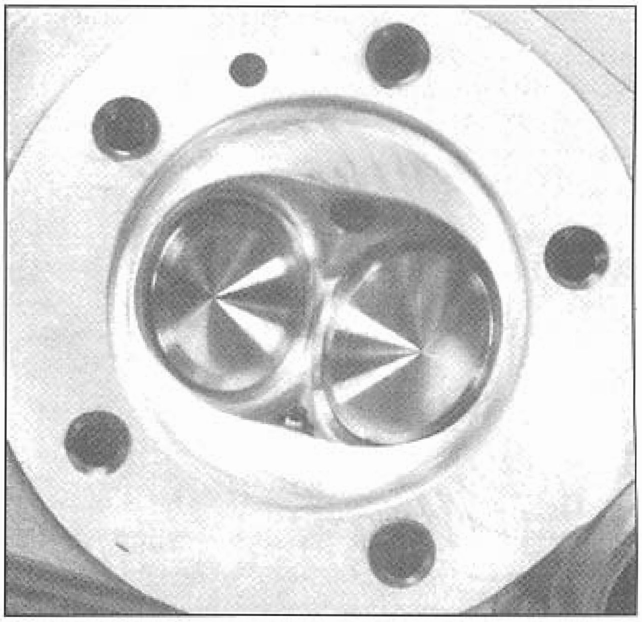
raising the piston dome or by increasing the engine's displacement. With a stock or near stock displacement Shovelhead engine, a piston dome at least 3/4-inch high or higher is usually needed to get a reasonable amount of compression due to the deep hemispherical chamber. With large displacement engines it is easier to make compression with a lower dome piston. If the head's gasket surface is machined to ensure a flat surface, make sure a sufficient recess is retained for the cylinder's top spigot. Also, counter sink each head bolt threaded insert .006-inch for a good head gasket seal.

Some engine builders and cylinder head porters prefer to weld the Shovelhead's chamber for higher compression and better turbulence. The result is either a bathtub or hemitub shape. The bathtub chamber resembles a welded and reworked Evolution bathtub chamber and it is used in conjunction with an Evolution flat top piston. This style chamber greatly increases the squish area and gives reasonably high compression with a flat top piston. The hemitub chamber is similar to a bathtub design. The major difference is the hemitub chamber uses a standard crowned dome piston rather than a flat dome to generate higher compression. This results in an angled instead of horizontal squish band.

It is possible to inadvertently shroud the valves and reduce airflow when welding Shovelhead chambers. The key to making high horsepower with a bathtub or hemitub Shovelhead chamber is to ensure that intake and exhaust airflow does not suffer at the benefit of increased compression and squish.

FLAME PROPAGATION

Up to this point, much has been said about maximizing cylinder head airflow for increased horsepower. However, high airflow does not guarantee high horsepower because the efficiency of the combustion chamber plays a major role in the amount of horsepower made from a given amount of air/fuel mixture. An efficient chamber not only must have a high degree of turbulence and good cross-flow (scavenging) characteristics, but it also must burn efficiently a high percentage of the air/fuel mixture. And this requires good flame propagation in the chamber.



Head Quarters modified this Shovelhead combustion chamber by welding and machining it into a bathtub shape. This modification reduces combustion chamber size, shortens flame travel and significantly improves chamber turbulence for a more complete burn. A piston with a 30 degree angled dome is now used to generate turbulence between the twin squish bands. One key point to this modification is to keep port airflow high by not shrouding the valves with the chamber walls. Photo courtesy of Head Quarters.

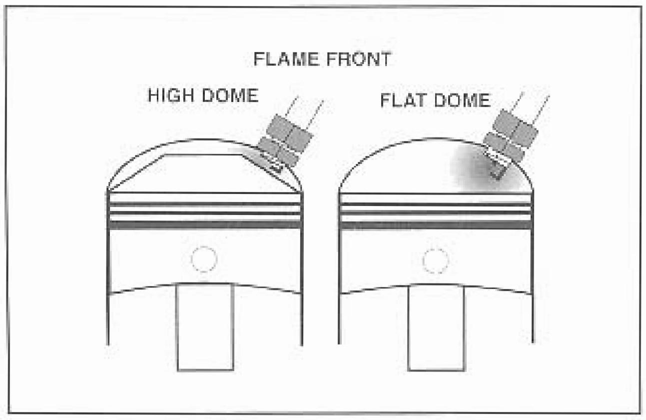
One of the best methods for determining an engine's combustion efficiency is to note the optimum point where full advance ignition timing is set. The closer ignition is to TDC (read: "less ignition advance"), the more efficient the combustion chamber is. Under typical combustion conditions, peak cylinder pressure occurs well after TDC. For efficient chambers it is generally in the area of 12 to 17 degrees ATDC and with inefficient chambers it can be much later. If peak pressure takes place too late, the piston's downward movement reduces cylinder pressure faster than the expanding gases create it and power is lost.

Since the optimum ignition advance for a Big Twin engine usually ranges between 27 and 40 degrees BTDC, a considerable amount of cylinder pressure is working against the piston before TDC. The upward movement of the piston is opposed by the downward pressure exerted on it from the burning fuel charge. This is because the combustion process is started while the air/fuel mixture is still being compressed. The optimum ignition point is reached when less ignition advance moves the maximum cylinder pressure point too far ATDC, and more advance creates too much resistance on the piston. Therefore, the better the combustion chamber design, the less ignition advance it should need, all other things being equal.

To produce maximum power, combustion ideally should take place between TDC and the point at which the exhaust valve opens (generally between 42 and 65 degrees BBDC for the Big Twin). To be more exact, maximum cylinder pressure should occur at the point where the piston exerts maximum leverage on the crankshaft. Very efficient engines (not the Big Twin) produce 90 percent of maximum cylinder pressure by the time the piston reaches 45 degrees ATDC. Consequently, the combustion process must be extremely rapid.

Assuming a high level of mixture turbulence is already present, a few combustion chamber modifications can be performed to speed up the combustion process. These modifications are designed to increase the area of the flame front, increase the flame speed and increase the density of the intake charge. Each modification results in centering the combustion event more closely around TDC, which will allow the combustion pressure to perform more efficient work on the piston and thereby produce more horsepower from a given amount of work.

When a flat top piston or a piston with a small dome is used, the flame front can grow across the chamber easily without any obstruction from the piston dome. However, when compression is increased by a high dome piston, the chamber shape is modified and the dome can interfere with the flame front, especially with the deep hemispherical Shovelhead chamber. This can increase the required ignition advance due to a slower combustion rate and increase brake specific fuel consumption (BSFC) due to pockets of unburned fuel. The important point to remember is that it is possible to lose horsepower when increasing compression with a high dome piston if you do not pay attention to combustion efficiency.



A large combustion chamber like the Shovelhead's needs a high piston dome for high compression. The high dome can interfere with flame propagation, slow combustion, require more ignition advance and raise the engine's BSFC. Don't use a higher piston dome than is necessary for a given compression ratio and consider fire slotting the dome near the spark plug location for better flame propagation.

Regardless of the piston dome shape, any sharp corners on the dome should be smoothed as much as possible so the flame front can propagate smoothly to the farthest corners of the chamber. Also, the piston dome should be shaped so it does not shield the main body of the air/fuel charge from the spark plug. Instead, it should force the intake charge directly toward the spark plug or flame front. Furthermore, a high piston dome can include a spark plug flame slot to enhance the flame front.

With the Evolution engine it is generally better to increase compression by machining the head's gasket surface than it is to add piston dome height. However, be aware that machining the head may change the engine's rocker arm geometry, which can reduce the effective valve lift. Changing the pushrod length and reshaping the rocker arm contact pad can bring the geometry back to proper specifications.

DUAL SPARK PLUGS

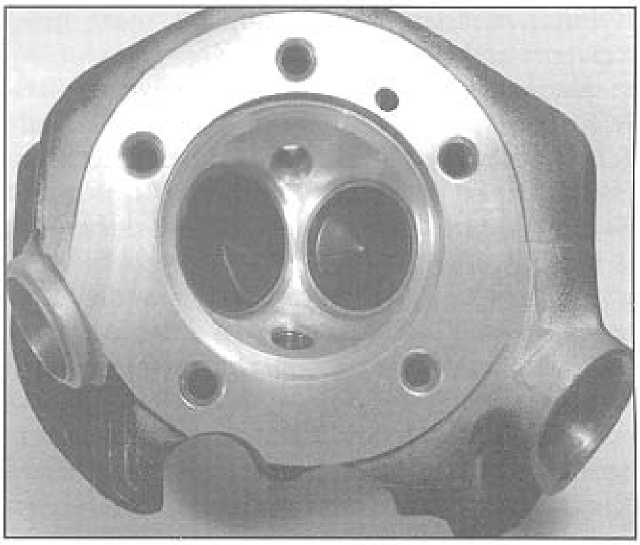
Minimizing the distance between the spark plug and the farthest part of the combustion is another factor to pay close attention to. The ideal spark plug location is one centrally located in the combustion chamber. This would minimize the distance the flame front needs to travel, thereby reducing the required amount of ignition advance and the potential for detonation. However, centrally locating the spark plug in either an Evolution or Shovelhead will cause valve size to be restricted. As a result, a single spark plug is located on the side of the chamber directly opposite the pushrod side.

Adding a second spark plug to the cylinder head effectively reduces the distance the combustion flame front needs to travel. This can increase the amount of air/fuel mixture burned and reduce the time needed for combustion. This results in less potential for detonation.

Detonation raises the engine's temperature dramatically. In some situations detonation is inaudible, but regardless of whether you hear it, in a worse case scenario it will destroy a piston and in a best case situation it will stop your engine from reaching maximum power. Assuming the combustion chamber shape is optimized and there is no oil contaminating the intake charge, there are three major methods for eliminating detonation: the fuel's octane can be increased; cylinder pressure can be reduced through installation of a different cam or reduced mechanical compression ratio; or the heads can be dual plugged. Raising the fuel's octane level higher than pump gas is generally not practical with a street driven engine and lowering cylinder pressure reduces horsepower. But adding a second plug offers neither of these negatives.

By installing a second spark plug opposite the original plug, the air/fuel mixture starts combustion at two separate locations instead of one. The second plug reduces the distance between a plug and the farthest part of the chamber. As a result, the time required for combustion decreases and a more complete and uniform burn of the air/fuel mixture is achieved, particularly with a high dome piston. Furthermore, the potential for detonation is greatly reduced and maximum horsepower can be realized.

Since combustion time is reduced with two spark plugs, the point of spark ignition can be delayed so the timing of maximum cylinder pressure remains correct. For example, normal ignition timing for the Big Twin is approximately 35 degrees BTDC. With two plugs, the normal amount of advance should be reduced (retarded) by 5 to 8 degrees, which results in timing between 27 and 30 degrees BTDC. Since the crankcase timing hole is about 10 degrees in width, you can adjust the ignition advance by placing the flywheel timing mark at the front edge of the hole or just beyond the front edge instead of centered to get the correct timing. Remember, this is only an example. For your application, subtract the 5 to 8 degrees advance from the optimum single plug ignition setting for your engine. Be aware that failure to retard the ignition timing will nullify any potential benefits of dual plugs.



Dual plugs reduce flame travel and minimize the potential for detonation. This Shovelhead chamber is setup with dual plugs and bored for 3-5/8" bore cylinders. Second plug must be located high enough in the chamber for good efficiency and sufficient clearance between the piston and plug electrode. To obtain some squish area, the chamber is machined at a 30 degree angle to match the piston dome. This is about all the squish area you can get with a Shovelhead chamber without welding. Note how the head bolt holes are counter sunk .006-inch so the head gasket seals properly.

If you ever revert to using one spark plug, don't forget to advance the timing back to normal specifications because a severely retarded ignition will significantly increase engine heat. Also, take note that dual spark plugs (4 plugs total) require the use of two ignition coils (dual tower each), and both must have the proper voltage and ohm rating.

When determining the location for the second plug, be sure to locate it high enough above the head gasket surface. The proper location (check the level of the stock plug) will not only allow the piston to miss the spark plug, but also will provide better flame front propagation around and over the piston dome. Also, ensure that the plug can be installed and removed without having to remove or loosen the pushrod tubes. This is not too difficult on the Shovelhead, but it must be watched closely with the Evolution. For Evolution models, some racers install a

12mm instead of 14mm plug for more head strength and easier access.

Be sure never to install a short reach (3/8-inch) spark plug directly into aluminum threads because it can be blown out if the engine backfires. Instead, use a carbon steel thread insert for a secure mounting. And make sure the plug is not installed too deep into the head because plug threads exposed in the chamber will cause pre-ignition. Finally, gently blend the new plug hole into the combustion chamber, but be sure you only remove the minimum amount of material. Aftermarket performance cylinder heads normally include two spark plugs per combustion chamber, so you don't need to go through the expense of installing a second plug.

The major benefits of dual spark plugs are higher performance and a slightly smoother running engine. For a given fuel octane rating, dual plugs allow the use of a higher mechanical compression ratio without incurring detonation; or for a given mechanical compression ratio, dual plugs allow the use of a lower fuel octane rating. The larger and more inefficient the combustion chamber and the higher the piston dome, the greater the benefit from dual spark plugs.

Just about every Shovelhead engine can benefit from dual plugs. The only exception might be extremely low compression (7.4:1) engines manufactured from 1981 through 1984. On a properly setup Shovelhead engine, the stock mechanical compression ratio usually can be raised between one and two points (for a 9:1 to 10:1 compression ratio) when using 92 octane gas at sea level locations. It can be even higher at high altitude locations. However, keep in mind that the combustion chamber principles previously discussed must be adhered to. This means you must minimize chamber obstructions, create the maximum amount of squish area and eliminate all oil contamination to the fuel due to poor sealing piston rings or worn valve guides and seals.

COMPRESSION RATIO RECOMMENDATIONS

Over the last 20 or 25 years Big Twin engine's have steadily increased in performance. Some of the increases are directly attributed to larger displacement engines, better breathing heads and chassis that hook up better. The ability to run a higher compression ratio also can be credited for some of the performance increase. Today's Big Twin engines can effectively use higher compression ratios not only because they have a more efficient combustion chamber, but also because the gasoline is better quality and ignition systems are much improved.

For maximum engine performance when burning gasoline, the benefits derived from increased compression flatten out at approximately 14.5:1 — maybe 15:1. A stock Big Twin Evolution has an 8.5:1 compression ratio. For a stock Shovelhead, compression is either 7.4:1, 8:1 or 8.5:1, depending on the bike's year and model. As you can see, Big Twin compression ratios span a wide range. The question now becomes, "What compression ratio should I run?" The answer is, "It just depends."

The are many factors that may influence your decision, but the major ones to consider are:

- Will race gas or 92 octane pump gas be used?
- What is the objective for the bike and how will it be used?
- How competitive is your racing class?
- How much engine maintenance are you willing to perform?
- Are you able to afford other necessary performance components such as a high-performance ignition system?
- How high is the engine's volumetric efficiency?
- Is the bike a heavy 800 pound "dresser," a 600 pound cruiser or a 450 pound dragster?

Answers to these questions should provide the correct answer.

It is difficult to give specific compression ratio recommendations because of the variables just mentioned; however, the following guidelines can be considered.

If your bike is intended mainly for street use, the only practical solution for fuel is 92 octane pump gas. Although you can use race gas, having to depend on it for a street machine is nothing but aggravation unless you have what is referred to as a "Friday night special" — a bike used sparingly and seldom driven farther than the distance one tank of fuel will take it.

If you limit your engine to pump gas, then your compression ratio will also be limited. On pump gas, an Evolution Big Twin can generally run between 9:1 and 10:1 compression without incurring detonation. Of course, this depends on the combustion chamber design, amount of turbulence (read "effectiveness of squish band"), whether the chamber is single or dual plugged, besides the cam specifications (duration, and LSA).

With a Shovelhead engine you're probably limited to between 9:1 and 9.5:1 compression on pump gas without encountering detonation, although some astute engine builders are able to squeak out about 10:1. Again, the same dependencies listed for the Evolution also apply to the Shovelhead.

Any engine with about a 10:1 or higher mechanical compression ratio normally will require race gasoline. For an Evo street engine running on race gas, strive for an 11:1 to 12.5:1 compression ratio. For a maximum effort Evo engine you'll usually need to be in the 14:1 to 16:1 range to be competitive. For a maximum effort Shovelhead engine set the compression between 13:1 and 15:1. However, keep in mind that parts break more frequently with extremely high compression and the engine becomes much harder to start, especially with a stroke length greater than 4-5/8 inch.

A beefed up engine is no fun if you can't start it, so don't forget that the electric starter motor must be able to turn over the engine. Carl's Speed Shop and Tech Products offer heavy duty starter motors for Evo Big Twins that are just the ticket for a high compression engine. Also, a high amp battery can help a stock starter motor.

CHECKING COMPRESSION

A high mechanical compression ratio is wasted if you cannot contain the compression and combustion pressures within the cylinder and the combustion chamber. Containing these pressures requires a good seal not only between the piston rings and cylinder wall, but also between the intake and exhaust valves and their seats. Either a compression pressure test or a cylinder leakage test can be performed to verify whether the engine's rings and valves are properly retaining cylinder pressure.

The *compression pressure* test uses a compression gauge screwed into the cylinder's spark plug hole to record the cylinder pressure in pounds per square inch (psi) while the engine is cranked over. To perform this test, the engine should be warm and the carburetor throttle must be held *wide open* when the engine is cranked. The pressure of both cylinders and the variance of pressure between the cylinders are important to note.

With a compression gauge, the engine's dynamic compression ratio at cranking speed can be roughly determined. This can be calculated by adding one atmosphere (14.7 psi for atmospheric pressure) to the cranking pressure and then dividing the sum by 14.7. Due to the dynamics of cylinder filling, the calculated ratio will usually be higher than the engine's mechanical compression ratio. If we assume an engine's cranking pressure is 170 psi the calculation is as follows:

$$CR = \frac{170 + 14.7}{14.7}$$

This results in a dynamic compression ratio of 12.57:1 at cranking speed. The important factor here is not the calculated dynamic compression ratio, but the 170 psi cranking pressure. For a moderate compression engine (8.0 to 10.0:1), a reduction in cranking pressure of two atmospheres (about 30 psi), reduces power output by about 10 percent and this power change can be noticed by an average rider. A performance engine is not going to be competitive when its cranking pressure is two atmospheres (30 psi) less than optimum. Additionally, an average street driven engine will exhibit signs of lazy acceleration at this point and should be rebuilt.

The engine's service manual normally specifies the acceptable cylinder pressure value. Depending on the year, a stock Evolution Big Twin engine will typically have a cranking compression of about 145 to 150 psi. Remember, many factors such as air temperature, air density, mechanical compression ratio, carburetor size, port size, camshaft and valve lash settings can alter the pressure reading. After an engine is broken-in, it is a good idea to perform a compression test for a baseline value that can be referenced later.

Knowing the engine's cranking compression can also provide important insight about the engine's starting characteristics, detonation potential and performance expectations. Some cam manufacturers provide cranking compression information. You can compare your engine's cranking compression to the manufacturer's guidelines. For example, comparing the information can tell you if the engine's mechanical compression is too low for the camshaft and whether poor low speed performance can be expected. It can also tell you if the compression is very high and hard starting and detonation is probable. A compression test is a quick and easy method for determining the condition of the engine's rings, bore and valves, but the next test is more informative.

A *cylinder leakage* test is sometimes referred to as a leak-down test because it uses a special gauge and a compressed air supply (usually 100 psi) to measure the percentage of air that leaks from the cylinder. With this test, the piston for the cylinder to be tested must be positioned at TDC. Also, the transmission should be placed in high gear and the rear brake applied to stop the flywheels from rotating. The test gauge is connected to the cylinder's spark plug hole and to an air supply of at least 100 psi. Next, the gauge is opened to fill the cylinder to 100 psi. The gauge will then reflect the percentage of air leaking from the cylinder. This test not only specifies the percentage of leakage, but it also allows you to hear the leaking component such as rings or valves.

You need a good sealing pump to make high horsepower. Use a leak down tester to check cylinder sealing. Check your engine after every race or when something seems to be wrong. This test can identify leakage past a valve or cylinder rings. Stock engines usually have between 6 and 8 percent leakage. Strive for no more than 2 percent.

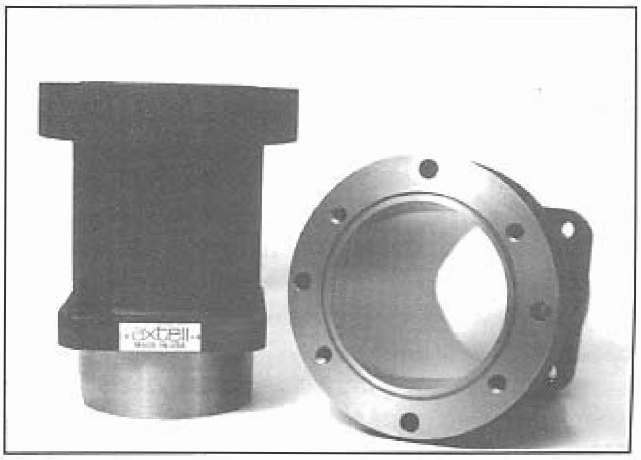
A typical new engine will show a leak-down rate of six to eight percent. As the engine wears, this can increase to twenty or more percentage. An engine with twenty percent or greater leak-down rate is due for a rebuild. Any engine with more than a ten percent leak-down rate is going to be significantly down in power. Quality built engines and maximum effort engines will reflect a leak-down rate of no greater than two percent.

Of the two types of cylinder tests, the cylinder leak-down test is better because it accurately defines the amount of cylinder leakage and allows the location of the leakage to be determined.

HIGH COMPRESSION TIPS

- To make reasonable power with a Big Twin engine, the mechanical compression ratio should be at least 9:1.
- Due to the V-Twin's uneven firing pulses, power sometimes can be increased by setting the rear cylinder's compression ratio to 1/2 point higher than the front cylinder's.
- There are no tuning procedures that will recover lost power due to lost compression.
- For a given engine combination, compression can be raised at higher altitudes due to thinner air.
- Engines with low VE due to a restrictive carburetor or limited cylinder head airflow can benefit from higher compression (read: "large displacement engine with stock heads or carb").
- For high rpm power, it is better to sacrifice compression than cylinder head breathing (read: "high piston dome interfering with cylinder filling").

- In some cases, engines that do not respond to tuning changes may be helped by an increase in compression.
- As compression is lowered, throttle response will decline.
- Higher compression can effectively be used when the combustion chamber(s) temperature can be evened up to within 50 degrees F. from the intake to the exhaust valve sides.
- Aluminum cylinder heads can generally tolerate between 0.5:1 to 1.0:1 point higher compression than iron heads, given all other things are equal.
- Heavy "dresser" model bikes will tend to detonate easier when launched or when under full throttle load than lighter models.
- Detonation can easily become present when the slightest amount of oil leaks into the combustion chamber.
- The potential for detonation increases during sustained full-throttle operation; consequently, compression may need to be reduced for long race tracks or endurance type racing.
- Engine's with a long duration, high overlap cam or a cam with a late closing intake valve can tolerate higher compression ratios without incurring detonation.
- Deliberately over caming an engine with a long duration, high overlap cam will reduce the potential for detonation, but it will also reduce power at low and mid rpm.
- If a cam with a long intake duration and/or late closing intake valve is used, a higher mechanical compression ratio can help to compensate for the loss of dynamic compression at low rpm.
- Advancing an engine's cam (a maximum of 4 degrees) can increase cylinder pressure and torque in the lower rpm ranges.
- The mechanical compression ratio must be matched to the cam, induction system and combustion chamber design.
- Low compression engines generally perform better with a cam that has shorter duration, higher lift and an early closing intake valve.
- Higher compression speeds up the intake charge's burning rate, so slightly less ignition advance may be required.
- Raising the compression ratio may require a higher performance ignition system.
- It generally is better to run maximum ignition advance and sacrifice compression, than to run retarded ignition advance and higher compression.
- Compression ratio and fuel octane requirements are closely related. As the compression ratio goes up, so does the fuel octane requirement.
- Higher compression generally improves fuel economy.
- For a given combustion chamber design and engine combination, as compression is increased, fuel octane must also be increased.
- For a given fuel octane rating, dual plugs will allow the use of a higher mechanical compression ratio without incurring detonation; or for a given mechanical compression ratio, dual plugs will allow the use of a lower fuel octane rating.
- Increasing compression places additional stresses on the connecting rod, rod bearings and crank pin. It also increases piston and ring wear.
- Higher compression engines (greater than 10.5:1) may require an O-ring instead of a head gasket for a good seal between the cylinder and head.
- A high compression engine may require a high torque starter motor.



These Axtell 3-13/16 inch bore Evo racing cylinders are setup with an eight bolt head pattern and O-ringed for maximum sealing between the cylinder and head. Photo courtesy of Axtell Sales.

DETONATION TIPS

The potential for detonation increases under the following conditions:

- Compression is increased
- Ignition advance is increased (peak cylinder pressure and temperature increase)
- Combustion chamber turbulence is reduced
- Engine rpm is reduced (combustion time increases)
- Inlet pressure is increased
- Inlet temperature is increased
- Engine temperature increases
- Fuel atomization is reduced
- Flame propagation is reduced

CALCULATING COMPRESSION RATIOS

Winners don't win by accident. They win by proper planning, preparation and execution. Although the difference between winning and losing is governed by many factors, one major factor is precision engine assembly. Consequently, building a quality engine cannot involve guess work.

Deciding what compression ratio is consistent with the engine's objectives and verifying the ratio during final assembly is part of the engine building process winners go through. When racing in restricted classes, checking the engine's compression ratio and adjusting it according to plan can be the difference between being legal or illegal. It also can be the difference between winning or losing. With a street engine running on 92 octane pump gas, it can be the difference between experiencing no detonation or a high amount of detonation.

When you buy a set of pistons or a large displacement kit, a mechanical compression ratio is usually listed in the order catalog. However, when the parts are assembled, the actual compression ratio can easily range three quarters of a point or more on either side of the advertised ratio — and there is no guarantee that both cylinders have the same ratio. This is because the advertised ratio is based on an assumed volume for the combustion chamber and engine displacement. Yet there are many reasons why actual values can differ from assumed values.

For example, piston and combustion chamber tolerances can vary from cylinder to cylinder. The piston's valve reliefs may have been modified. Head gaskets and cylinder base gaskets can vary in thickness. Cylinder length and cylinder bore can differ from assumed values. Valves may be sunk deeper into the head, the shape of the valves can vary from stock or the combustion chamber may have been enlarged during previous head modifications. Additionally, starting in 1974 the Big Twin connecting rods are 7.440-inches in length, which is .030-inch shorter than 1973 and earlier rods. Some high-performance rods are only made in the short late model length. If a different rod length is used without an accompanying adjustment in cylinder length, the compression ratio will vary from that advertised.

Any of these conditions will affect the mechanical compression ratio. Determining your engine's true compression ratio not only will verify your legal for a particular race class, but also will prove whether you've achieved the compression ratio you want. Additionally, setting the compression ratio equal for both cylinders can potentially increase power by equalizing heat in the combustion chambers, besides smoothing the engine's operation.

In most cases, the compression for either cylinder will vary slightly due to one or more of the conditions listed above. When compression varies from cylinder to cylinder, combustion temperature and pressure also will vary between cylinders. The cylinder with the higher compression will tend to run hotter, given all other things are equal. The hotter running cylinder will tend to reach the point of detonation first. Knowledgeable tuners normally enrich the fuel mixture and/or retard the ignition timing to reduce the potential for detonation. However, doing this will penalize the power output of the cooler running cylinder. By checking the volume of each combustion chamber and then equalizing them, the compression ratio of each cylinder will be more balanced. Consequently, the engine can be tuned for greater power and it also will tend to run smoother.

To determine a cylinder's compression ratio, the combustion chamber volume and the cylinder displacement (or swept volume) must be known. Once these two values are determined, a relatively easy mathematical formula is used to calculate the mechanical compression ratio. The hardest part involves determining the exact volume of the combustion chamber.

An approximate value for combustion chamber volume can be mathematically calculated, but it is much easier and more accurate to fill the combustion chamber with a thin fluid to measure it precisely. This procedure is referred to as "cc-ing" the head. The term refers to the chambers being measured in cubic centimeters or cc's. CC-ing is easier and more accurate than mathematical calculations because the combustion chamber contains many unusual shapes.

The following formula is used for determining compression ratio:

$$= \frac{\left(\text{cylinder displacement} + \text{net combustion chamber volume}\right)}{\text{net combustion chamber volume}}$$

When figuring the compression ratio, it is best to work in cc's rather than cubic inches. To convert cubic inches to cc's, multiply cubic inches by 16.387.

Cylinder displacement is the displacement of one cylinder specified in cc's. Cylinder displacement is equal to:

$$\text{displacement} = \text{bore} \times \text{bore} \times .7854 \times \text{stroke}$$

If you visualize a piston positioned at the top of its stroke or TDC, the volume above it is *net combustion chamber volume*. (Refer to Figure 6.1) This volume is governed by the size and shape of the combustion chamber, the thickness of the head gasket, the piston's deck height and the mass of the piston dome, if any.

Adding cylinder displacement (swept volume) and net combustion chamber volume together gives *total volume*. To visualize total volume, picture the piston positioned at the bottom of its stroke, or BDC. The entire volume above the piston is total volume.

The ratio between total volume and net combustion chamber volume is the compression ratio.

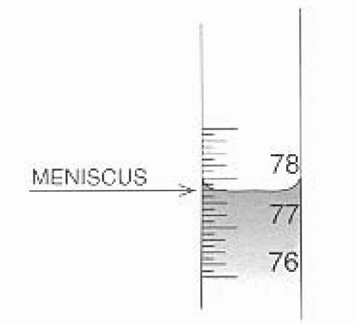
COMPRESSION MEASURING PROCEDURE

As stated previously, the most accurate method for determining mechanical compression ratio is to cc the engine's combustion chamber volume when the engine is assembled. For the Evolution engine, it is possible to cc only the head's gross combustion chamber volume and then use mathematical calculations to determine the compression ratio. However, this method does not guarantee the same accuracy as cc-ing the net combustion volume of an assembled engine.

In the case of the Shovelhead engine, due to its unique combustion chamber areas, the only practical method for accurately determining compression ratio is to cc the assembled engine's combustion chamber volume.

An engine cannot be cc-ed properly unless the flywheels are completely assembled, the pistons are fitted to the cylinders, cylinder head work is complete (a light coating of grease on the valve seats will ensure a good seal) and proper valve-to-piston clearance has been established. Also, it is extremely difficult, if not impossible to accurately cc an engine while it is mounted in a bike's frame because the spark plug hole must be positioned at the highest point of the combustion chamber.

READING THE BURETTE



Surface friction in a burette causes the fluid to crawl up the sides and form a crescent shape called a meniscus. For an accurate measurement, always read the fluid level at the lowest point of the meniscus.

The cc-ing procedure requires the use of either a plastic or glass burette. A burette is a graduated tube with a petcock located at its bottom end for accurately dispensing a measured amount of fluid. Most burettes are graduated in tenths of a cc, which is ideal for cylinder head cc-ing. One cc is equal to one milliliter (ml) in the metric system. Burettes come in a variety of sizes, but it is best to get a 100 cc size because a burette smaller than the chamber size requires refilling and this can introduce inaccuracies. However, if you're short on dollars a simple 60 cc surgical or veterinary syringe can be used.

For a measuring fluid, a thin colored liquid such as solvent and automatic transmission fluid, or solvent and a few drops of machinist's dye or food coloring works well. If you're competing in a class that may be inspected, it is best to use the same fluid that the tech inspectors use.

Take note that the fluid's surface tension in the burette causes the fluid to form a crescent shape called a *meniscus*. When reading the fluid level, remember to take each reading at the lowest point or the bottom of the *meniscus* to ensure accuracy.

CC-ING PROCEDURE—ENGINE ASSEMBLED

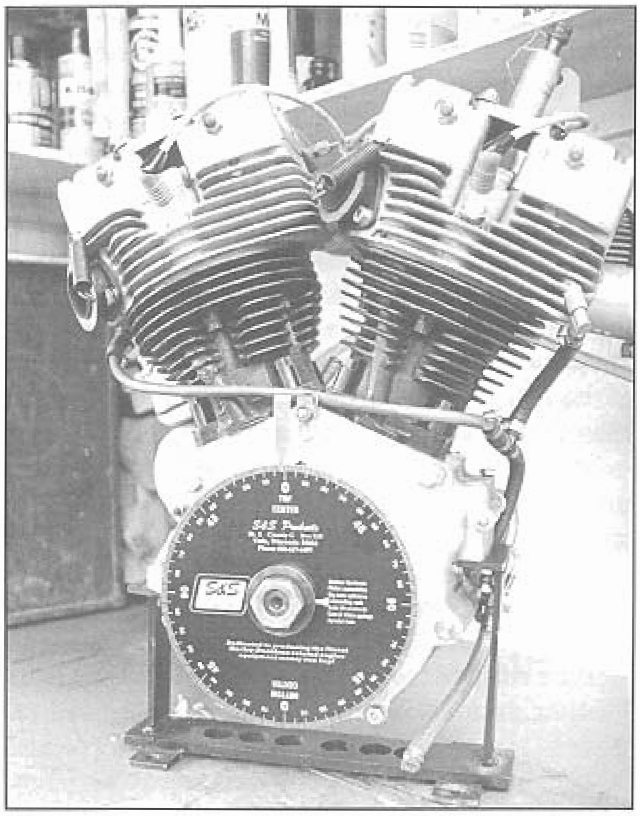
- Assemble the front piston (including the compression rings) to the connecting rod.
- Assemble the front cylinder using the same base gasket that will be used during final assembly.
- Bring the piston to TDC using a dial indicator or the positive-stop method. Modify a crankcase timing hole plug so that when it is gently screwed into the timing hole, it locks the flywheels in position.
- Seal the piston to the cylinder wall by applying grease to the crevice between the piston ring land and cylinder wall.
- Install the front head gasket or O-ring and the cylinder head. Tighten the head using all head bolts.
- Tilt the engine and shim it against a wall or workbench so the spark plug hole is positioned at the highest point of the combustion chamber.
- Fill a 100 cc burette with fluid and open the petcock to prime the drain tube below the petcock. Also, reduce the fluid level until the bottom of the *meniscus* is at the 100 cc mark.
- Fill the combustion chamber with fluid to the height of the second spark plug thread. Take your time and be precise.
- Record the number of cubic centimeters it took to fill the chamber. This value becomes the net combustion chamber volume. When reading the fluid level, remember to take each reading at the lowest point or the bottom of the *meniscus* to ensure accuracy.
- Determine the displacement of the cylinder in cubic inches using the following formula:

$$\text{displacement} = \text{bore} \times \text{bore} \times .7854 \times \text{stroke}$$

- Convert cylinder displacement to cubic centimeters by multiplying cubic inch displacement by 16.387. If you wish, you can eliminate this step by changing the constant .7854 in the previous displacement formula to 12.87 (.7854 x 16.387). Now the formula will give the result in cc's, instead of cubic inches.
- Calculate the compression ratio for the front cylinder by adding cylinder displacement in cc's to net combustion chamber volume in cc's and then divide the sum by the net combustion chamber volume.

$$CR = \frac{(\text{cylinder displacement} + \text{net combustion chamber volume})}{\text{net combustion chamber volume}}$$

- Now calculate the compression ratio for the rear cylinder by repeating the same steps.



The most accurate way to cc a Shovelhead combustion chamber is to assemble the engine, bring the piston to TDC and fill the chamber with fluid. Due to the Evolution's combustion chamber design, you can get about the same results without assembling the engine. Photo courtesy of S&S Products.

Once the compression ratios for each cylinder are calculated, they can be compared. If the ratio differs for each cylinder or is different from the desired compression ratio, the combustion chamber volume for one or both cylinders must be altered. Any alteration that increases the combustion chamber volume, decreases the compression ratio, while decreasing chamber volume increases the compression ratio.

ADJUSTING COMPRESSION

Combustion chamber volume can be changed by machining a cylinder shorter or shimming it longer. Also, a different thickness of head or cylinder base gasket can be used. Another method is to machine material off the cylinder head's gasket surface. Some engine builders change combustion chamber volume by sinking a valve deeper into the head. However, this can reduce airflow, change valvetrain geometry and consequently hurt performance.

If a discrepancy exists between each cylinder's combustion chamber volume, the displacement formula can be used to determine how much the height of one cylinder must be lengthened or shortened to even the volumes.

STOCK CYLINDER LENGTHS

Engine	Year	Cyl. Length
Shovelhead 74" & 80"	1966 to 1984	5.330"
Evolution Big Twin	1984 to Present	5.550"

Table 6.1

Also, if a discrepancy exists between the calculated compression ratio and the desired compression ratio, the compression ratio formula can be recalculated using a different value for the net combustion chamber volume. This will determine how much the chamber volume must be altered (either up or down) to arrive at the desired compression ratio. Then the displacement formula can be used to determine how much the cylinder length must be changed to provide the necessary change in combustion chamber volume.

When building a quality, high-performance engine, the combustion chambers should always be equalized. Always strive for an exact match, however, in no case should the volumes vary by more than 2 cc.

CYLINDER SHORTENING

For example, let's assume that 5 cc's must be removed from the combustion chamber volume to obtain the desired compression ratio. To determine how much the cylinder must be shortened to accommodate the 5 cc reduction in volume:

- First, convert 5 cc's to cubic inches. There is .06102 cubic inch for each cubic centimeter.

$5 \times .06102 = 0.305$ cu. in.

- Now insert the product 0.305 into the displacement formula.

$0.305 = bore \times bore \times .7854$

- Lets assume the engine's bore is 3.498-inches. The formula now appears as follows:

$0.305 = 3.498 \times 3.498 \times .7854$

$0.305 = 9.610$

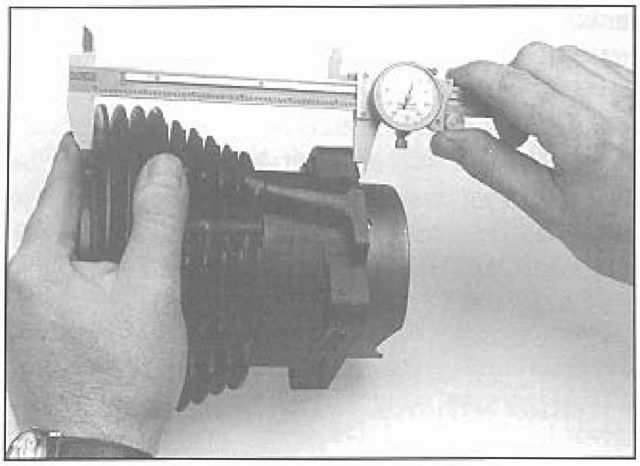
$= \frac{0.305}{9.610}$

$= .032$ inch cylinder height

The cylinder must be shortened .032-inch to achieve the required combustion chamber volume.

Most Evolution engine's ship with .060-inch thick head gaskets; however, some have shipped with 0.040-inch gaskets. Stock cylinder base gaskets crush down to about 0.015-inch. Bartels' Performance and Custom Chrome, Inc. offer head and base gaskets in various thicknesses.

Keep in mind, however, that a change in cylinder height may require a change in the length of other parts such as the intake manifold, pushrods, pushrod keepers and top engine mount. Also, piston-to-valve, piston-to-head and squish band clearance must be rechecked and maintained. Furthermore, the piston's top compression ring must not extend beyond the top of the cylinder bore and the oil ring must not extend below the oil return hole located near the bottom of the cylinder.



Most stroker engines use longer than stock length cylinders. Cylinder length affects piston deck height, intake manifold width, pushrod length and frame clearance. Measure the cylinder from head gasket surface to base gasket surface. Don't forget to include the thickness of any cylinder base plates. Photo courtesy of S&S Products.

HEAD MACHINING

One situation you might encounter is where you know what your mechanical compression ratio is, but you want to increase it. This can be accomplished by machining some material off the head's gasket surface. The question now becomes, how much do you need to machine for a specified increase in compression? The equation to accomplish this is based on the compression ratio minus 1.0. The formula is as follows:

$$= \frac{(new\,cr - 1) - (old\,cr - 1)}{(new\,cr - 1) \times (old\,cr - 1)} \times stroke$$

In this equation, "cr" equals compression ratio. Now, let's assume that the engine's current compression ratio is 8.5:1 and we want to increase it to 10.5:1. The engine's stroke is 4.25-inches. The formula now looks as follows:

$$= \frac{(10.0 - 1) - (8.5 - 1)}{(10.0 - 1) \times (8.5 - 1)} \times 4.25$$

$$= \frac{9.0 - 7.5}{9.0 \times 7.5} \times 4.25$$

$$= \frac{1.5}{67.5} \times 4.25$$

$$= 0.0222222 \times 4.25$$

$$= .094 \text{ inch to machine}$$

In this example, .094-inch of material must be machined from the head's gasket surface to raise the compression ratio from 8.5:1 to 10.0:1. However, this is too much material to take off the head's gasket surface. It would be better to machine a maximum of .060-inch off the head and compensate for the remainder by reducing the head gasket thickness or shortening the cylinder.

CC-ING PROCEDURE—ENGINE DISASSEMBLED

If the engine is disassembled, its compression ratio can be calculated by individually determining the following volumes: gross combustion chamber, piston dome, piston deck height, piston valve reliefs and head gasket. These values can then be used to calculate the mechanical compression ratio.

COMBUSTION CHAMBER VOLUME—GROSS

Gross combustion chamber volume must be known when determining the compression ratio for a disassembled engine. However, there are other times when its value can be helpful. For example, a valve job may have been performed on your heads, or you may have bought a set of used heads that were previously milled. In either situation, the combustion chamber volume will vary from stock (stock is about 78 cc for an Evolution Big Twin, excluding the head gasket, piston deck and valve relief volumes). Knowing how much the chamber volume varies from previously established values suggests the expected compression ratio change.

For the Evo, measuring the gross combustion chamber volume does not require the heads installed on the engine. Instead, a Plexiglas plate is used during the cc-ing process to determine the gross volume of the combustion chamber. For example, to cc the head's gross combustion chamber volume, a burette and a perfectly flat, 6-inch x 6-inch x 1/4-inch thick piece of Plexiglas will be needed. Additionally, a burette stand and clamp are handy to have, but optional.

The Plexiglas plate is used to seal the combustion chamber. It must be large enough to cover the entire combustion chamber with at least one inch overlap for sealing. A 1/4-inch hole is drilled into the Plexiglas plate about one inch from its edge. Counter sinking the hole on one side will help facilitate the filling process.

The cylinder head must have the valves and valve springs installed. During assembly, apply a light coating of grease to the valve seats for a good seal. Install the exact spark plug you run in the engine because plugs with a different heat range or manufacturer will vary the chamber displacement.

Position the head with the chamber facing up and one end of the head slightly raised. Spread a light coating of grease on the head's gasket surface and then press the Plexiglas plate on the gasket surface. Be sure to position the drilled hole (recess facing up) at the edge of the combustion chamber and locate it at the head's highest raised point. The grease will provide a seal between the head and the plate. Be careful because using too much grease will cause it to flow out into the chamber and affect the accuracy of the measurement.

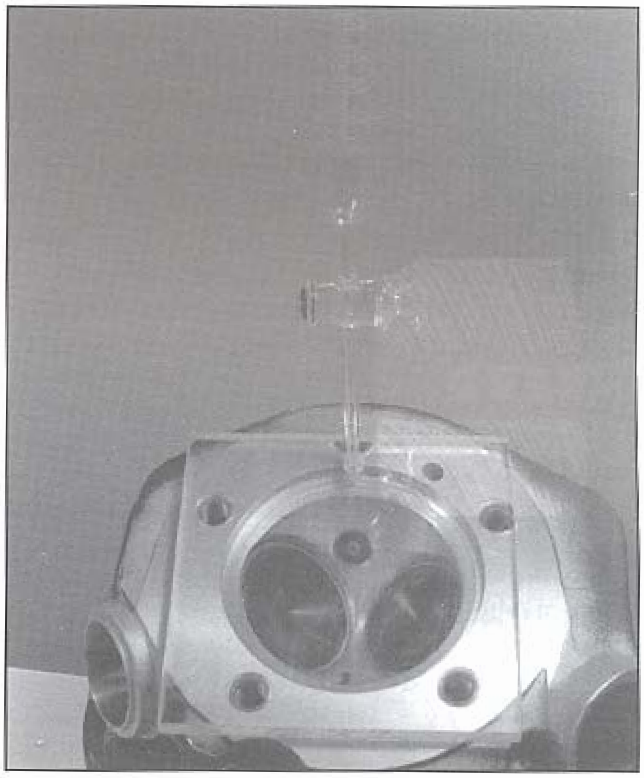
To measure the chamber, follow the same procedure for the burette as described under "CC-ing Procedure." Using a clamp and stand to hold the burette can make the process easier. Check for leaks at the spark plug area, in both ports and between the gasket surface and plate. Eliminate any air bubbles in the chamber by tapping on the Plexiglas plate. Stop filling the chamber when the fluid level reaches the bottom of the hole in the plate. Remember to read the fluid level in the burette at the bottom of the *meniscus*.

PISTON DOME VOLUME

Knowing the volume of the piston dome is helpful in many circumstances, especially if you have radiused the dome or modified the valve reliefs.

The easiest method for measuring dome volume is with the piston in the cylinder bore. With both compression rings on the piston, install the piston on the connecting rod and slip the cylinder over the piston. Move the piston toward the bottom of the bore and apply a coating of grease to the cylinder wall approximately 1-1/2 to 2-inches below the top of the cylinder. Next, measure the height of the dome.

For example, if the dome height measures .500-inch, slowly move the piston up the bore so the piston deck's flat surface is .750-inch below



To cc a combustion chamber, place the head on a slight angle and cover the chamber with a Plexiglas plate. Use grease to seal the plate to the head and place the plate's oil hole at the head's highest point. Use the same spark plugs you will run. Use a 100 cc burette to fill the chamber with thin fluid and take a reading at the bottom of the meniscus.

the top of the cylinder. The distance the piston is positioned below the top of the cylinder is arbitrary as long as it is greater than the dome height. Use a depth micrometer or dial indicator to measure the .750-inch. Check that the piston is square in the bore and then carefully remove any excess grease off the piston dome and cylinder wall. Now seal the Plexiglas plate to the cylinder's gasket surface using grease. Position the hole in the Plexiglas at the highest point of the cylinder. Next, fill the space above the piston's dome with fluid, following the same procedure used when cc-ing the combustion chamber and record the amount of fluid used.

To determine the dome's volume, you must first calculate the volume of the cylinder in cubic inches using the diameter of the cylinder bore and a height of .750-inch (only for this example), which is the distance the piston is positioned below the top of the cylinder. For example, if we assume the bore's diameter is 3.498-inches, the formula appears as follows:

$= 3.498 \times 3.498 \times .7854 \times .750$

$= 7.21$ cu. in.

Convert this value from cubic inches to cubic centimeters.

$= 7.21 \times 16.387$

$= 118.15$cc

Next, subtract the volume of fluid needed to fill the cylinder from 118.15 cc. The difference is the actual piston dome volume.

VALVE RELIEF VOLUME

The volume of the piston's valve reliefs has an affect on the compression ratio. The volume of both valve reliefs for a stock, flat dome Evolution piston is about 1 cc. If the reliefs are machined larger for added valve-to-piston clearance, the following procedure can be used to determine their volume. This procedure requires the use of modeling clay or something equivalent and only works if the piston dome is perfectly flat like a stock Evolution Big Twin piston.

Start by pressing a piece of modeling clay into the valve relief(s), making sure the relief is completely filled. Next remove excess clay by scraping the top of the piston with a metal straight edge. The clay should now be level with the top of the piston. Finally, remove the clay (using a sharp instrument) from each relief without compressing or expanding it.

Measuring the valve relief's volume requires filling a burette with fluid to a suitable level, such as 25 cc. Next, drop the clay impression(s) into the burette and observe how far the fluid rises. Remember, always take a measurement at the bottom of the *meniscus*. For example, if the fluid increased from the 25 cc level to the 28 cc level, the volume of the valve relief(s) is 3 cc.

HEAD GASKET VOLUME

For the Evolution engine, head gasket volume is required to determine net combustion chamber volume.

Assume the engine's bore is 3.498-inches and the compressed head gasket thickness is .040-inch. For the following formula, the constant 12.87 is derived from multiplying 0.7854 by 16.387. This will provide an answer in cubic centimeters rather than cubic inches. The formula will now look as follows:

$= bore \times bore \times 12.87 \times compressed\ gasket\ thickness$

Example:

$= 3.498 \times 3.498 \times 12.87 \times 0.040$

$= 6.2991$ cc, or 6.3 cc

PISTON DECK HEIGHT VOLUME

Piston deck height is the distance the piston is positioned above or below the cylinder's head gasket surface when the piston is located at TDC. A piston that is flush with the cylinder's gasket surface has a zero deck height. Most Evolution engines ship from the factory with about a zero deck height.

If the piston is below the top of the cylinder, use a depth micrometer or a feeler gauge to measure its position. If it is above, use a dial indicator or a feeler gauge for measuring. To measure, make sure the cylinder is clamped down securely, clean all carbon from the piston's dome and bring the piston to TDC. Ensure the piston is level in the bore and measure the piston's height at its outer circumference and in line with the piston pin.

The following formula is useful for determining the volume in cc's of an Evolution engine's piston deck height.

$= bore \times bore \times 12.87 \times piston\ deck\ height$

If we assume the cylinder bore is 3.498-inches and the piston deck height is .010-inch *above* the cylinder's head gasket surface, the formula appears as follows:

$= 3.498 \times 3.498 \times 12.87 \times 0.010$

$= 1.5747$ cc or 1.6cc

COMBUSTION CHAMBER VOLUME—NET

The easiest and most accurate way to determine *net* combustion chamber volume is to cc the combustion chamber of an assembled engine.

However, it can also be determined mathematically if the volumes of the following areas are known: gross combustion chamber, piston dome, piston deck height, head gasket and valve reliefs. The formula appears as follows:

$$= \text{gross combustion chamber volume} - \text{dome volume} - \text{piston deck height volume} + \text{head gasket volume} + \text{valve relief volume}$$

For the next example, assume the following: gross combustion chamber volume is 78 cc; a flat dome piston is used, so dome volume is zero; piston deck height is .010-inch above the top of the cylinder for a 1.6 cc volume; head gasket volume is 6.3 cc; and valve relief volume is 1.0 cc. This results in the following formula:

$$= 78 - 0.000 - 1.6 + 6.3 + 1.0$$

$$= 83.7 \text{ cc net combustion chamber volume}$$

Remember that gross combustion chamber volume is the measured volume of only the chamber and does not consider the piston dome, piston deck height, head gasket or valve reliefs.

The stock gross combustion chamber volume for an Evolution Big Twin is approximately 78 cc's (this does not include head gasket, piston deck height or valve relief volumes). For an exact value, use a chemistry burette and fluid to measure your head's chamber volume.

The previous example uses a piston with a flat dome, so dome volume is zero. With a raised dome piston, *subtract* the dome's volume. With a reverse (dished) dome, *add* its volume to the chamber volume.

When computing net combustion chamber volume, the piston is assumed to be positioned at Top Dead Center (TDC). In the previous example, the piston's deck height is .010-inch above the gasket surface when positioned at TDC. This requires the piston's volume above the gasket surface to be *subtracted* from the chamber volume. If the piston is positioned below the cylinder's gasket surface, its volume below the gasket surface is *added* to the combustion chamber volume. A piston that is even with the gasket surface has a *zero* deck height. In general, for each .010-inch a stock bore (3.498-inches) Evolution piston is above or below the gasket surface, either subtract or add 1.6 cc volume from the combustion chamber volume.

Valve relief volume is added to the combustion chamber volume when a flat top piston is used. For a flat top Evolution piston, each valve relief is usually between 0.5 and 1.0 cc in volume.

For a Shovelhead, the easiest and most accurate method to determine net combustion chamber volume is to measure the chamber's volume with a burette filled with fluid while the head is mounted to the cylinder and the piston is at TDC.

COMPRESSION RATIO

Once the net combustion chamber volume is determined, its value can be placed in the following formula to determine the mechanical compression ratio:

$$= \frac{\left(\text{cylinder displacement} + \text{net combustion chamber volume}\right)}{\text{net combustion chamber volume}}$$

Let's assume the engine's single cylinder displacement is 40.85 cubic inches or 669.4 cc's and its net combustion chamber volume is 83.7 cc. The formula now looks like this:

$$= \frac{669.4 + 83.7}{83.7}$$

$$= \frac{753.1}{83.7}$$

$$= 9.0{:}1 \text{ mechanical compression ratio}$$

EVOLUTION CAPACITIES

- 669.4 cc = Cylinder displacement of stock 81.7 cubic inch engine (3.498" bore by 4.250" stroke).
- 78.0 cc = Stock Evolution Big Twin combustion chamber (this does not include head gasket, piston deck height or valve relief volumes).
- 9.45 cc = .060" compressed head gasket.
- 6.30 cc = .040" compressed head gasket.
- 2.36 cc = .015" compressed cylinder base gasket.

- 1.00 cc = Two stock valve reliefs.
- 1.6 cc = The approximate volume for each 0.010-inch the piston's deck height is above or below the top of the cylinder. Most engine's are shipped from the factory with a zero deck height.
- 1.50 cc = The approximate volume for each 0.010-inch material machined from the head gasket surface of a stock Evolution Big Twin cylinder head.
- 8.57:1 CR = Stock chamber volume, .060-inch head gasket, two valve reliefs and zero piston deck height (88.45 cc total).
- 8.85:1 CR = stock chamber volume, .040-inch head gasket, two valve reliefs and zero piston deck height (85.3 cc total). ❖

Huck Cameron on his 108 ci Pro Gas Evolution has run a 5.76 E.T. at 116.75 mph in the 1/8 mile. Photo courtesy of Larry Smith/ Handcrafted American Racing Motorcycles.

Chapter 7

Exhaust System

An Exhausting Subject

It's been said that the engine is nothing more than an airpump.

We already know that more power can be made by improving the pump's breathing capability through induction system modifications. However, the pump also can be improved by the efficient extraction of exhaust gases. Consequently, a free flowing exhaust system plays a vital role in moving air through the engine and it is critical for increasing horsepower.

The stock Big Twin engine suffers significant horsepower losses due to a very restrictive exhaust system because the factory has to compromise performance for the sake of quiet operation. Street riders wanting a high-performance exhaust system are faced with other compromises. They want a great looking pipe that isn't too loud, yet one that performs in the low and midrange where they do most riding. Racers, however, are not concerned with noise. Instead they want the broadest power curve possible through the upper rpm range where their engine spends most of its time.

Street riders and racers both want maximum power from their exhaust system, but at a different rpm. Nevertheless, the same exhaust factors control their engines' power output: backpressure, gas velocity and energy pulse timing. Not only do these factors determine how much power the engines make, but also where in the rpm band the power occurs.

Black and white answers defining the perfect exhaust system are hard to come by. For example, an exhaust system that produces high power with one engine, can be down in power with another engine. For a "happy" engine, the key requirement is to match the exhaust system to the engine combination and rpm band. With complex principles involved and many factors entering into the equation, it is impossible to determine the perfect exhaust system without some cut-and-try testing.

The exhaust system is an integral part of the components that regulate airflow through the engine — the others being the carburetor, intake manifold, intake and exhaust ports, and camshaft. For maximum performance, each of these components must be tuned together as a system to a specific rpm band. Whenever any one component is changed or altered, the entire group must be retuned as a system for maximum performance.

EXHAUST FUNDAMENTALS

Before covering exhaust systems and their applications, a fundamental understanding of the elements involved and their relationships should be helpful. The elements include: backpressure, blowdown, mixture dilution, valve overlap, scavenging, volumetric efficiency, reversion and mean flow velocity.

Backpressure is the flow resistance created within the exhaust system that reduces net exhaust flow somewhere within the engine's rpm band. The diameter of the header pipe affects backpressure as does obstructions in the system such as muffler baffles. Although the intent of a performance exhaust system is to reduce backpressure, all engine combinations require some level of backpressure to give maximum performance. Different carburetor and cam combinations require various levels of backpressure for smooth tuning over a broad rpm range.

Blowdown pertains to how efficiently combustion residue is expelled from the cylinder by expanding gases and it is controlled in part by when the exhaust valve is timed to open. Initially the high pressure of the hot expanding combustion gases near the end of the power stroke cause evacuation (blowdown) of exhaust. Blowdown causes most gases to exit the cylinder before the piston reaches BDC. Then the upward movement of the piston displaces any remaining gases during the final stages of the exhaust stroke.

For maximum performance, the exhaust system must efficiently evacuate combustion byproducts from the cylinder during high rpm operation. Any combustion gases remaining in the cylinder after the exhaust cycle will contaminate the subsequent air/fuel mixture drawn into the cylinder. This will result in reduced power and fuel economy. Blowdown and backpressure work hand-in-hand in that higher backpressure can prevent the efficient evacuation of exhaust gases by blowdown. However, reducing backpressure increases blowdown efficiency. Also, the mean flow velocity of the exiting exhaust gases plays an important part in clearing the cylinder of combustion gases. For the most part, the exhaust header's inside diameter governs the mean flow velocity. Low exhaust gas velocity causes rough low speed running, reduces low-end power and causes the engine to come on the cam abruptly.

Mixture dilution reduces the net amount of combustion pressure on the piston because unburnable exhaust gases occupy combustion chamber space. Consequently, a given amount of burnable air/fuel mixture is displaced. Mixture dilution is caused by incomplete exhausting of the cylinder during the exhaust cycle. The higher the backpressure or the lower the blowdown effect, the greater the mixture dilution will be in the combustion chamber at the time of combustion. Also, the timing of the cam's overlap period and reversion are also factors at certain engine speeds.

The *valve overlap* period becomes involved in the exhaust process because it is at this time the intake and exhaust tracts come in direct contact with one another. Overlap is the period near TDC when the intake valve is opening and the exhaust valve is closing. During this period, the intake tract and exhaust tract have direct access to each other so exhaust system pressures and pulses directly affect the carburetor and intake tract. An improperly tuned exhaust system can inhibit the incoming air/fuel mixture flow into the cylinder, while an excessively efficient system can over-scavenge the cylinder and draw fresh, unburned air/fuel mixture out of the cylinder and through the exhaust.

Scavenging is the process where a column of fast moving exhaust gases (inertia scavenging) or supersonic energy pulses (wave scavenging) helps extract combustion gases from the cylinder and assists fresh air/fuel mixture into the cylinder.

Volumetric efficiency (VE) refers to how completely or efficiently the engine's cylinders are being filled. VE is normally expressed as a percentage with 100 percent being the maximum fill for a normally aspirated engine without the help of tuned intake or exhaust systems. Unmodified engines usually range between 70 and 85 percent cylinder fill, while highly tuned engines can reach up to 115 and even 130 percent fill. The design of a given exhaust system will affect the cylinder's VE. An exhaust system with good scavenging capabilities for a given engine combination will increase a cylinder's VE.

Reversion describes the backward flow of exhaust gases into the cylinder and intake port due to low exhaust velocity in the header. Reversion normally occurs at low rpm when using a long overlap cam. Wave scavenging energy pulses that are out of synchronization with the cam's overlap period are another cause of reversion. Signs of carbon build up in the intake port are indications of reversion. Reversion can be minimized by high velocity exhaust ports, anti-reversionary header designs, exhaust system tuning and induction tract tuning.

Mean flow velocity is the speed at which exhaust gases exit the exhaust system. The velocity of the exhaust gases affects cylinder scavenging and is associated with the rpm at which the engine produces peak torque. Engine displacement, rpm and exhaust header cross-sectional area (diameter) determine mean flow velocity through the exhaust system. Low velocity not only hurts high-end power, but also low-speed driveability.

Now that terminology has been covered, the next step is to tie everything together to understand how the elements relate to total engine performance.

MEAN FLOW VELOCITY

As the engine increases in rpm, the mean flow velocity of the exhaust gases will increase in a nearly linear fashion to and beyond the rpm point at which peak torque is produced. For a given engine displacement and rpm, mean flow velocity is a function of the exhaust header's inside diameter. The smaller the header's diameter, the sooner a given flow velocity will be reached. Conversely, the larger the header's diameter, the higher the rpm required to reach a given flow velocity.

Optimum exhaust system efficiency is usually achieved when mean flow velocity is between 280 and 300 feet per second. If the header's inside diameter is too small, gas velocity will increase to the point where friction between the gas and header wall will increase and thereby cause excessive backpressure. On the other hand, too large a header diameter reduces mean flow velocity below acceptable levels. This can reduce maximum power and cause poor low-speed throttle response and carburetion tuning problems. This is why large 2-inch diameter straight pipes frequently cause low-speed tuning problems with small displacement engines. A large header diameter merely means that a higher rpm will be needed to achieve optimum exhaust efficiency.

As a general rule, the following formula can be used to calculate mean flow velocity through a given port:

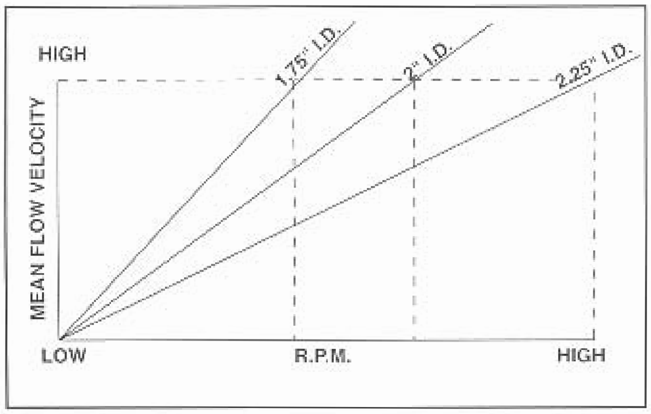
$$V = \frac{\text{piston speed}}{60} \times \frac{B^2}{d^2}$$

Where:

V = gas velocity in ft./sec.
B = cylinder bore diameter
d = header inside diameter
piston speed = ft./minute

To calculate piston speed, double the engine's stroke and divide by 12, then multiply by rpm. Now divide this ft./minute value by 60 to get ft./second.

Use this formula to calculate mean exhaust velocity at the middle or your rpm band. For example, if your engine's shift point is 6500 rpm and the rpm drops down to 4500 after an up shift, then 5500 rpm is the middle of the power band. You now would calculate mean flow velocity at 5500 rpm. As a general guide, any value much



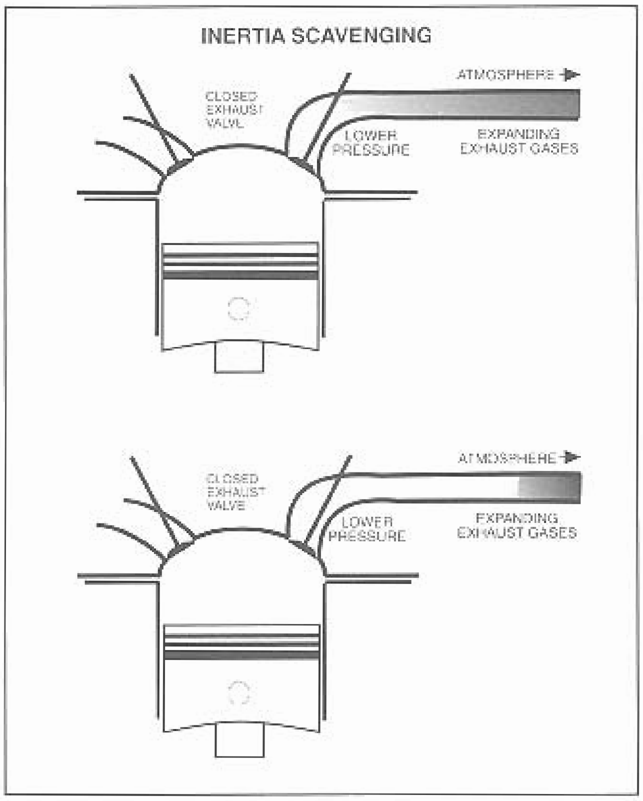
Mean flow velocity changes with pipe diameter and rpm. The larger the pipe diameter, the higher the rpm needed to reach a given flow velocity. This is why large diameter pipes have low flow velocity and scavenging capability at slow speeds, which leads to poor throttle response. Match header diameter to your engine combination and riding style.

less than 300 feet per second indicates the header's inside diameter is too large, and a value much greater indicates it is too small. Remember that 1-3/4 inch O.D. headers have a 1-5/8 inch inside diameter, while 2-inch O.D. headers have a 1-7/8 inch inside diameter.

INERTIA SCAVENGING

Inertia scavenging involves the actual physical movement of exhaust and intake gases. The fast moving column of exhaust gases possesses considerable inertia due to its speed. When the exhaust valve closes, the exiting exhaust gases do not stop. Instead, they keep moving due to inertia. The column's movement creates a vacuum (low pressure) behind it and under the exhaust valve. When the exhaust valve again opens, the vacuum helps extract the combustion gases from the cylinder. As engine speed increases, mean exhaust gas velocity also increases, therefore the scavenging effect increases.

Inertia scavenging is controlled primarily by header pipe diameter. For optimum scavenging, the header cross-sectional size must be small enough to keep the mean gas velocity as high as possible, yet large enough to keep backpressure at a minimum when running at high rpm. A large header diameter moves the engine's torque band to a higher rpm, while a smaller diameter moves it lower in the rpm range. If correctly matched to the engine combination and rpm range, open headers can make significant use of inertia (momentum) scavenging.



When the exhaust valve seats, it closes off one end of the header. The expanding exhaust gases continue flowing down the header to the atmosphere and leave behind a low pressure area that extends from the valve to the end of the header. When the exhaust valve again opens, the combustion gases exit into a low pressure area. This improves scavenging by purging more combustion residue from the chamber and assists the new intake charge in filling the cylinder during valve overlap.

WAVE SCAVENGING

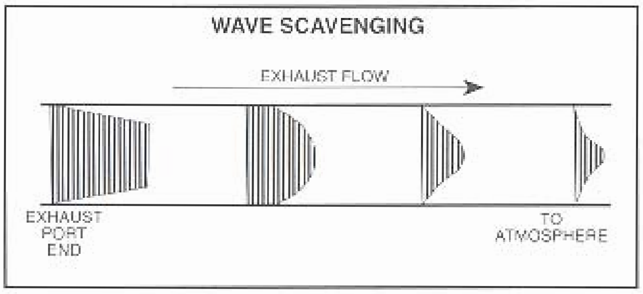
Header length also plays a major part in exhaust system scavenging. As the exhaust valve opens, hot expanding exhaust gases rush past the exhaust valve and into the header at 200 to 300 feet per second. This creates a positive pressure energy wave that moves toward the pipe's open end. When the positive wave reaches the open atmosphere, it expands and sends a negative wave back toward the combustion chamber. While the engine is running, alternating positive and negative waves continue to resonate through the exhaust system.

To maximize torque at a given rpm, the length of the header pipe can be adjusted so the negative wave arrives back at the combustion chamber during the valve overlap period (remember this is the period when both valves are slightly open). The negative wave is surrounded by a low pressure area that helps scavenge residual exhaust gases out of the combustion chamber and helps the fresh air/fuel mixture flow past the intake valve into the cylinder. This principle is commonly referred to as wave scavenging. If the energy wave is timed properly, exhaust scavenging is improved and the cylinder's volumetric efficiency is increased.

Although exhaust gases flow at 200 to 300 feet per second through the header, the positive and negative energy waves move at about the speed of sound, which is near 1700 feet per second at the temperatures and pressures present in the header. At this speed, it takes less than two thousandths of a second to travel the length of a 36-inch long header pipe. This discussion uses 1700 feet per second as an approximate speed for the energy waves. However, depending on the temperature of the exhaust gases, the wave's actual speed can range between 1700 and 2100 feet per second.

For example, let's say the negative energy wave was timed to arrive at the combustion chamber during valve overlap when the engine was turning 5500 rpm. However, at slower engine speeds, each revolution of the flywheels takes longer, but the energy wave still moves at about 1700 feet per second. This results in the negative wave arriving before the overlap period starts and this wastes the potential benefit of improved scavenging and increased VE.

At some low engine speed, the positive wave arrives back at the combustion chamber during overlap. This forces the exiting exhaust gases back into the combustion chamber, which dilutes the fresh air/fuel charge. The wave then



Wave scavenging uses energy pulses that reverberate through the exhaust system to improve exhaust scavenging and cylinder filling. By tuning the header's length, negative energy pulses can be timed to reach the combustion chamber during the valve overlap period, thereby improving volumetric efficiency.

continues pushing the diluted charge back into the intake port, manifold and finally through the carburetor venturi. As the diluted charge moves backward through the carb, it picks up fuel because the carburetor flows fuel regardless of which direction gases move through it. When the next intake stroke begins, the diluted mixture again moves through the carb's venturi picking up more fuel. Now the excessively rich fuel charge is drawn into the cylinder, resulting in poor throttle response and lost torque.

When engine speed is reduced even more, it takes so long for each flywheel revolution that a second negative wave reaches the combustion chamber at the optimum time — during valve overlap. This generates a secondary torque peak at about one-half to two-thirds of the peak torque rpm. This torque peak is especially noticeable when long overlap cams are combined with highly tuned exhaust systems.

Headers can be tuned for optimized wave scavenging by changing their length, since the energy wave's speed remains relatively constant for a given temperature (about 1700 feet per second). To maximize power at high rpm, header length is normally shortened because there is less time between each cylinder firing. Conversely, to optimize power in the lower rpm ranges, header length is increased because there is more time between each cylinder firing. The range over which the headers are tuned only spans a maximum of 1500 to 2000 rpm, so if you tune for top-end power, bottom-end power is hurt. The opposite holds true if you tune for bottom-end power. Header length is normally tuned for the middle of the engine's rpm band.

COLLECTORS

Some exhaust systems are designed with a collector tube that both primary header pipes terminate into. A collector provides a common plenum for the two primary headers and works on the principle of inertia scavenging instead of wave scavenging. With a collector, as the exhaust valve opens, exhaust gases speed down the exhaust pipe at about 300 feet per second. This leaves behind an area of low pressure that expands as the exhaust gases escape. By the time all the exhaust gases reach the collector, the entire header pipe for that cylinder is filled

A two-into-one collector uses the low pressure area created by the exiting exhaust gases from one cylinder to help scavenge the other cylinder. Shown is the Thunderheader from Rich Products.

with a low pressure area. When this low pressure area reaches the collector area, it spills over into the primary header pipe for the other cylinder. This reduces the pressure the exhaust gases encounter when exiting the pipe.

Ordinarily this pressure would be equivalent to ambient atmospheric pressure (roughly 15 psi). Normally aspirated engines have about a 15 psi advantage on the induction side and a 15 disadvantage on the exhaust side. If the collector cuts the 15 psi disadvantage in half, the lower pressure will enhance the exhaust scavenging effect. This will help draw fresh air/fuel mixture from the intake port into the cylinder, which results in improved volumetric efficiency.

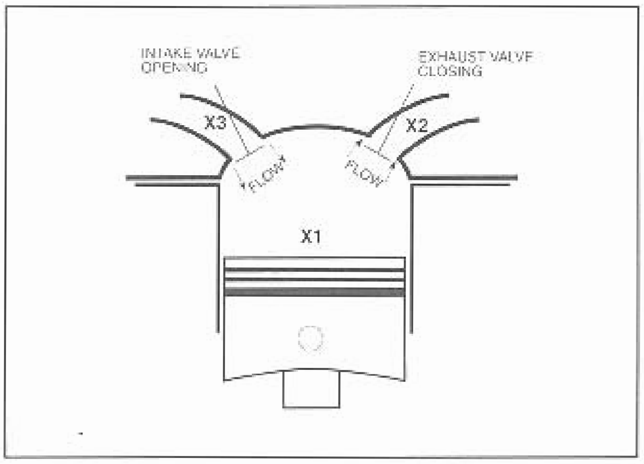
It is more difficult to get collectors to work with a V-Twin engine design than with a V8 or in-line four because of the uneven 315 and 405 degree firing pulses. As a result, some collector-style exhausts use uneven primary pipe lengths to help compensate for the V-Twin's unequal firing pattern. Collectors are tunable, both in length and diameter and primarily affect the amount of torque an engine produces below its torque peak. To a point, the larger the volume of the collector, the more the torque range is spread out and the more low-end torque the engine will produce.

ANTI-REVERSION TUBES & CHAMBERS

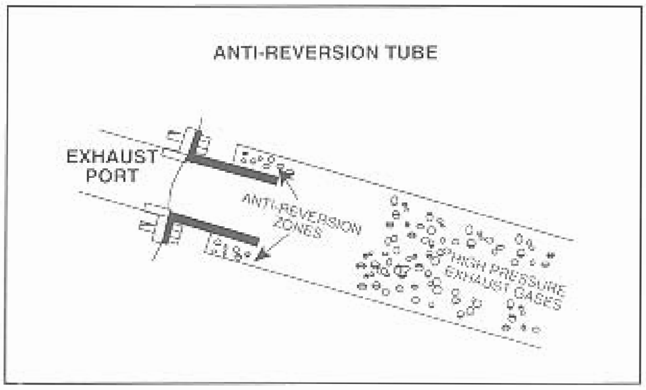
The flow of exhaust gases back into the cylinder, past the intake valve and into the intake tract during valve overlap is the phenomenon known as reversion. Reversion is caused by slow moving exhaust gases on the short-side radius or bottom side of the port. These slow moving gases can more easily backflow into the chamber than the faster moving gases on the port's long-side radius, especially when the piston is near TDC and the engine's rpm is low. The backflow dilutes the intake charge and disrupts carburetion. Large, low velocity ports encourage reversion. Reversion is the cause of the fog or fuel "standoff" that frequently is seen at the carburetor's inlet.

An *anti-reversion tube* can be used to reduce reversion by making it more difficult for exhaust gases to backflow into the combustion chamber. The tube acts as a one way valve and is located inside the primary header pipe where the pipe mounts to the cylinder head. The tube generally varies in length from one to four inches. The header pipe normally is enlarged about 1/4 to 3/8-inch in diameter where the anti-reversion tube is located.

An *anti-reversion chamber* works on the same principle as the tube, except it is welded in-line to the header pipe. The chamber creates a bulge in the header and can be positioned anywhere throughout the header's length, although it is



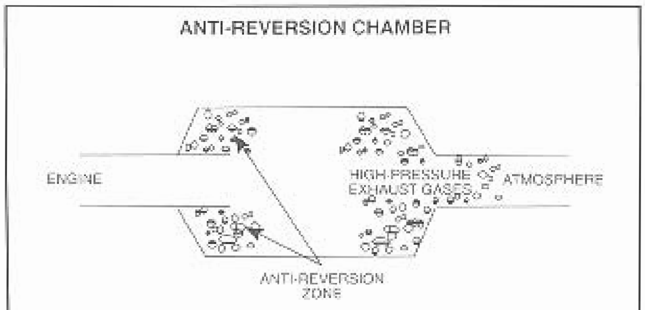
When the piston is near TDC and moving very slowly, slow moving exhaust gases on the port's floor can backflow into the combustion chamber and intake port. This dilutes the intake charge and disrupts carburetion. High mean flow velocity minimizes reversion. Reducing pressure at X1 minimizes reversion into the intake port and maximizes VE.



An anti-reversion tube is designed to limit the backflow of exhaust gases into the cylinder by acting as a one way valve. It consists of a short tube placed in the header near the exhaust port and can have a significant affect on low speed performance.

generally placed close to the cylinder head. With anti-reversion tubes, it is sometimes necessary to use an adapter plate to mount the header to the cylinder head because the pipe's larger diameter at the cylinder head end intrudes into the area required for the mounting bolts. An anti-reversion chamber eliminates this problem since it can be located away from the header mounting flange.

Anti-reversion tubes and chambers permit unrestricted gas flow out of the exhaust system. However, they provide considerable restriction to unwanted reverse flow of combustion gases into the intake tract, which can occur when gas velocity is low and inertia scavenging has insufficient energy to help cylinder filling during valve overlap. Reversion usually occurs at low engine speeds, especially when a large diameter header pipe is used in conjunction with a long overlap cam.



An anti-reversion chamber works on the same principal as the anti-reversion tube. Instead of being located near the cylinder head, the chamber can be located anywhere in the header. This makes for easier mounting of the header to the cylinder head.

Small header diameters maintain high mean gas velocity, which increases inertia scavenging for good low and midrange performance. However, small diameters also generate significant backpressure at high-rpm. On the other hand, headers with a large cross-sectional area reduce backpressure and improve high-rpm power, but low-rpm scavenging is reduced due to low mean gas velocity. Consequently, low-rpm torque, throttle response and gas mileage are reduced.

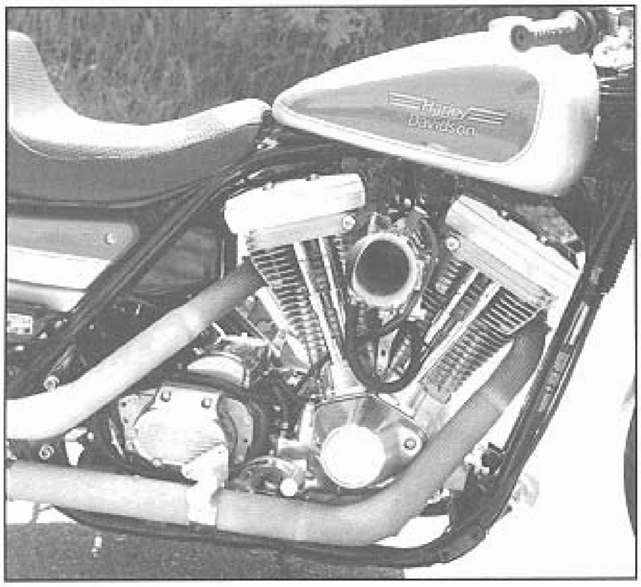
For the most part, a header with an anti-reversion tube or chamber is less sensitive to large pipe diameters. It is sometimes found that an anti-reversion header allows the use of a larger diameter header pipe for strong top-end performance without a resultant loss in low-speed power.

Another method for combating reversion is to use a header pipe slightly larger in diameter than the exhaust port. This will create a one way step where the port and header join, which will reduce the backflow of exhaust gases.

STEPPED HEADERS

Through the placement of small 1/8-inch variations (steps) in the header pipe, exhaust gas speed (inertia scavenging) and energy wave pulses (wave scavenging) can be slightly altered to optimize scavenging. Stepped pipes are frequently based on a three-section design with the pipe's outer diameter usually increasing in size starting from the cylinder head.

As explained under wave scavenging, the cylinder's volumetric efficiency can be increased by timing the header's negative energy wave to arrive at the combustion chamber during valve overlap. The negative pressure generated by the negative energy wave can be prolonged by additional negative energy waves created by the steps in the pipe. Each change in the pipe's cross-sectional area creates a pulse. This increases scavenging and improves VE. However, the negative energy wave created by each step is of smaller amplitude than the one generated by the header's open end because each pulse absorbs some energy, but not all of it. As the negative waves generated by the step travel toward the combustion chamber, they are followed closely behind by the wave generated at the pipe's open end. The multiple waves prolong the negative



Stepped headers create more energy pulses and can improve cylinder filling. These straight pipes are wrapped with braided ceramic insulation for greater heat retention and higher gas velocity. If you look closely, you can see the pipes have three steps under the wrap. Frank Rayburn on the Neidengard's Cycle Center 80 ci Evo holds the AHDRA E Street 1/8 mile record at 6.67 E.T. and 105.10 mph. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

effect in the pipe, which increases cylinder filling and may even allow a longer overlap period without any ill effects.

Stepped headers are most often divided into three sections, however, more or less sections can be used. Each successive section is usually 1/8-inch larger in diameter than the previous one. The smallest diameter section is located at the cylinder head mounting flange. As a starting point for length, the section closest to the cylinder head should be about 25 percent of the total header pipe length, the middle section 32 percent and the last section 43 percent.

A multiple section design allows tuning the header's length for optimum wave scavenging while reducing backpressure through each successively larger step. While header diameter sets the location of the engine's torque band, steps in the header augment the band. For example, if a 1-3/4 inch O.D. header pipe sets the engine's power band between 5000 and 6500 rpm, then a three-step header using 1-3/4, 1-7/8 and 2-inch O.D. steps would extend the torque band about 400 rpm on the top-end. The resultant band would now range from 5000 to 6900 rpm.

HEAT RETENTION

Inertia scavenging works by using the momentum of high velocity exhaust gases to create a low pressure area in the pipe. The energy contained in the exhaust gases is derived from combustion heat. The more heat that can be retained in the gas until it is discharged into the atmosphere, the more efficiently the header system will function.

Exhaust gases exit an Evolution's combustion chamber at approximately 1350 to 1450 degrees F. The Shovelhead's exhaust gas temperature is even hotter because its chamber is less efficient than the Evolution's. As the gases leave the combustion chamber, heat from the gases radiates from the exhaust pipe's surface. Heat radiated away from the engine results in lost power. This is because as the gases cool they lose velocity. As velocity is lost, the efficiency of inertia scavenging is reduced. Since heat generates velocity, retaining as much heat in the header pipe will keep flow velocity high, lower backpressure and increase the efficiency of inertia scavenging. Keeping the velocity high may allow slightly less exhaust cam timing and reduced overlap. This may help cylinder filling while increasing the cylinder's working pressure.

Exhaust insulating wrap is available in rolls, tubes and sheets to cover header pipes. The wrap is usually woven or braided from either ceramic or fiberglass material and has proven effective in retaining heat in header pipes. This improves scavenging and can allow the use of a less aggressive exhaust cam profile with less overlap for a stronger low and midrange without a reduction in high-end power.

EXHAUST SYSTEM DESIGNS

Exhaust systems for the Big Twin can be divided into two major categories: 2-into-2 and 2-into-1 designs. The 2-into-2 means that the two individual header pipes do not merge. Instead, each remains separate and has its own outlet. Stock staggered duals and standard straight pipes are examples of 2-into-2 systems. Systems that connect the header pipes from both cylinders into one common collector are referred to as 2-into-1 systems. SuperTrapp and others manufacture 2-into-1 systems.

Both categories of systems can include mufflers. Some systems use a one piece design in that the header pipe and muffler cannot be separated, while others use slip-on mufflers for easy removal.

One key factor for making high horsepower with mufflers in the upper rpm range (beyond 4000 or 4500 rpm) is to have sufficient muffler volume. Most performance modifications for the Big Twin engine involve increasing its breathing ability, either through more efficient induction, larger displacement or a combination of both. This results in a large volume of exhaust gases that has to be expelled from the cylinders. Small capacity mufflers, regardless of their design, cannot meet the flow demands of even an 80 cubic inch engine without introducing considerable power robbing backpressure. As a result, small volume, good looking mufflers or baffles inserted into small diameter straight pipes severely restrict flow, increase backpressure and limit top-end horsepower. So remember that there is a given tradeoff between high-performance and the ideal custom look.

Large "dresser" model Big Twins ship from the factory with relatively large volume mufflers that include a crossover pipe. These mufflers flow better than the mufflers on smaller Big Twin models. In fact, when comparing each Big Twin's factory specifications, the "dressers" have between a one and 2-1/2 foot pound engine torque advantage (depending on year) over other models. The torque advantage can be directly attributed to large volume mufflers.

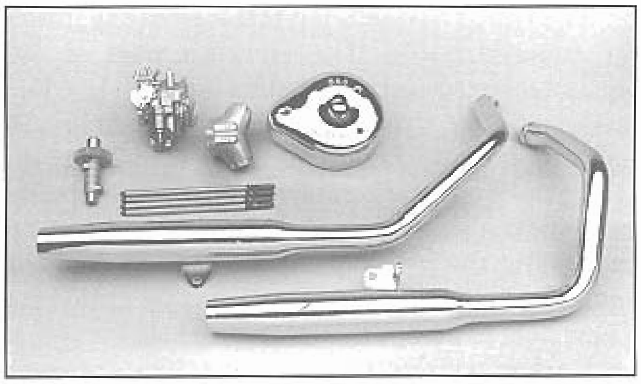
Stock mufflers substantially restrict exhaust flow, which increases backpressure and reduces scavenging. To minimize restriction, Big Twins are equipped with a crossover pipe that connects the headers of both cylinders. This equalizes exhaust pressure, increases the effective muffler volume for each cylinder and adds torque throughout the engine's rpm range. When running small volume mufflers, it is helpful to retain the crossover pipe.

Straight pipes use a 2-into-2 design that helps scavenge exhaust gases from the cylinder and assists the induction system with cylinder filling. For optimum performance, straight pipes are tuned by changing their length and diameter. Years of racing have proved that straight pipes provide maximum performance for the V-Twin engine, but not necessarily over the entire rpm range. Straight pipes are effective primarily above 4000 rpm and frequently cause low and midrange carburetion tuning problems on stock and small displacement engines. Two inch diameter straight pipes are great looking on many bikes; however, from a performance standpoint (with few exceptions), they should be limited to about 98 cubic inch and larger engines and very high rpm small displacement engines.

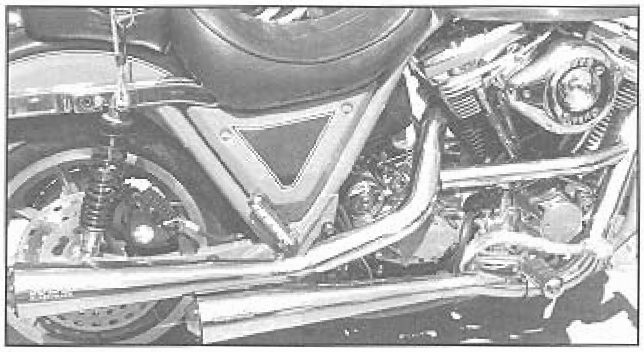
SYSTEMS with MUFFLERS

Mufflers reduce noise by restricting, absorbing and reflecting or by a combination of these techniques. The *restriction* method works by forcing the exhaust gases through a baffle that is perforated with small-diameter holes or a louvered core. The *absorption* technique uses a fiberglass type of packing material that converts the sound entering the muffler into heat as it interacts with the packing material. This technique generally creates less backpressure than restriction, but also produces less sound suppression. The *reflector* method reduces sound by reflecting sound waves back toward the inlet end of the muffler.

Big Twin mufflers frequently combine multiple techniques to reduce sound while minimizing backpressure. When choosing an exhaust system, not only consider increased performance, but also the system's sound quality, styling and fit.



Staggered duals retain the classic Harley look and are proven performers. These pipes from Carl's Speed Shop can be tuned for low noise on the street or maximum performance on the race track. Photo courtesy of Carl's Speed Shop.



These slip-on mufflers from SuperTrapp are tunable by adding or removing diffuser disks. Randy Diamond's stock stroke 97 ci Evo retains the header crossover pipe and uses SuperTrapp open end caps for more power and sound. Photo courtesy of Great Lakes Cycle.

Various companies offer exhaust systems with mufflers for the Big Twin. BUB Enterprises, Carl's Speed Shop and Cycle Shack are a few that offer staggered dual systems with either slash-cut or tapered ends. For FX models, Harley-Davidson offers a 2-into-2 Sport Exhaust System with tapered ends that is made by Kerker. Harley also offers slip-on off-road mufflers for FX models and slip-on performance touring mufflers for "dresser" models. Kerker, who is now affiliated with SuperTrapp, offers a 2-into-2 system with tapered ends. SuperTrapp offers 2-into-2 and 2-into-1 systems. They also offer separate slip-on mufflers for stock header pipes. Rich Products has a 2-into-1 anti-reversion system. These systems offer relatively high-performance with a reasonable sound level. The 2-into-1 systems with their collector design generally increase torque in the lower rpm range while the 2-into-2 systems provide greater muffler volume and perform well on the top-end.

The Carl's Speed Shop pipes have one internal tapered baffle. The smallest part of the baffle's tapered end has about a 3/8-inch I.D. For increased flow, this end can be shortened in 1/8-inch increments to a maximum of about 2-inches.

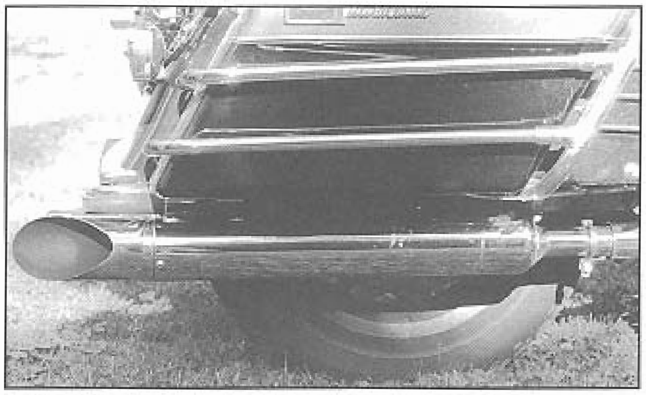
Cycle Shack mufflers have one removable baffle secured by a single bolt. Flow can be increased by shortening the baffle about 4-inches at its front end. The baffle and pipe then must be drilled for a new hole to accept the mounting bolt.

Kerker systems usually have three internal baffles wrapped in fiberglass. The inner most baffle, referred to as the "C" baffle is extremely restrictive and can be eliminated by removing the entire baffle set and then drilling out the pop rivets that attach the solid "C" baffle to the perforated "A-B" baffle set. This modification can increase airflow through the system from 60 cfm to about 200 cfm.

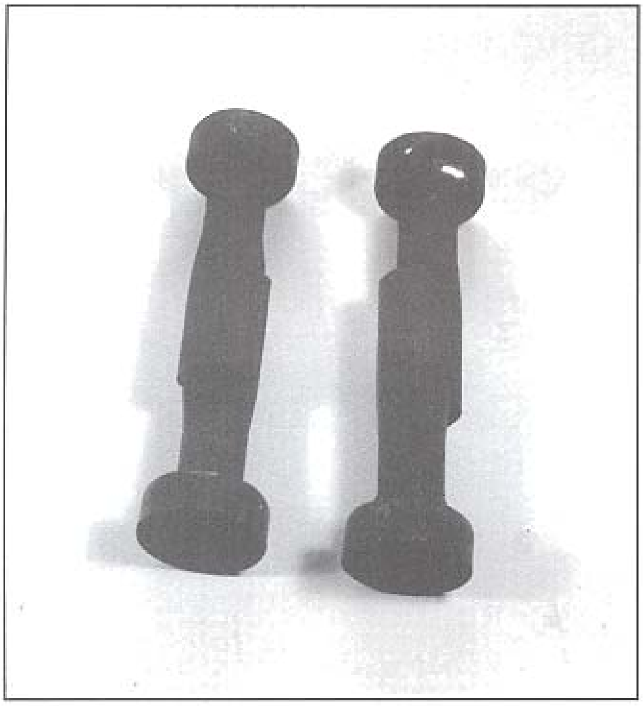
The SuperTrapp exhausts use 3-inch internal diffuser disks stacked in the muffler's tip and fiberglass packing for silencing. Adding or removing disks allow control over the amount of backpressure and the noise level. Adding more disks reduces backpressure but increases noise. Reducing the number of disks reduces noise, but increases backpressure. Twelve to 21 disks are normally used, but as many as 24 have been used with good results. Fifteen to 18 disks often give good results on the street without too much noise. Eighteen to 21 disks frequently give maximum power. For increased power and noise on the race track, open end caps can be used.

The Harley-Davidson slip-on off road mufflers work well with stock header pipes and provide a good low-dollar performance exhaust system.

Harley-Davidson's slip-on Touring Mufflers (made by Kerker) include the same "A-B-C" baffle design as described above along with a choice of end cap design. These mufflers fit "dresser" models and have a large volume. They sound good and can be modified as described above to flow very well. SuperTrapp makes a turn-out muffler that is similar in size and shape to the Touring Mufflers, but it incorporates diffuser disks instead of baffles for noise control. Both muffler sets improve performance.



These slip-on touring mufflers are large in volume and offer different end cap designs. The large volume improves exhaust flow while keeping noise down. It also is the reason the touring bikes have more torque than the models with smaller mufflers. The Harley/Kerker touring mufflers use the A-B-C baffle set for silencing. The SuperTrapp version uses diffuser disks.



Shown are the "C" baffles used in the Kerker touring mufflers. For racing, you can increase exhaust flow over 300 percent by drilling out the pop rivets and removing these baffles. Don't forget to enrich the carburetor jetting.

The "dresser" models have a very short rear cylinder header pipe that results in extremely unequal pipe lengths between the front and rear cylinders. To keep pressure equalized, it helps to retain the crossover pipe connecting the two header pipes.

When installing slip-on mufflers to stock header pipes, consider retaining the crossover pipe for initial tuning. After arriving at a baseline tune, remove the crossover and plug the holes in the headers. This will allow any tuning differences to be determined.

When installing a performance exhaust system, carburetor jetting needs to be checked and generally enriched. For the stock Keihin CV carburetor (1990 and later), the slow speed jet (49-state version) usually must be enlarged (enriched) between .0015 and .002-inch.

STRAIGHT PIPES

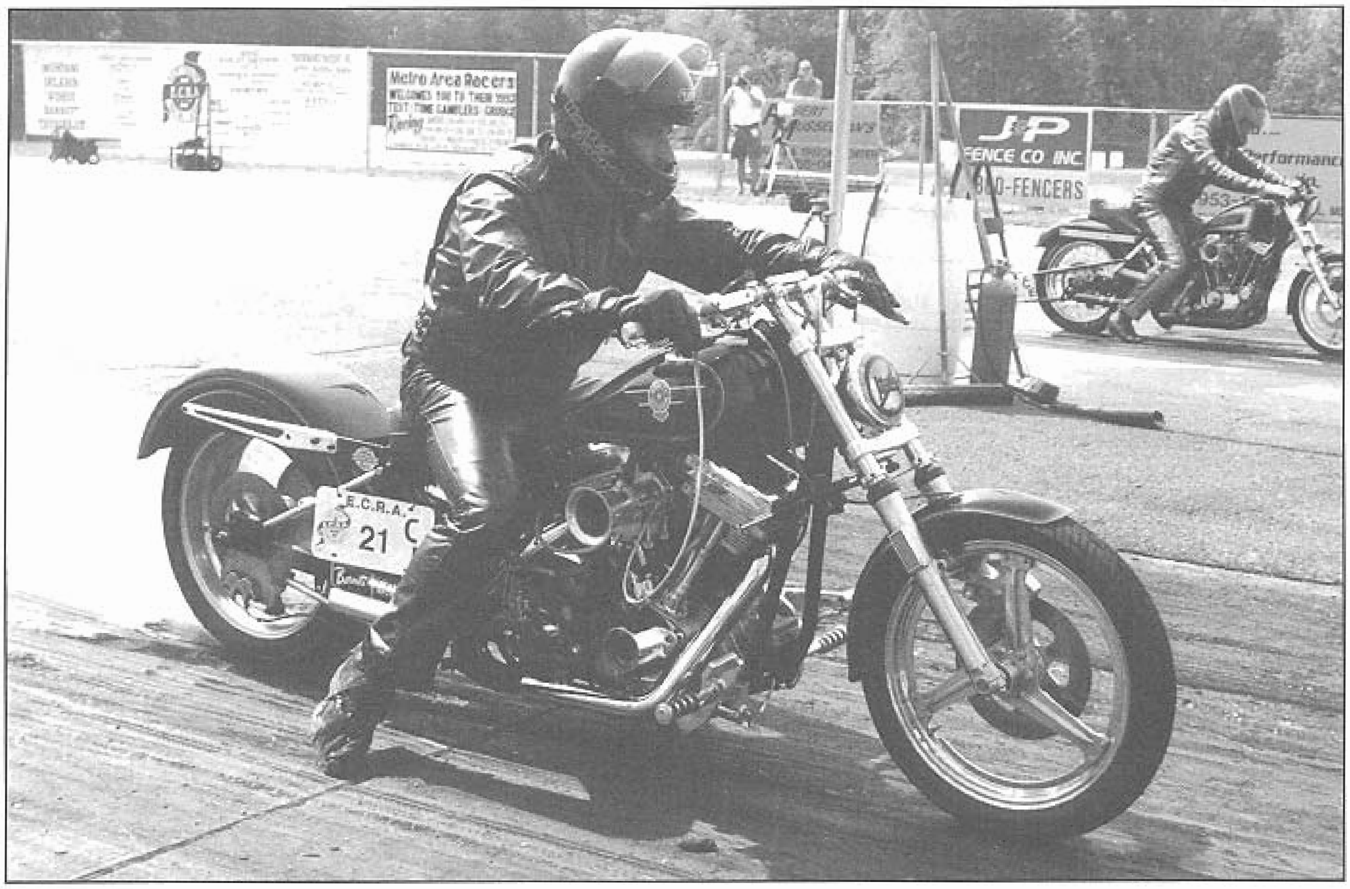
For racing, straight pipes (headers) offer maximum performance because they provide the optimum scavenging with the least backpressure. Performance gains are normally realized on the top-end — from about 4000 rpm and up. Due to high scavenging, straight pipes increase the engine's sensitivity to tuning. In general, the larger the engine's displacement, the easier it is to tune for good low speed throttle response with straight pipes.

The optimum header diameter and length are dependent upon engine displacement, cam specifications, valve and port size, rpm and bike weight. However, subtle factors can enter the picture and influence pipe specifications. For example, an engine with a high velocity carburetor will usually perform better with a higher scavenging header design than a similar engine with a lower velocity carburetor. Also, an engine with a slide type carb usually can handle a higher scavenging header than an engine with a butterfly style carb. The carb's slide seems to minimize the negative effects of high scavenging.

For 98 cubic inch and smaller engines, start with 1-3/4 inch O.D. pipes. For larger displacement engines, 1-7/8 or 2-inch O.D. is a good starting place. Engines with 117 or more cubic inches can benefit from 2-1/8 or 2-1/4 inch O.D. Remember, large diameters increase power at high rpm, but hurt power down low because gas velocity is too slow for much inertia scavenging to take place. Large diameter pipes usually work best when combined with large exhaust valves or engines consistently run at high rpm.

Optimum pipe length typically varies between 28 and 50-inches, although 36 to 42-inches often work well. Remember, short pipes work efficiently at high rpm, while long pipes help performance at lower rpm. It is best to cut the pipes off square; however, if you run slash-cut ends, the measurement is always taken at the short-side of the cut (where exhaust gases first exit into the atmosphere). Also, there should be no more than a 1/4-inch difference in each pipe's length because equal length headers scavenge equally. The measurement should be taken through the center of the pipe, but this can be impractical to do. As long as the diameter remains constant, filling each pipe with a given amount of water sometimes can help measure length.

Straight pipes with stepped sections and anti-reversion tubes or chambers are the optimum pipe for performance, but only are available on a custom made basis. Start with three sections (25,



Straight pipes improve power above 4000 rpm, but hurt it down low. They come in many diameters, lengths and shapes, and most racers have their own idea about what works best. Remember, the exhaust system is highly dependent on the cam, heads, induction system and rpm. What works best on one engine combination may not work well on another. The engine combination and bike's application are the key factors. The only sure way to know what exhaust works best is through cut-and-try testing. Start tuning your exhaust system only after the cam, heads and induction system are set. And make sure the engine is capable of revving within the intended rpm band, otherwise change the gearing. Joe Campoli and his 114 ci Evolution compete in ECRA Competition Eliminator class. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

32 and 43 percent of the total pipe length) and place the smallest diameter section closest to the cylinder head. Avoid "kinked" bends and keep all bends as gentle as possible, especially close to the cylinder head. A pipe diameter slightly larger than the port will help reduce reversion. If you're making your own pipes, use a piece of 1/4-inch brake line tubing with urethane molded over it as a template. The urethane should be the same O.D. as the pipe. To stop mounting nuts from vibrating loose on the Shovelhead, use Stage "8" locking fasteners to fasten the pipe to the cylinder head.

If your bike has aerodynamic panels, try to exit the pipes in a low pressure zone created by the panels. Since atmospheric pressure is about 15 psi, dumping the exhaust into a low pressure area improves scavenging. Adding baffling around the low pressure area to exaggerate the pressure drop also may help. The improved scavenging may allow reduced cam timing and overlap for optimum performance.

EXHAUST TUNING

The exhaust system is only one part of the engine's total breathing system. The other elements include: the induction system, camshaft, cylinder head ports and valves, besides the engine's total displacement. The cam is the central focal point or "brain" of the engine's breathing capability. For the most part, the camshaft determines at what rpm the engine will produce peak torque and horsepower. Radical cams tend to push the point of peak torque and horsepower higher up the rpm scale, while milder cams keep it lower. Header diameter and length should complement the camshaft and induction system.

There is no "caned" method for determining the best header length and diameter for a given engine combination because every component of the engine's breathing system leans on another. Change just one component, such as the cam, and the whole system has to be retuned to guarantee optimum performance. It takes a lot of cut-and-try track testing or a considerable amount of dyno testing to prove which exhaust combination gives the best performance.

Seat-of-the-pants testing does not guarantee accurate results because it takes almost a 10 percent difference in power for the rider to feel a difference. Engine torque curves can change and give the illusion of a power gain. An example is when the engine comes on the cam, only to fall off on the top-end. In this situation, the rider tends to believe erroneously the bike is faster because he can easily feel the increase in torque when the engine comes on the cam. But overall the engine is down in horsepower. As an aside, take note that some engines produce two separate torque peaks because the intake tract is tuned for one rpm level while the exhaust system is tuned for a different level.

Formulas are available to calculate a baseline header length and diameter. However, many factors such as pulsing, velocity, rpm, cam specifications, valve size, gearing and the bike's weight are involved so it is difficult for any formula to give the perfect answer. Some formulas give contradictory answers because each one is designed to take advantage of a different energy pulse.

In the end, if you don't have access to a dyno and most of us don't, the best approach to header design is to start with a thorough understanding of the principles discussed above and then combine this knowledge with cut-and-try testing. If you don't have a good understanding of the principles involved, you can never consistently beat your competition.

Even if you're not racing, knowing what affect changing pipe diameter or length has on the engine's power band may save you from buying the wrong exhaust system. For example, how many times have you read a magazine editorial about a guy who installed a set of large diameter straight pipes and then asks, "Why can't the carburetion be tuned for good throttle response?" The answer is relatively simple once the principles of velocity, scavenging and reversion are understood.

To determine a starting point for header design, you should establish the engine's displacement, cylinder head airflow, valve sizes, gearing, total bike weight and rpm range. From this information a cam can be selected.

At this point, a starting pipe diameter and length can be roughly calculated. The three-step header design with an anti-reversion tube has proven to be a strong performer, although many racers go fast using a standard straight pipe design. If you're not sure where to start, follow the previous recommendations.

Large diameter pipes usually perform better with oversize exhaust valves. When a large pipe diameter is used with a small engine displacement, the result is generally a narrow torque band placed at a high rpm. And this is acceptable for certain maximum effort engines. However, for street engines, keep pipe diameter small to boost low-speed torque. Use the mean gas velocity formula to check that you're in the 300 feet per second range when at the *middle* of the rpm band—not the top-end of the band. Calculate for the middle because you want to achieve the highest average horsepower over the entire rpm band.

For street engines the rpm band is easy to determine — you're probably doing most of your riding between 2000 and 4000 rpm, so 3000 is the midpoint. In some situations, the midpoint could be as high as 3500 rpm. To verify your rpm band for drag or road racing, you need to consider not only the engine's bore, stroke, airflow and cam, but also the gear ratios. For example, lets assume you up shift at 6500 rpm with your engine combination. After each up shift, the engine's rpm drops down to a level that is a function of the percentage of gear change. You need to calculate the rpm reduction for each gear change and then average them out. You can assign a weighted priority to a specific gear if you want.

Let's assume the transmission's internal first gear ratio is 3.25 and the ratio for second gear is 2.20. Use the following formula to calculate the percentage of rpm change when up shifting between the two gears:

$$percent = \frac{(from\ gear\ ratio - to\ gear\ ratio)}{from\ gear\ ratio}$$

Example:

$$\frac{(3.25 - 2.20)}{3.25}$$

$$\frac{1.05}{3.25}$$

.32 rpm reduction

The gear ratio used in the calculation can be either the internal transmission ratio or the overall ratio, but both cannot be mixed within the formula. This example results in a 32 percent rpm reduction after the up shift. The smaller the rpm drop and the closer the drop is to the maximum torque rpm, the better the bike's acceleration should be. The following formula can now be used to calculate the rpm after the up shift:

$$after\ rpm = shift\ rpm \times (1.0 - percent\ of\ gear\ change)$$

Example:

$6500 \times (1.0 - .32)$

6500×0.68

4420 rpm after up shift

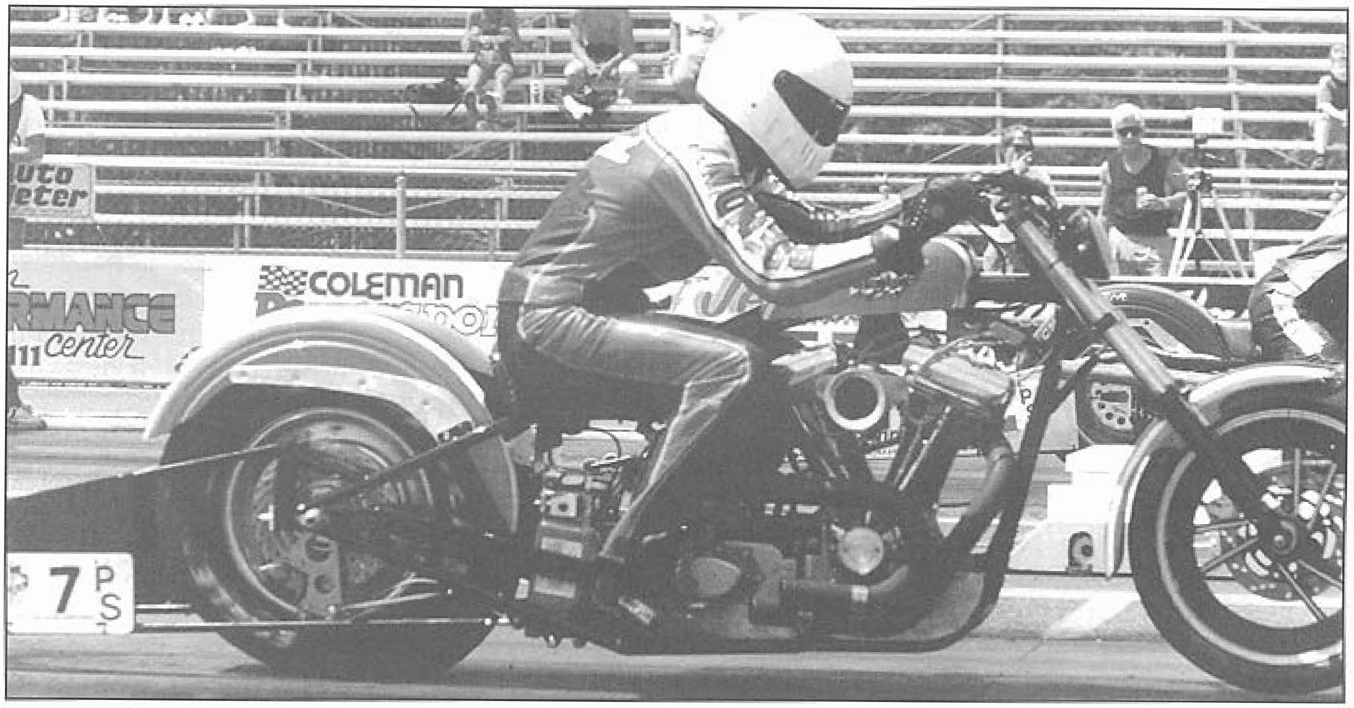
This works out to 4420 rpm after the up shift. The midpoint between 6500 and 4420 rpm is 5460. Now base all your exhaust calculations on 5460 rpm. Remember that the percentages of change between each gear are not always the same, so calculate each gear separately and average them out.

THE OPTIMUM SHIFT RPM

The optimum shift point for maximum acceleration is generally a rpm 10 to 15 percent greater than the engine's maximum horsepower rpm. The transmission's internal ratios and the engine's torque curve will affect the percentage. With a stock 5-speed transmission, a rpm 10 to 13 percent above maximum horsepower rpm is about optimum.

For example, a mildly hopped up Evolution Big Twin (80-inch engine) with a performance cam, carb, exhaust and ignition system will realize maximum power at about 5750 rpm. This results in an optimum shift point of about 6300 to 6500 rpm. Add performance heads to this combination and maximum horsepower is achieved closer to 6200 rpm. Now the optimum shift point moves up to between 6800 and 7000 rpm. Furthermore, with a high dollar, maximum effort 80-inch engine, maximum horsepower is between 6800 and 7000 rpm and the optimum shift point is between 7500 and 7800 rpm. Although an engine's optimum shift rpm is about 10 percent higher than its maximum horsepower rpm, the bike should pass through the speed traps at a rpm about equal to maximum horsepower.

With a Shovelhead engine, the above values are about 500 rpm lower. Remember, the stock Evolution valvetrain can be safely revved to about 6300 rpm. Any greater rpm requires valvetrain modifications.



Maximum effort dragsters run high rpm and use close ratio gears for minimal rpm drop between shifts. This allows tuning the headers for maximum power over a narrow high rpm band with little concern for power down low. Large diameter headers provide excellent high rpm scavenging for large displacement engines. Pierre Frenette on his Evolution Pro Stock dragster. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

CALCULATING HEADER LENGTH

Any formula that calculates header length must consider, at the very minimum, exhaust gas temperature, speed of the energy wave, exhaust duration and rpm. However, every formula includes certain assumptions. Some formulas calculate a long pipe length because they are based on a different energy wave than other formulas. Remember, energy waves resonate within the pipe. Retarded ignition timing and other conditions increase the exhaust gas temperature, which causes the energy wave to travel faster. This requires using a different constant in the formula for the energy wave's speed. As a result, the following formulas make certain assumptions that may not be valid for every engine combination.

Exhaust Header Length
Formula A:

$$L = \frac{1380 \times ex}{rpm}$$

Formula A results in a very long header length that some racers do not like to use because they believe it doesn't look aesthetically pleasing. However, it works extremely well with large diameter stepped headers.

Exhaust Header Length
Formula B:

$$L = \frac{950 \times ex}{rpm}$$

Formula B results in a shorter header length because it is factored against a different energy wave pulse.

Where:
L = header length in inches.
ex = 180 plus cam's exhaust valve opening BBDC
rpm = rpm in the middle of the power band

Use formula B and assume that 5460 is the engine's rpm midpoint and the cam's exhaust timing is 56/24. The variable "ex" is determined by adding 180 to 56 for a sum of 236. This formula calculates a header length of 41.06 inches.

$$L = \frac{950 \times 236}{5460}$$

$$L = \frac{224200}{5460}$$

41.06 inch header length

Header length should be measured through the middle of the pipe starting from the backside of the exhaust valve to the open end of the pipe.

Remember that shorter pipes require a higher rpm to reach the point of optimum scavenging, while longer pipes are optimized for the lower rpm ranges. If you're not sure where to begin, start with a pipe length between 36 and 42 inches. Refer to the guidelines listed under "Straight Pipes" for more information.

MAKING A RUN

If we assume that you're at the track and ready to tune the exhaust system, there are a number of points to consider. First your bike must have a chassis and drivetrain setup that allows you to make consistent runs. A bike that cannot launch properly or run at full power through the entire track's length without encountering large wheelies or other handling problems will give inaccurate results. If you're having handling problems with the bike, get them corrected before conducting any tuning tests.

Exhaust system tests should be conducted after the cam, carb, intake manifold and head combination is determined. Also, the exhaust system can be accurately tuned only after the carburetion, ignition and gearing are tuned. The carburetion should be done first because the optimum ignition advance will vary based on the air/fuel mixture. Remember that open headers require a richer air/fuel mixture than mufflers.

The bike doesn't have to be perfectly geared to conduct exhaust tuning, but it should be close enough so the engine accelerates to redline in each of the lower gears and close to redline in high gear. For example, if the engine is up shifted at 6800 rpm, exhaust tests would be invalid if the engine only turned 4000 rpm through the track's mph timing lights. In this case, the gearing is too high (low numerically) for the engine and bike combination, and should be lowered to get the rpm up to at least 6000 through the timers. Remember that factors such as the total weight of the bike and rider, cam specifications and wind direction will have an effect on the optimum gearing.

If you're running a solid lifter cam, you might try increasing the valve lash setting (from one to three flats on the adjuster nut) to see if the engine performs better with less cam duration. Increasing the pushrod clearance reduces duration.

With this said, one of the easiest methods for tuning the exhaust for a given diameter is to start with long pipes and then start shortening their length in two inch increments. Strive for the lowest elapsed time on the drag strip. Once you have determined the optimum length, you might consider increasing the diameter and again conducting tests for optimum length. Now you will have a comparison between different pipe diameters.

Carburetion, timing and gearing may need to be refined during the exhaust tests. If the camshaft or anything else is changed, all prior testing must be redone to verify the combination. Remember, each element in the tuning chain leans on something else.

Once the best combination of header diameter and length is identified, you can try reducing the cam's overlap. With effective exhaust scavenging, the engine may not need as much overlap. Also, you might try opening the exhaust valve a little later since the engine may not need as much blowdown effect for scavenging. This will dump less power out the exhaust. ❖

Chapter 8

Ignition System and Spark Timing

Let The Sparks Fly

Most racers concentrate their performance efforts on improving airflow through an engine.

However, burning the highest percentage of the air/fuel mixture is just as important for increased horsepower as high airflow. Although the ignition system is one of the least visible and glamorous parts of an engine combination, it is responsible for the initiation of efficient combustion.

The ignition system performs a relatively simple task: it supplies a spark at a given moment that initiates combustion in the cylinder. Its ultimate goal is complete combustion of the air fuel/mixture in the cylinder. Realistically this is impossible, so the goal then is to burn as much of the intake charge as possible. The key to an effective ignition system is the quality of the spark and the time at which the spark is generated. A quality spark in conjunction with steady timing will ensure that the greatest amount of air/fuel mixture will be ignited over the widest range of conditions so the engine reaches its maximum power potential.

The spark that ignites the air/fuel mixture in the combustion chamber is produced by a discharge of high voltage electrical energy developed by the ignition coil's circuits and delivered to the spark plugs through the plug wires. The spark is generated because the high voltage energy is traveling through the ignition's wiring searching for a ground. The ground happens to be the plug's side electrode, which is positioned just beyond an air gap that separates it from the plug's center electrode. When the electrical energy bridges the gap, a spark occurs, which hopefully ignites the compressed air/fuel mixture.

With a stock ignition, the duration of the spark is short and not very powerful. If the air/fuel ratio and other conditions in the combustion chamber are just right, the mixture will ignite. However, in some cases the air and fuel may not be evenly mixed or the mixture may be lean. Either of these conditions can allow chamber turbulence or high compression to extinguish the spark and cause misfires.

All stock Big Twins use an inductive-discharge ignition (ID) system that combines a coil, a switching device and some type of advancing mechanism to fire the spark plugs.

The 1980 and later ignition is an electronic system that uses a microprocessor based module (black box) to switch the ignition coil on and off electronically and control the ignition's timing for greater reliability and longevity than breaker points. It has an ignition advance curve designed for emissions regulated engines and a built in rev limiter that limits the engine to about 5250 rpm to minimize wear. Police bikes are limited to 5800 rpm. The stock system essentially is a reliably weak system. A modified, free-breathing Big Twin engine not only can benefit from a quicker advance curve, but also from a higher rev limit and a fatter spark to increase the amount of mixture burned.

Engines before 1980 are equipped with a mechanical advance mechanism and either an electronic sensor (1978 and 1979) or breaker points. These setups have a better advance curve than the electronic ignition module and no built-in rev limiter, but they can be improved for better performance.

Additionally, all stock Big Twin ignitions, regardless of the year or model, fire the plugs in both cylinders (dual-fire) at the same time. This results in a cost effective ignition, but causes engine roughness at lower rpm. Some performance ignitions are designed to fire only one plug at a time (single-fire mode) for better power, easier starting and smoother operation.

To receive the maximum power from a performance carburetor, cam or exhaust system, the ignition system must maximize combustion of the intake charge while keeping spark timing accurately advanced. It also must allow the engine to rev beyond 5250 rpm. Understanding the fundamentals of ignition system operation will help you evaluate different systems and optimize your system for maximum performance.

IGNITION TYPES

There are several basic types of ignition systems. Most use a battery and coil to operate, although some generate their own current. The major difference between the systems is how they cause current to buildup in the coil to generate a spark. The current buildup determines the quality of the spark and its duration. Ideally, the stronger the spark and the longer its duration the better performance should be.

All stock Big Twin ignition systems and most aftermarket ignitions use a battery and coil design. They also use either a mechanical or electronic device to turn the coil on and off and control the spark advance curve. The exception to this is a magneto since it does not require a battery for a power supply because it generates its own power.

Battery and coil ignitions for the Big Twin generally use either the inductive-discharge (ID) or the capacitive-discharge (CD) method for generating a spark.

INDUCTIVE-DISCHARGE (ID)

All stock Big Twin ignition systems use the inductive-discharge method for generating a spark. This system requires a battery and coil to operate. It also requires a method for turning the coil on and off and a technique for regulating the amount of ignition advance. Engines made before 1980 use breaker points for switching the coil on and off and a centrifugal mechanism for ignition advance. The 1978 and 1979 models use an electronic sensor in place of breaker points. Starting in 1980, all models use an electronic sensor for switching the coil and an electronic module (black box) for advancing the ignition. Also, most aftermarket performance ignition systems use the ID method for generating an ignition spark.

With the ID ignition, the coil acts as a transformer to increase or decrease voltage levels.

Every coil actually consists of two separate coils or windings of wire. One winding is the primary and it often has 100 to 150 turns of heavy copper wire wound around an iron core. The other winding is the secondary and it is normally wound around the primary. The secondary winding consists of about 20,000 turns of fine copper wire and the ratio of turns between the primary and secondary windings partially helps generate the high voltage energy needed to bridge the spark plug gap. The ratio of the secondary winding's 20,000 turns to the primary winding's 100 turns is 200 to one, so you only get about 2400 volts (200 x 12) from a 12 volt coil. However, since most high-performance coils generate between 30,000 and 40,000 volts (stock coils are about 24,000 volts) another element, a magnetic field, enters into the act.

As voltage from the battery enters the primary winding, it generates a magnetic field that passes through both coil windings and the core. Electrical current can be generated by changing the magnetic field in the windings. It takes about 10 to 15 milliseconds (thousandths of a second) for the magnetic field to build up to maximum strength at which time the coil is considered saturated. Switching off the flow of current to the primary winding, which takes less than one thousandth of a second, causes the magnetic field to collapse. Early model Big Twins (1977 and earlier) use breaker points to switch the coil while late models use an electronic sensor. The sudden collapse of the magnetic field increases the primary winding's voltage from 12 volts to between 150 and 200 volts. The actual voltage is proportional to the speed at which the magnetic field builds and collapses. Now the secondary winding, with its 200 to one turn ratio, increases the coil's voltage to about 40,000 volts (200 x 200).

COIL SATURATION

The coil's secondary voltage is determined not only by how fast its magnetic field collapses, but also by how strong the field is at the time it is collapsed. One way to increase the strength of the magnetic field as it reaches full saturation is through improved coil design. High powered aftermarket coils have a higher secondary to primary turn ratio, which increases the secondary output voltage. A performance coil is required for highly modified engines and is recommended even for a relatively stock engine if the engine is to perform up to its maximum potential.

CAPACITIVE-DISCHARGE (CD)

As previously discussed, inductive-discharge ignition systems are relatively simple in that they only require a coil and either a mechanical or electronic activated switch for the coil. However, they also have an inherent disadvantage because time is required to build a strong magnetic field and at high rpm the time between cylinder firings is reduced.

A capacitive-discharge system works opposite of an ID system. Instead of building and collapsing a magnetic field in the coil, it uses capacitors inside a control module that are charged with high voltage. A breaker point or electronic switch is then used to trigger the capacitors in the control module to release their high voltage to the coil's primary winding. The high voltage in the primary winding creates a powerful magnetic field that generates the secondary winding's high voltage. The intent of a CD system is to generate a higher secondary current for a hotter spark. Instead of being limited by how fast the coil reaches saturation, the CD system is limited by the speed at which the capacitors can be recharged.

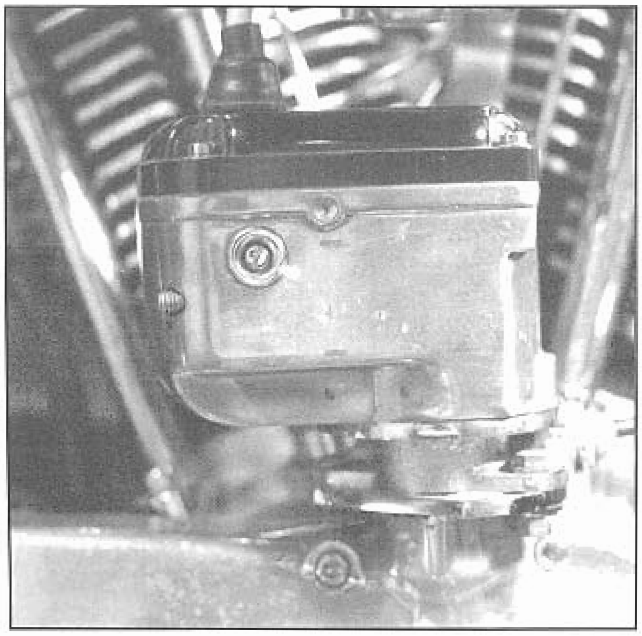
The benefit of a CD system is its ability to generate high secondary energy during a short recharge time interval that allows it to fire wide-gapped plugs at high rpm. Its short recharge time provides less time for energy to leak off. Although a CD system works well in a high rpm environment, its spark is relatively short in duration. In many instances the spark is many times shorter than that of an inductive-discharge system. This results in a tradeoff between a high energy spark with short duration (CD system) or a longer duration spark with lower energy due to little time for coil saturation at high rpm (ID system). An ID system distributes its energy over a long period with relatively little energy at the spark's beginning. The CD system, however, expends most of its energy up front, but has little remaining to maintain a long duration spark.

MAGNETOS

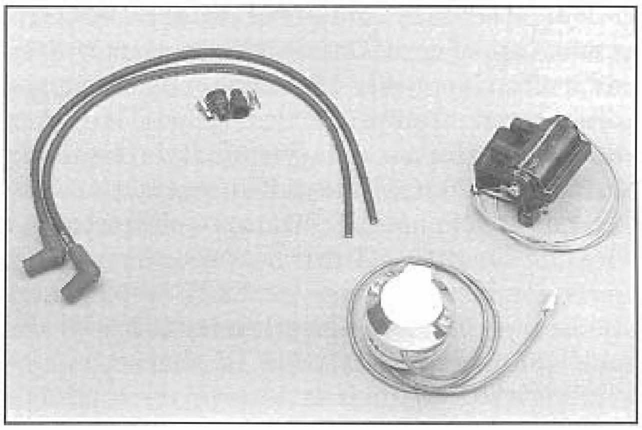
Although a magneto was never included as standard equipment, many Big Twins have been updated with an aftermarket mag because the engine builder wanted better performance or a custom look. Most magnetos for the Big Twin are similar in design to the Fairbanks-Morse mag that was used on early model Sportsters.

The magneto has been around since the beginning of motorcycles and has proved itself because it is small, light weight, has few moving parts, doesn't need a battery or power supply and provides an excellent spark at high rpm. The disadvantages to using a mag are that at low rpm the mag has a difficult time generating enough power to fire the spark plug. It also can require more engine horsepower to operate than other ignition systems and it periodically needs to be rebuilt to operate at peak performance. Most, but not all magnetos work in a manner similar to an inductive-discharge ignition system with the exception that the mag includes its own power generating source to energize the primary coil.

A magneto consists of a rotating shaft with attached magnets besides a stator, coil, breaker points and condenser. A magneto generates current by rotating the magnets inside the stator.



The Fairbanks-Morse magneto was used on early model Sportsters and is often custom fitted to Big Twins. Since it includes a self-contained energy source, it is not dependent on an electrical source and heavy battery. It has a custom look, but lacks the performance features of electronic ignition systems.



This simple CDI magneto from Carl's Speed Shop produces a high energy spark and requires little engine power to turn it. It uses a maintenance free Hall Effect magnetic trigger sensor instead of points and does not require a battery or charging system. Photo courtesy of Carl's Speed Shop.

The stator includes a laminated steel core that is wound with copper wire. As the magnets pass by the stator's copper wires, a current is developed in the stator or primary coil. When the points are switched off, the primary coil's magnetic field collapses and its voltage is passed to a secondary coil. The secondary coil increases the voltage and then passes the current to the spark plug, jumping the gap and generating a spark.

An interesting point is that the current generated by the magneto's magnets and stator is the same as an alternating current (AC) generator. Also, as the speed at which the magnet rotates within the stator increases, the rate at which current is built-up in the coil increases proportionally — doubling the speed doubles the rate at which current is built-up. This phenomenon allows a magneto to generate a constant spark voltage as engine rpm increases even though the time between sparks is reduced. This is directly opposite of an induction-discharge ignition that inherently reduces spark voltage with increased rpm. Also, the mag's high voltage spark has a long duration, which means the spark is spread over more degrees of flywheel rotation. With these inherent characteristics, a magneto does an excellent job of igniting the intake charge at high rpm.

One negative factor of a magneto is that it requires more engine power to operate. This is because no power is ever self-generating — it can only be converted. Therefore, a mag requires some engine horsepower to spin the shaft and magnets that generate current. The amount of horsepower it requires depends on engine rpm and other factors, but it can range between one and three horsepower for a Big Twin engine. Since a magneto struggles to generate current at low rpm (500 rpm or lower), kick start bikes may be difficult to start and the possibility of flooding the engine or fouling the spark plugs is increased during starting. A heavy duty condenser that doubles the stock condenser's 2.5 microfarads capacity usually helps hard starting.

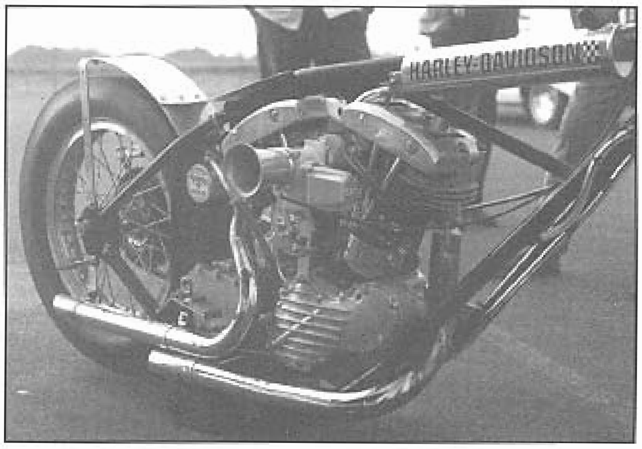
The magneto's breaker points, condenser and bearings require some maintenance and the magnets should be remagnetized periodically (yearly for a race engine) for maximum spark energy. For Fairbanks-Morse magnetos, set the breaker point gap between .014 and .015-inch and the spark plug gap between .019 and .020-inch.

Carl's Speed Shop sells a state-of-the-art CD magneto that produces a high energy spark, uses little engine power and is small in size. Instead of using out dated points and condenser, it uses a maintenance free Hall Effect magnetic trigger sensor and requires no battery or charging system.

BREAKER POINTS

Early model Big Twin ignitions and most magnetos use breaker points to switch the coil on and off. Even some high-performance ignitions use breaker points as a switching device.

Although archaic, breaker points, if setup properly, can offer a low cost, efficient alternative to cutting-edge electronic ignitions. Their main disadvantage is that they require periodic maintenance to keep adjusted and



Joe Smith was the first Harley drag racer into the 8 second bracket and during the 1960s, and 70s he won Top Fuel titles with both single and double engine bikes. Joe's extremely clean looking single engine machine was well-known as meticulous prepared and reliably fast. Notice the magneto ignition, Panhead crankcases mated to Shovelheads, S&S fuel carb and rocker cover cut-out for easy valve clearance checking. On this June day in 1970, Joe ran a 9.07 E.T. at 164 mph on his 102 ci Top Fueler.

working efficiently.

As breaker points open up and turn off current to the coil's primary winding, an electrical arc develops across the contacts. The arc allows current to continue to flow to the primary winding, which slows the collapse of the magnetic field and lowers secondary voltage. By connecting a condenser (capacitor) across the points, arcing is almost eliminated and secondary voltage is increased because the magnetic field is collapsed much quicker.

Ideally, points should have a rapid opening, slow closing and no bounce at full open or close. This requires the use of a quality ignition cam lobe and points with sufficient spring pressure. Blue Streak points (for the Chevy in-line six) fit without modifications and not only have stron-

IGNITION COMPARISON GUIDE

	SPARK DURATION (Microseconds)	SPARK ENERGY (Milliouls)	SPARK CURRENT (Milliamp)	RISE TIME (Microseconds)	SPARK GAP SIZE (Thousands Inch)	POWER TO OPERATE (Horsepower)
INDUCTIVE - DISCHARGE (Points)	MEDIUM	MEDIUM	LOW	MEDIUM	MEDIUM	LOW
INDUCTIVE-DISCHARGE (Electronic)	MEDIUM to LONG	MEDIUM to HIGH	MEDIUM	MEDIUM	MEDIUM to LARGE	LOW
CAPACITIVE - DISCHARGE	SHORT	LOW	HIGH	SHORT	LARGE	LOW
MAGNETO	LONG	HIGH	MEDIUM	LONG	SMALL	HIGH

Table 8.1

ger spring tension than stock units, but also include several other benefits.

The benefits start with vented contacts that have a small hole drilled through their centers. The hole reduces operating temperature and allows gases formed by electrical arcing, such as ozone to escape. A felt lubrication cam wiper is also included to help minimize rubbing block wear, which is usually the area of highest wear. The Blue Streaks are also copper plated for better electrical ground contact, increased heat conduction and improved cooling. The thin bladed tension spring, which doubles as a ground connection, is also permanently fastened to the points' body for a positive ground. The spring also has higher tension than a stock unit. For the race track, a second tension spring can be placed next to (on the inside) the main spring for increased tension. However, remember that increased spring pressure wears the point's rubbing block quicker.

Points generally have a thin protective film on their contacts, which should be removed with fine emery cloth. A heavy duty points backing plate can be installed to help eliminate plate distortion for more accurate timing.

CONDENSERS

A condenser is required for ignitions that use breaker points. The condenser reduces excessive point arcing and speeds the collapse of the coil's magnetic field, but it must be properly matched to the breaker point set and ignition system.

Condensers have various amounts of capacity and the capacity affects the pitting of the points' contact surfaces. The capacity is rated in microfarads. If material collects on the points' negative contact, the condenser has too high a capacity. On the other hand, if material collects on the positive contact, its capacity is too low.

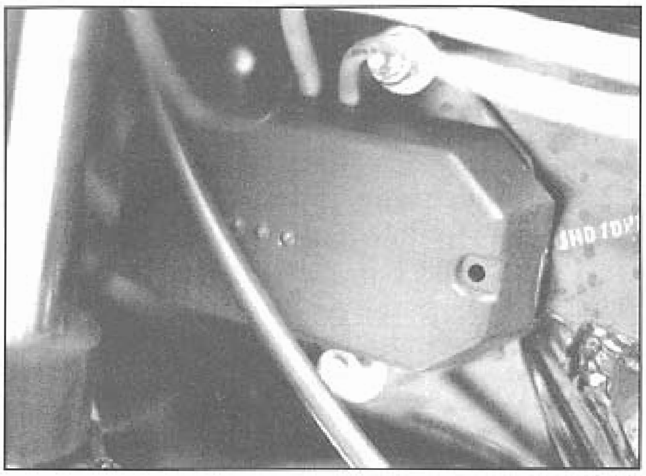
Use the highest capacity condenser possible to minimize contact arcing so the coil's magnetic field collapses quickly. Also, don't forget to lightly lube the felt cam wiper with point cam lubrication.

ELECTRONIC IGNITIONS

As discussed earlier, an electronic ignition offers an alternative method to breaker points for switching an ignition coil on and off. Modern electronic ignitions use a combination of microprocessors and sensors to perform various ignition functions. The microprocessors provide for programmable advance curves and other cutting-edge features. The sensors, which are either magnetically or optically controlled, eliminate the need for breaker points. When compared to breaker point ignitions, the combination of microprocessors and sensors offer the potential for steadier ignition timing, besides reduced maintenance requirements.

Some electronic ignitions use sensors to switch the coil on and off while others do much more. Exact capabilities vary from system to system, but can include features such as a programmable advance curve, adjustable rev-limiter, single-fire operation, multi-spark capability, corrected tachometer output and retention of the stock vacuum operated electric switch (V.O.E.S.). The V.O.E.S. retards ignition advance under high load conditions.

Accel, Crane, Dyna, Harley-Davidson Screamin' Eagle, MC Power Arc and others make electronic ignitions for the Big Twin. Some of these ignitions include all the previously mentioned features, while others include a few. For example, the stock factory Evolution ignition includes a magnetic sensor for switching the coil on and off, non-adjustable electronic advance curve, non-adjustable rev limiter and (V.O.E.S.) capability.



The M.C. Power Arc single-fire electronic ignition has an adjustable advance curve and uses a maintenance free optical sensor as a trigger. Adjustable screws and pins allow you to set the advance curve and built-in rev limiter to your engine's needs. It also can fire a second spark to the same cylinder and has V.O.E.S. capability for street engines. Use with a high-performance coil for maximum performance. Shown is the Power Arc control module mounted on an FLHT frame. Notice the LED lights used for setting the ignition advance curve.

Although electronic ignitions contain state-of-the-art electronics to perform various functions, keep in mind that they must be combined with high voltage coils to produce a high energy spark for maximum power potential.
Having a basic understanding of the features and capabilities of different ignitions will help you choose the best system for your engine combination.

ELECTRONIC SWITCHING

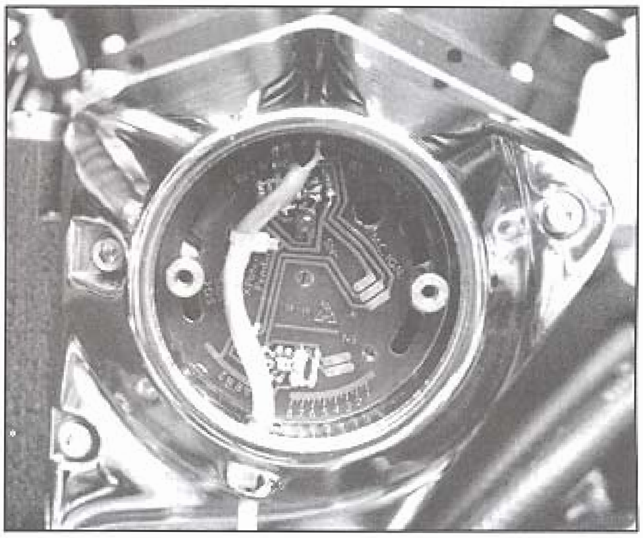
The electronic sensor for turning the coil on and off includes a trigger rotor and sensor assembly, that are located in the engine's cam gearcase cover on the engine's right side. The Big Twin's stock sensor operates magnetically using a Hall Effect magnet; however some performance ignitions use an optical sensor. An electronic sensor maintains adjustment much longer than breaker points and can provide more accurate timing.

ELECTRONIC ADVANCE CURVE

Every electronic ignition requires a mechanism for ignition advance so the intake charge is ignited before TDC. This ensures cylinder pressure is maximized at the optimum time. Some electronic ignitions, such as the Power Arc, create the advance curve within the electronic module (black box) while others use a conventional mechanical advance unit. Also, some electronic modules are programmable in the sense that either DIP switches (Accel) or adjustable screws (M.C. Power Arc) are used to provide a choice of different advance curves that can be matched to a particular engine combination and riding style.

SINGLE-FIRE

All stock Harley ignitions, both Evolution and pre-Evolution, fire the spark plug in both cylinders at the same time. This is nothing more than a low cost method for building an ignition, since it does nothing to enhance performance. In fact, it can actually hurt it. With this setup, the rear cylinder receives a spark when its intake valve is open at the beginning of its intake stroke. This robs some of the coil's energy from the plug in the cylinder that is about to fire, which can reduce the spark's intensity. Additionally, the incoming charge in the rear cylinder can be ignited at the wrong time, causing a backfire through the carburetor and sometimes harder starting. These characteristics are more noticeable when using a long duration cam with an early opening intake valve.



The M.C. Power Arc optical trigger replaces the stock trigger mechanism. The sensor plate includes adjustment slots and a red LED (top center) to simplify static timing.

The Dyna "S" single-fire electronic ignition uses a Hall Effect magnetic trigger and a mechanical advance unit for the advance curve. It should be combined with a high-performance coil for maximum performance. Shown is the Dyna single-fire magnetic trigger module.

On the other hand, a single-fire ignition fires the front and rear cylinders independently of each other, so there is no wasted spark to drain some of the coil's energy. This allows for easier starting, smoother running and less vibration at most rpm levels. Additionally, crisper throttle response and higher performance may be experienced.

Another important point is that single-fire ignitions require two coils, instead of one. This requires finding an acceptable location for mounting the second coil, besides obtaining a bracket that holds both coils. CCI and Rivera Engineering offer lightweight billet aluminum brackets that hold two coils and fit most Big Twin models. On FLT models, a second coil can be mounted under the gas tank where the original coil is mounted. However, be aware that this is a tight fit and requires fabrication of a special U-shaped mounting bracket. Another mounting location is under the seat.

For single spark plug heads, each coil can have a single plug wire tower. When dual plug heads are used, coils with dual wire towers are required. However, dual tower coils can be used with single plug heads as long as their second tower is grounded.

A single-fire ignition only sends one half the signals to the tachometer as a dual-fire system does. As a result, the ignition's signals must be corrected for an accurate tachometer reading. Some electronic ignitions, such as the MC Power Arc, include a built in tachometer correction, while other systems need a separate adapter.

All ignition systems require a compatible coil for proper operation, and single-fire systems are no exception. Therefore, be sure to select coils with the proper ohms and voltage rating for your ignition.

MULTI-SPARK

One of the major goals of an ignition system is to provide total combustion of the intake charge. Realistically this is impossible because of inefficient combustion chamber design. Some unburned fuel will be pushed out the exhaust. The only question is how much.

To burn the intake charge more completely, some electronic ignitions generate either a second spark or several sparks to the *same* cylinder during the ignition process. For example, instead of producing a single spark for ignition, the MC Power Arc ignition can be setup to deliver a second spark at five degrees BTDC. (Be careful not to confuse this multi-spark capability with the stock ignition system that inefficiently fires *both* cylinders at the same time.) The second spark attempts to ignite unburned fuel remaining in the combustion chamber as the cylinder's power stroke is about to begin. MSD's motorcycle ignition generates up to six sparks when the engine is at low rpm and the time between each power cycle is the longest. As engine rpm increases, the number of sparks is reduced to approximately two at high rpm.

Horsepower increases from multi-spark ignitions depend on the combustion chamber's efficiency. Efficient, small volume chambers with a short flame travel and high turbulence generally benefit less than non-efficient chambers. On the other hand, chambers that have a long flame travel, low turbulence or shroud the flame front generally benefit more. Other benefits from multi-spark operation include reduced plug fouling, smoother idle and improved gas mileage.

RPM LIMITERS

A rpm limiter is recommended for all engines to prevent damage to the valvetrain, pistons and other components in the event a missed shift or powertrain failure occurs. A built-in programmable rpm limiter is available with some electronic ignitions. Electronic modules that do not include a rpm limiter can have a separate, free-standing rpm limiter added at an additional cost.

The stock Big Twin electronic ignition includes a rev limiter, however, it is non-adjustable and limits the engine to only 5250 rpm. This is a major drawback for using the stock ignition system on a free breathing engine.

KV Products manufactures the freestanding Dyna two stage rev limiter for improved drag strip launching. This limiter allows a controlled rev limit during launch while providing a high rpm limit after launch. Its benefits include easier staging of the bike and more consistent launches. Both rev limit stages are controlled by a switch on the clutch lever. Also, the limiter is designed to operate with all inductive-discharge electronic ignition systems on two cylinder engines.

VACUUM OPERATED ELECTRIC SWITCH (V.O.E.S.)

The stock Evolution ignition module includes a vacuum operated electric switch (V.O.E.S.) that retards the ignition advance when the engine is under high load (low intake manifold vacuum) detonation prone conditions. The V.O.E.S. is primarily helpful on street engines where low octane pump gas, stock cylinder heads and heavy bikes are more the rule than the exception. Some electronic ignitions retain this feature.

Be aware that the V.O.E.S. is a "stopgap" measure that retards the ignition timing to reduce detonation caused by poor quality gas and/or low combustion chamber turbulence. Eliminating the switch and grounding the wire (if your ignition supports a V.O.E.S.) connecting it to the ignition module can help performance, but may allow unacceptable levels of detonation. An engine will have snappier throttle response and stronger acceleration (assuming no detonation is present) when the V.O.E.S. is deactivated. If eliminating the switch allows too much detonation on a heavy FLT, install a V.O.E.S. from a lightweight FXR instead. This switch will still retard, but not as easily as the switch from the FLT.

Another option is to connect a ground switch to your bike's handlebars and to the ignition module's V.O.E.S. wire. This will allow the V.O.E.S. to be turned off selectively. Under high load detonation conditions, flipping the ground switch off (ungrounded) will temporarily activate the vacuum retard and reduce the potential for detonation.

The V.O.E.S. is usually located near the top engine mount and includes two wires besides a vacuum hose. One wire is grounded and the other connects to the ignition module. The V.O.E.S. is deactivated when grounded, which means a simple on off ground switch can be connected to the ignition module wire to deactivate it. The vacuum hose connects the V.O.E.S. to a fitting located either on the carburetor or intake manifold.

The internal contacts of the V.O.E.S. can stick and cause an unexplained drop in performance. If your engine's performance drops and you have checked for all possible causes without locating the problem, try grounding the *ignition module's* V.O.E.S. wire to see if performance returns. If it does, the switch is the problem.

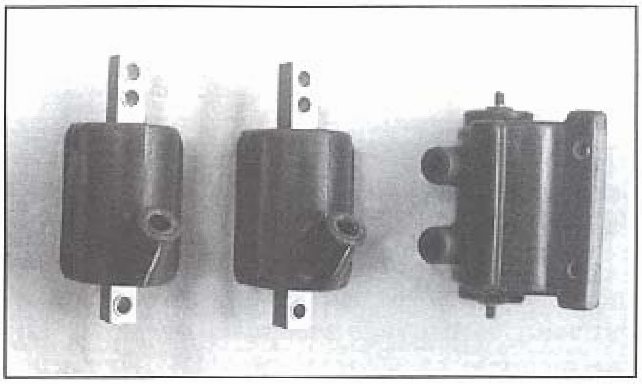
HIGH-VOLTAGE COILS

Much of the development devoted to stock electronic ignition systems is the result of more stringent EPA emissions testing. Although electronic ignitions have improved reliability and eliminated the need for high maintenance breaker points, in general, stock ignitions produce a relatively weak spark. This not only limits the engine's power potential, but it also can cause part throttle missing and surging.

Adding a high-voltage coil and performance spark plug wires to a stock ignition provides a definite improvement in starting and idling, gives crisper throttle response and allows the engine to perform to its potential. It's also the lowest cost method for improving the performance of the stock ignition, although it does not provide all the features and benefits of a high-performance system.

All high-performance ignitions, with the exception of some magnetos, require a high-voltage coil(s) to produce a high energy, long duration spark. So be sure to match high powered coils to performance ignitions.

Coils are rated by several factors. The most noted factor is its output voltage to the spark plug (20,000 to 40,000 volts). Two other important factors are the coil's operating power (6 or 12 volts) and its output resistance (for example,



Dyna single tower performance coils are on the left while the stock Big Twin coil is on the right. Be sure to match the coil's ohms and voltage ratings to the ignition system. Use either single or dual tower coils depending on whether the heads are single or dual plug and whether the ignition is single or dual-fire.

1.5 or 3.0 ohms). The output voltage to the spark plug plays a part in determining the spark's energy and duration while the coil's operating power and output resistance must be correctly matched to the ignition for proper operation.

Most high-performance coils are referenced based on their output or secondary voltage such as 30,000 or 40,000 volts. Voltage is the electrical standard of pressure. Powerful high-performance coils, by their inherent design, store greater energy, which results in a more powerful, longer duration spark. However, the coil's secondary voltage is only one factor to consider when installing a new ignition system. Although it's true that the coil must provide enough output voltage at the spark plug to cause a spark to jump the gap, it really is current (amperage) and spark duration that do the actual work to ignite the fuel mixture. So, increased voltage alone does not necessarily lead to more efficient combustion.

To start a well-developed flame front which moves rapidly across the combustion chamber, more "fire" or electrical energy must be placed across the plug gap. The higher energy reduces smolder time, which is the time between the firing of the spark and the development of the flame front. By igniting the intake charge with a big fat spark, smolder time is reduced and the time interval between ignition and maximum cylinder pressure is reduced. This allows the engine's spark lead to be slightly retarded while still allowing maximum pressure to be reached at the desired time. Now the piston spends less time fighting against initial combustion pressure on its compression stroke, which allows more usable horsepower to be made. Combining an electronic ignition with a high powered coil will maximize benefits because spark duration and spark energy can be increased beyond that derived only from a high powered coil.

High powered coils have a higher secondary to primary turn ratio for greater secondary output voltage. Although secondary voltage is only one factor to consider, remember that performance coils put out at least 25 percent more secondary voltage than a stock coil and are needed for a performance engine or even a relatively stock engine to perform at its maximum potential. An engine with a performance coil usually has improved starting, a smoother idle, crisper throttle response and better top-end performance.

Note that a coil must be properly matched to an ignition system by operating voltage (6 or 12 volts) and resistance (ohms). The resistance indicates the amount of current the coil draws from the primary circuit. If necessary, the primary resistance can be checked with an ohmmeter. Using a coil with the wrong resistance can damage an ignition.

Some coils have no polarity and are identified by two unmarked terminals. In this case, either terminal can be connected to the ignition switch or ignition system. On the other hand, coils with polarity are identified by two terminals marked positive and negative. These coils have two secondary windings that are supplied current by one primary winding and must be connected according to the ignition manufacturer's instructions.

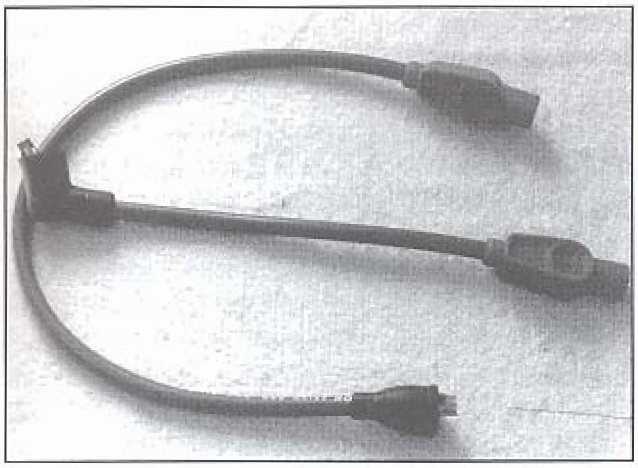
Also, coils are available in either single or dual-tower configurations for supplying energy to either one or two spark plug wires. The required number of coil towers is determined by whether the ignition system is single or dual-fire and whether the cylinder heads are single or dual plugged. Dual plugged heads are normally setup with two dual-tower coils wired in series, although a single four-tower coil can be used.

Performance coils are available from Accel, Andrews, Dyna, H-D Screamin' Eagle, MC Ignition and others.

SPARK PLUG WIRES

Moving the high-voltage from the coil to the spark plug is difficult to do with standard ignition wire. The standard wires are unable to hold the high-voltage, leading to "crossfire" between the wires, engine misfires and reduced power. "Crossfire" is the phenomenon where one spark plug wire induces a voltage into another plug wire, thereby reducing spark energy and possibly causing damage to the a cylinder. Accel, H-D Screamin' Eagle, MSD and Taylor offer 8mm, silicone insulated, high temperature cable that will contain the high voltage to the spark plug.

Some manufacturers, such as Taylor, offer silicone insulated spiral-wound metal core cable that can be used with almost any type ignition.



High-voltage ignition systems need spark plug wires capable of containing a high energy spark so "crossfire" is eliminated. Silicone insulated, spiral-wound, metal core cable does an excellent job of containing high voltage and suppressing radio interference, besides being compatible with most electronic ignitions. Shown are Taylor spiral-wound plug wires.

Spiral-wound cable usually is the best choice for electronic ignitions and is highly recommended for most systems since it does a significantly superior job of maximizing voltage energy and suppressing radio frequency (RF) interference when compared to the stock carbon-impregnated wire. It also is compatible with onboard data recorders and computers. The spiral winding uses an inductive instead of a resistance method for suppressing RF interference. As a result, there is only a slight resistance to current flow. Good quality spiral-wound wire includes a dielectric insulator and silicone insulation for maximum protection against voltage leaks.

Silicone insulated solid metal core wire is frequently used on the race track with magnetos and breaker point ignitions because it does an excellent job of containing high-voltage. But it is not compatible with all electronic ignitions and doesn't offer RF suppression. The RF interference can cause problems with onboard data recorders and computers.

Some racers use heat resistant, glass fiber cloth tubing to sleeve the spark plug wires for additional heat and voltage insulation. Don't allow plug wires touch any metal surfaces and keep them separated at least one half inch from each other to eliminate misfire and "crossfire" problems. Check with your ignition's manufacturer for compatible plug wires and spark plugs.

IGNITION BOOSTERS

An ignition booster is an electronic control designed for use with breaker point ignitions. The booster increases the energy signal to the coil and reduces point arcing. Maintenance is reduced because plugs and points tend to last longer and condensers are eliminated. Dyna offers a booster for single and dual point ignitions.

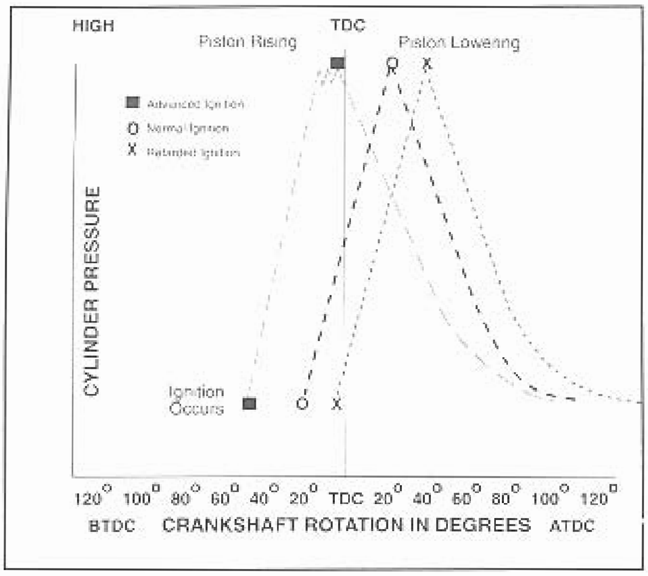
IGNITION ADVANCE FUNDAMENTALS

An established fact is that the cylinder's air/fuel mixture does not explode all at once. If it does explode instantaneously, the pressure wave from the violent explosion can cause engine destruction. To obtain maximum torque, combustion

IGNITION FEATURES GUIDE

Manu-facturer	Single-Fire	Dual-Fire	Magnetic Sensor	Optical Sensor	Mechanical Advance	Electronic Advance	Adjustable Advance	RPM Limiter	Separate Cyl. Timing	Multi-Spark	High RPM Retard	VOES Switch
Accel		Yes	Yes			Yes	Yes	*	Yes			
Carl's CDI Magneto	Yes	Yes	Yes					*				
Crane HI-4	Yes	Yes	Yes			Yes	Yes	Adjustable	Yes			Yes
Dyna-S	Yes	Yes	Yes		Yes			*	Yes			
Dyna 2000	Yes	Yes	Yes			Yes	Yes	Adjustable				Yes
MC Power Arc	Yes			Yes		Yes	Yes	Adjustable	Yes	Yes	Yes	Yes
Screamin' Eagle		Yes	Yes			Yes		*Fixed at 8000				Yes
1200cc XL Module		Yes	Yes			Yes		Fixed at 6250				Yes
Stock Electronic		Yes	Yes			Yes		Fixed at 5250				Yes

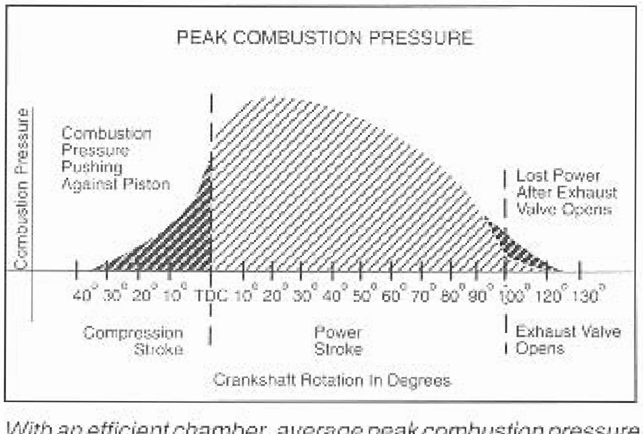
Table 8.2 *External RPM Limiter Can Be Added. Refer to Appendix E for additional ignition information.



At high rpm, if ignition occurs late, the piston will be extremely low in the cylinder when combustion pressure peaks. This results in the piston being too low in the cylinder to receive maximum pressure. If ignition takes place too early, excessive combustion pressure will push against the piston moving up the bore. When ignition is extremely early, detonation will occur. Each engine rpm requires a different ignition advance. However, combustion chamber design, temperature, fuel octane, air/fuel ratio and engine load also determine the optimum advance.

pressure must be developed smoothly and timed correctly so the maximum amount of energy is expended forcing the piston downward.

The optimum point for maximum cylinder pressure is very narrow. If maximum cylinder pressure is developed just a few crank degrees too early or too late from optimum, power drops off substantially. If ignition is too late, power



With an efficient chamber, average peak combustion pressure usually occurs between 12 and 17 degrees after TDC and 90 percent of the total pressure is achieved by 45 degrees after TDC. Notice how some pressure works against the piston before TDC and that the exhaust valve opening causes a loss of pressure.

drops off and engine heat increases because more cylinder wall surface is exposed to burning gases. If ignition occurs slightly early, power is reduced because excessive pressure is pushing against the piston as it is moving up the bore towards TDC. However, if ignition takes place extremely early, combustion chamber pressure increases dramatically and detonation occurs.

Detonation is the uncontrolled combustion of the intake charge taking place simultaneously at several places in the combustion chamber. This results in extremely high cylinder pressures, which can cause engine damage. Detonation generates a pinging noise that results from the actual collision of the multiple flame fronts in the combustion chamber.

To achieve the most power, maximum cylinder pressure must occur somewhere between TDC and the point at which the exhaust valve opens (generally between 42 and 65 degrees BBDC for the Big Twin). For this to happen, the ignition system must take into account the time interval that is required for the air/fuel mixture to burn. This requires ignition to take place before the piston reaches TDC.

Since ignition occurs before TDC, the piston is still rising in the cylinder. When the air/fuel mixture ignites, pressure increases due to the heat of combustion and starts to push the piston down the cylinder. This is what happens when the engine kicks back during the starting process. The increase in cylinder pressure due to the engine's mechanical compression is at its highest level when the piston is at TDC. However, the pressure increase attributed to combustion is still on the rise so maximum cylinder pressure occurs after TDC.

For maximum power, it has been determined that maximum cylinder pressure should occur approximately 12 to 17 degrees ATDC for an efficient combustion chamber and maybe as late as 30 degrees ATDC for one with an inefficient design. After this point, the downward movement of the piston reduces pressure quicker than the expanding gases can increase it. So, to obtain maximum cylinder pressure shortly after TDC, ignition must take place before the piston reaches TDC (usually between 5 and 50 degrees before TDC).

If ignition is set to occur at 15 degrees BTDC at 1000 rpm, there is a given time interval (about .0025 second) between ignition and TDC. When engine rpm is increased to 2000, the time interval is reduced to half the original time interval and halved again when it is increased to 4000 rpm. As rpm increases, less time is available for combustion. This requires more ignition advance to achieve maximum cylinder pressure near the beginning of the power stroke. However, beyond approximately 3000 rpm, the amount of ignition advance remains almost steady because the reduced time available for combustion is offset by a quicker burn rate, which is attributed to increased combustion chamber turbulence.

TIME AVAILABLE FOR COMBUSTION

RPM	TIME FOR COMPRESSION STROKE	TIME FOR FUEL BURN (35° BTDC TO 110° ATDC)
1000	.030 seconds	.0242 seconds
2000	.015 seconds	.0121 seconds
3000	.010 seconds	.0081 seconds
4000	.0075 seconds	.0060 seconds
5000	.0060 seconds	.0048 seconds
6000	.0050 seconds	.0040 seconds
7000	.0043 seconds	.0035 seconds
8000	.0037 seconds	.0030 seconds

Table 8.3

COMBUSTION ELEMENTS

Engine speed is only one element that governs proper ignition advance. The rate at which the air/fuel charge burns is another and it is a function of charge density, flame speed and the area of the flame front.

Combustion time is *lengthened* by the following conditions:

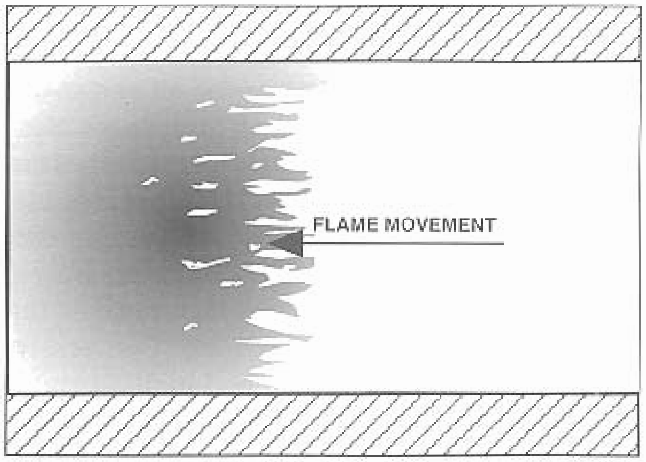
- Leaner mixture
- Less turbulence
- Lower temperature
- Lower pressure
- Lower air density
- Residual combustion gases

Combustion time is *shortened* by the following conditions:

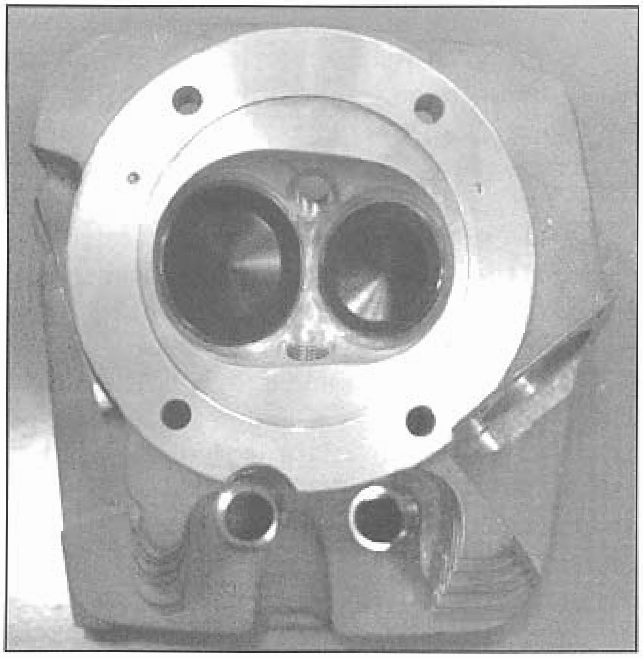
- Richer Mixture
- More turbulence
- Higher temperature
- Higher pressure
- Higher air density

The most chemically correct air/fuel mixture is referred to as stoichiometric. The exact ratio varies based upon temperature, humidity and air density, but it is normally just under 15:1. Maximum power, however, is achieved at a ratio closer to 12:1. Mixtures that are richer or leaner than stoichiometric burn slower. However, if you measure the burn rate for a lean mixture, then enrich the mixture and measure the rate again, you will find the burn rate will increase up to the point of stoichiometric and then it will decrease. The key point is that a richer mixture burns faster than a leaner mixture up to the point of stoichiometric. Rich mixtures enhance the speed at which oxygen burns and promote a more complete burn up to the point of stoichiometric, then the excessive amount of fuel molecules insulates one set from another. This reduces heat transfer along with the burn rate.

Turbulence has the greatest influence on the speed at which an air/fuel charge burns. High turbulence causes greater fuel atomization, which results in smaller fuel droplets that oxidize (burn) more quickly. High turbulence also fans the flame front into many ragged or bushy edges, which increases the area the flame presents to the air/fuel charge. This causes small portions of the flame front to break away from the main kernel and hurl forward. It also significantly increases the burn rate over a non-turbulent fuel charge and allows for reduced ignition advance.



High turbulence fans the flame front into many bushy edges, which improves flame propagation and speeds combustion.



A dual plug, bathtub combustion chamber provides a compact space with short flame travel and the twin squish shelves provide high turbulence. With a 29 degree included valve angle, the chamber roof remains shallow and allows relatively high compression with a flat top piston. Together, these characteristics shorten the burn time and reduce the required ignition advance, which lowers the potential for engine damaging detonation. Shown is an STD head setup for 4-inch bore cylinders giving 114 ci. Photo courtesy of Great Lakes Cycle.

Higher compression and higher temperature create higher pressure. Higher pressure compacts the air/fuel charge more, which allows the flame front to travel from one set of molecules to another set more quickly. This increases the speed of combustion and requires less ignition advance.

The density of the air/fuel charge affects the speed at which the mixture burns. Low ambient temperatures and low altitudes increase the density of air. Also, a wide open throttle position, which decreases manifold vacuum increases the density of the air/fuel charge. The higher the air/fuel density, the more mass there is to compress, thereby creating higher combustion pressures. This reduces the burn time, which reduces the required amount of ignition advance. Conversely, high ambient temperatures and high altitudes reduce air density. Also, nearly closed throttle positions create high manifold vacuum, which reduces mixture density. Low air/fuel density lengthens the burn time, thereby increasing the required ignition advance.

Inefficient exhaust systems allow combustion residue to remain in the cylinder and contaminate the incoming fuel charge. The exhaust gases separate the fresh fuel molecules and absorb some of their heat. This slows the burn rate, which increases the ignition advance requirements.

Combustion chamber design can also affect the burn rate and total burn time for the intake charge. A large, deep combustion chamber requires the flame front to travel a long distance, which increases the total time required for combustion. A single spark plug located on the side of a large chamber contributes to long flame travel. For high compression, a large chamber requires a high dome piston. The high dome frequently obstructs the flame front and increases the total burn time. Also, a hemispherical chamber such as the Shovelhead's has low turbulence, which decreases the flame front and slows down the burn rate. All of these conditions increase the required amount of ignition advance.

ADVANCE CURVE

Ignition timing can be used as an engine's power lever. It can be used to make more power, less power or to protect an engine from damage. More advance doesn't automatically produce more power because every engine has a specific requirement at a given rpm. For example, a street engine needs to make power from about 2000 to 6000 rpm. To make more power over such a wide range without detonation, the ignition's timing must change with different rpm levels.

The time interval between ignition and when the piston reaches TDC is expressed in degrees of flywheel (crankshaft) rotation. The greater the number of degrees before TDC is, the more advanced the timing. Reducing the number of degrees is referred to as retarding the timing.

At startup and idle speeds a Big Twin engine is usually setup with between zero and 15 degrees advance BTDC. As engine speed increases, the advance increases to approximately 27 to 42 degrees BTDC to allow maximum cylinder pressure to occur at the optimum point ATDC. Total advance is usually all in by 1800 to 2500 rpm. At approximately 3000 rpm, greater turbulence generated by the higher rpm increases the burn rate, which offsets the reduced time available for combustion. As a result, the need for more ignition advance levels off and becomes nearly constant up to the maximum engine rpm.

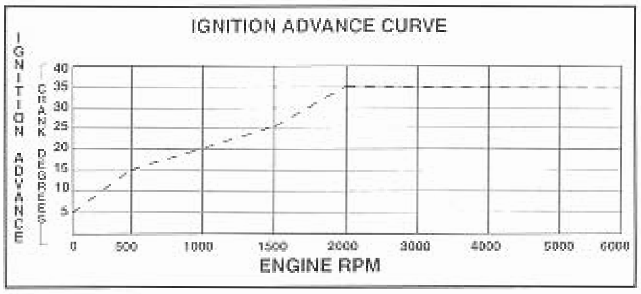
Each engine design has its own ideal advance curve and amount of ignition advance. A typical Evolution engine requires about 35 degrees advance to make maximum power in its higher rpm ranges. An Evolution Sportster needs about 40 degrees advance and iron Sportsters typically run best with 38 to 48 degrees advance.

Determining the best ignition advance curve for an engine is time consuming and is best done at the race track. However, it is most easily accomplished on an engine dyno. Although few of us can afford access to a dyno, understanding the concepts and principles involved will help you tune your engine's ignition for maximum performance.

The dyno process is called *Best Torque, Minimum Advance.* This process is based on the fact the internal combustion engine produces its best performance when ignition timing is adjusted for maximum torque at wide open throttle (WOT). Maximum power usually occurs when the timing is just short of detonation.

Several factors determine the detonation limit of any engine and some of the major ones include: combustion chamber size and design, amount of turbulence, piston dome shape, spark plug location, bore size, compression ratio, cam specifications, air/fuel ratio, fuel inlet temperature, and air/fuel density. Additionally, humidity, engine load and rpm enter into the picture.

During dyno testing, an engine is run at WOT under load. The ignition timing is advanced until there is a one percent loss in torque; then the ignition is retarded until there is a one percent loss in torque. Half way between the advance and retard points normally results in the best ignition advance. The testing is done at various rpm levels to generate an engine's advance curve requirement. This is because the ideal spark advance at 3000 rpm is not necessarily the same as that at 5000 or 7000 rpm or any other rpm for that matter.



This is one example of a performance advance curve. The factory advance curve is set for emissions control and combustion chambers with low turbulence. Some electronic ignitions allow an adjustable advance curve for lots of flexibility. Centrifugal advance mechanisms can be modified with different springs, weights and pins for better performance.

The essence of this is that ignition timing requirements vary. At cranking speeds, the plugs should fire relatively close to TDC. But once past idle, ignition timing should advance fairly quickly to its highest point.

Stock early model electronic ignitions provide a long, slow advance curve to satisfy emissions requirements. The late model electronic ignition includes a slightly more aggressive curve. An example of a performance advance curve would be five degrees advance BTDC at initial startup, increasing to 15 degrees at 500 rpm, 25 degrees at 1500 rpm, and finally full advance (approximately 35 to 40 degrees) at 2000 rpm. Magneto ignitions usually run a fixed advance where the entire advance is in effect from startup to maximum rpm. This can result in harder starting, especially if the engine has high compression or large displacement and is kick started.

DETONATION

Normal combustion is not a violent explosion. Instead it is a process where the air/fuel mixture burns in a smooth, steady motion with the flame front expanding from the spark plug until the entire mixture is burned. If unburned air/fuel mixture is excessively elevated in pressure or temperature, it will explode in several spots without waiting for the flame front to ignite it. This results in a violent collision of flame fronts *after* the spark plug has fired. In this situation, the pressure rise is quick and violent and can result in engine damage.

No engine can reach maximum power with detonation present and even mild detonation can increase physical and thermal stresses to the point of damaging pistons, cracking cylinder heads, destroying rods and bearing, and shattering spark plug insulators. The noise resulting from detonation is caused by the collision of multiple flame fronts in the combustion chamber and is frequently inaudible because it can be drowned out by exhaust sounds.

Excessive ignition advance, high compression and low octane fuel are common causes of detonation. Too much ignition advance adds pressure to the cylinder pressure generated by the engine's dynamic compression. In effect, it's as if the mechanical compression ratio was raised. Reducing ignition advance or the mechanical compression ratio reduces peak cylinder pressure, which reduces the possibility of detonation. Raising the fuel's octane level increases the fuel's ability to react in a controlled and orderly manner under high cylinder pressures and temperatures. Oil contamination of the intake charge due to worn piston rings, valve guides or oil seals will severely reduce the fuel's octane level and increase the potential for detonation.

Time also plays a part in detonation. The slower combustion takes place, the longer unburned air/fuel mixture is held at high pressure and temperature, and the more inclined detonation is to occur. A lean air/fuel mixture is more sensitive to detonation because it burns slower and hotter than a rich mixture. Lugging an engine at low rpm and large throttle openings creates a high load condition and poor fuel atomization. This increases heat and pressure while the time for combustion is lengthened. It also encourages detonation. The Big Twin is fitted with a V.O.E.S., which retards ignition timing at high load conditions to counteract this condition.

Assuming ignition timing and fuel octane is satisfactory, increasing the speed of combustion is the best protection against detonation. High turbulence atomizes fuel particles into smaller droplets for quicker combustion. It also creates a bushier flame front that consumes the air/fuel mixture more quickly. Additionally, a small, compact combustion chamber reduces the distance the flame front needs to travel and leads to reduced combustion time. Dual spark plugs also reduce combustion time as evidenced by the reduced ignition advance required when using two plugs. Low dome pistons that do not shroud or obstruct flame travel likewise help to reduce burn time.

Black specks or small shiny black and purple colored balls attached to the plug's porcelain are indications of detonation. Reducing the time required for combustion allows the use of a higher compression ratio for a given fuel octane without incurring detonation. Whenever detonation is suspected, immediately retard the ignition, enrich the fuel mixture or do both.

PRE-IGNITION

Pre-ignition is frequently confused with detonation. Where detonation occurs *after* ignition of the intake charge, pre-ignition ignites the charge *before* the spark plug firing. Pre-ignition is caused by a hot spot, which results from a red glowing body in the combustion chamber. Causes of hot spots include an overhanging gasket, sharp or thin piston dome areas, valve surfaces, combustion deposits, or an overheating spark plug. The glowing body retains enough heat to ignite the fresh mixture prematurely.

A common cause of pre-ignition is too hot of a spark plug heat range. This can easily result in the plug's thin ground electrode or positive center electrode to run hot enough to glow and prematurely ignite the charge. This is one of the reasons spark plugs are available in different heat ranges. Signs of too much heat include glazing of the plug's porcelain, bluing of the center electrode or the ground electrode turning green. Melted center or ground electrodes are signs of pre-ignition.

Pre-ignition will always occur *before* ignition takes place while detonation always occurs *after* ignition starts. Pre-ignition is similar to operating under excessively advanced ignition timing. As a result, pre-ignition can lead to detonation, but detonation will not lead to pre-ignition. Pre-ignition is caused by hot spots and destroys engines by heat. On the other hand, detonation is caused by high pressures, low fuel octane or air/fuel separation and destroys engines by mechanical shock. Pre-ignition conditions will usually melt away the spark plug electrodes while detonation generally leaves the plug's electrode intact, yet shatters the ceramic insulator.

DUAL SPARK PLUGS

Adding a second spark plug to the cylinder head effectively reduces the distance the combustion's flame front needs to travel. By installing a second spark plug opposite the original plug, the air/fuel mixture starts combustion at two separate locations, instead of one. This essentially reduces the distance between the plug and the farthest part of the chamber. The time required for combustion decreases and a more complete and uniform burn of the air/fuel mixture is achieved, particularly with a high dome piston. Furthermore, the potential for detonation is greatly reduced and maximum horsepower can be realized.

Since two spark plugs reduce combustion time, the point of spark ignition can be delayed so the timing of maximum cylinder pressure remains correct. Normal timing for the Big Twin is approximately 35 degrees BTDC. This should be reduced (retarded) by 5 to 8 degrees, which results in timing between 27 and 30 degrees BTDC. Since the crankcase timing hole is about 10 degrees in width, you can adjust the ignition advance by setting the flywheel timing mark at the front edge of the hole or just beyond the front edge instead of centered. This will provide the 5 to 8 degrees retard. Remember, this is only an example. For your application, subtract 5 to 8 degrees from the optimum ignition advance for your engine when running a single plug. Be aware that failure to retard the ignition timing will nullify any potential benefits of dual plugs. Also take note that dual spark plugs (4 plugs total) require the use of two ignition coils (dual tower each), and both coils must have the proper voltage and ohm rating.

The major benefits of dual spark plugs are higher performance and a slightly smoother running engine. For a given fuel octane rating, dual plugs allow the use of a higher mechanical compression ratio without incurring detonation. Also, for a given mechanical compression ratio, dual plugs will allow the use of a lower fuel octane rating. The larger and more inefficient the combustion chamber and the higher the piston dome, the greater the benefits from dual spark plugs.

ADVANCE MECHANISMS

The function of an ignition advance mechanism is to increase the spark advance up to a maximum amount as rpm increases. Ignition advance for the Big Twin engine is controlled by either a mechanical advance or electronic advance mechanism.

All engines through 1979 came equipped with a mechanical advance unit, which uses centrifugal force to move spring loaded weights to advance or retard the ignition based on engine rpm. Starting in 1980, ignition advance is controlled by an electronic system having two advance curves — one for part throttle and the other for full throttle operation. Also, starting about 1984, a vacuum operated electric switch (V.O.E.S.) was added (to some models) to retard the ignition from full advance under high engine load (low intake manifold vacuum) conditions.

Although the Big Twin was never equipped by the factory with a magneto, take note that magnetos generally use a fixed ignition advance setup where the timing is set to full advance and does not vary with engine rpm.

Some electronic ignitions include a programmable advance curve, which is adjustable through either switches or screws. This allows the curve to be optimized for the level of engine modifications, weight and gearing of the bike, quality of fuel and style of riding.

MECHANICAL ADVANCE

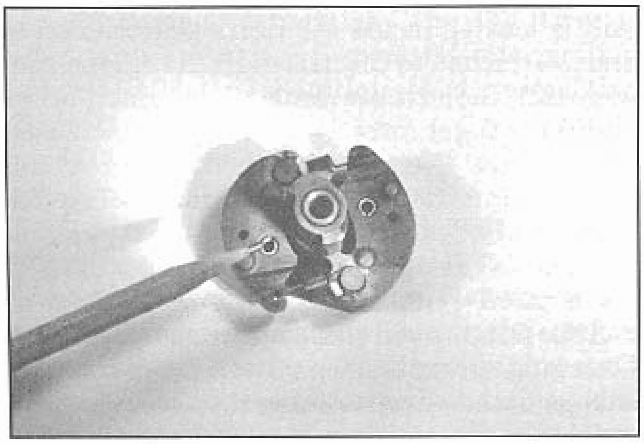
A mechanical advance mechanism uses centrifugal force, weights and springs to increase the amount of advance. The centrifugal advance unit is located inside the cam's gearcase cover and is attached to the camshaft. As engine rpm increases, the advance mechanism spins quicker and centrifugal force increases. The increased force causes the weights to overcome spring tension and rotate, thereby increasing the amount of advance BTDC. Stock weights and springs are designed to give total advance at about 2000 rpm. Lighter tension springs or heavier weights will speed up the advance curve. Changing spring tension gives a more dramatic change to the advance curve than changing the weight of the weights. You can also use one heavy and one light tension spring. Removing two grams at a time from the weights will slow the advance curve.

To function properly, the entire mechanical advance unit must work freely and smoothly with no jerking or dragging. Severely used advance units sometimes start sticking or operating erratically. Any problems can be detected by first removing the timing access cover and grasping the points or electronic sensor cam. Next, turn the cam as far as it will turn and then release it. It should return quickly and smoothly to its original position.

For an engine that is running, take note of how quick the engine drops down to idle after you rev it up. If it idles around 1300 to 1500 rpm for a while before dropping down to idle rpm, the mechanism is probably sticking. In either case, a sticking advance unit indicates it is worn, dirty or both. With use, the advance weights wear and the springs lose tension. Also, many mechanisms are low in quality and quickly wear out. This allows the unit to either stick or the weights to bounce, which results in inaccurate and irregular timing. The weight pivot pins can also deflect a few degrees during high rpm operation, causing erratic timing.

When inspecting a mechanical advance unit, look for worn pivots and pivot holes, weak springs and sloppy clearances. Also, check how smooth the weights pivot on their post.

Clean the entire assembly and make sure the holes in the weights and their pivots are perfectly clean. Carefully assemble the unit without stretching the springs and lightly lube the pivots and weights with molybdenum disulfide lubricant.



The pencil points to one of the mechanical advance retard pins. A pin is pressed into each weight. The larger the pin diameter, the less the centrifugal advance will be. Refer to the text for determining optimum pin size.

When assembling the advance mechanism to the camshaft, make sure it sits squarely against the cam and that its roll pin is located in the cam's notch. Also, be careful when screwing the long retainer bolt into the cam and only tighten it according to factory specifications.

Rivera Engineering sells a stainless steel mechanical advance mechanism that has heat treated polymer coated weights, high quality cadmium coated springs, besides stainless steel pivot pins and backing plate. This unit offers smooth operation and durability. D&S Performance sells American-made springs made from premium-grade spring steel that are designed to give full advance near 1800 rpm.

When setting up a centrifugal advance mechanism there are a few points you should consider. A stock mechanical advance unit gives about 30 degrees of centrifugal advance. When you add 5 degrees of initial advance, you get a total of 35 degrees advance. Let's say your heads are dual plugged and you want to run 30 degrees total advance. If you set the engine's total advance to 30 degrees with a timing light, you will now have zero initial advance and 30 degrees centrifugal advance. This will cause poor idling and "coughing" and hesitation when the throttle is quickly opened because there is too little advance (actually zero degrees) at idle. To correct the problem, you must reduce the centrifugal advance to 25 degrees. This can be done by increasing the diameter of the mechanical advance's retard pins. Stock pins are about .187-inch diameter. You may have to use pins up to .250-inch in diameter. You can make new pins or press brass ferrules over the existing pins to increase the diameter.

Before you can determine pin size, you need to determine two things: the total ignition advance the engine runs best at and the advance it idles best at. Total advance is determined by performance testing on the race track. The best idle advance is determined by changing the timing while the engine is idling. Use the setting that gives the best starting and idling. A dial-back-to-zero timing light can be used to verify the amount of initial advance after you have it set. Each engine is different, but many engines like 10 to 15 degrees initial advance.

For the following example lets assume 10 degrees initial advance and 30 degrees total advance is best. If we subtract the 10 degrees initial advance from the 30 degrees total advance, there is a remainder of 20 degrees. This means that 20 degrees centrifugal advance is needed. However, the stock mechanical advance mechanism gives 30 degrees. By increasing the diameter of the retard pins, the centrifugal advance can be reduced to 20 degrees.

You can increase the retard pin's diameter in .025-inch increments so the engine's centrifugal advance is reduced to 20 degrees. This will give a total advance of 30 degrees while keeping the initial advance at 10 degrees.

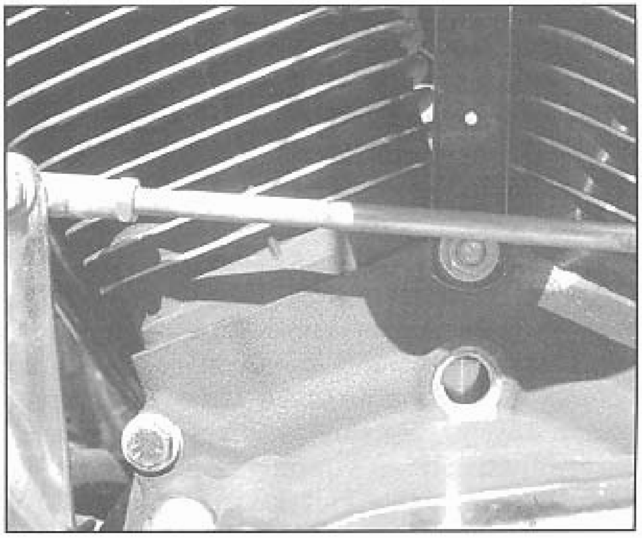
SETTING IGNITION TIMING

Procedures for adjusting the Big Twin's ignition timing vary with different ignition systems. However, there are several techniques and considerations that are common to all ignitions.

All ignitions are adjusted based on a specified number of crankshaft degrees BTDC. One or more timing marks are placed on the side of the left flywheel. These marks can be viewed through the timing inspection hole located on the left crankcase just below the bottom of the cylinders. Some flywheels have one front cylinder advance mark while others also have a TDC and rear cylinder advance mark. The advance mark for the front cylinder and rear cylinder (if it exists) are the most important. Due to variances of engine machining tolerances, the flywheel marks may be slightly inaccurate. So, if you're building an engine from the bottom up, you should verify the accuracy of all timing marks because key measurements are based on them.

Verification requires mounting a degree wheel on the engine sprocket shaft, locating TDC and then checking the advance mark for each cylinder. In some situations, a mark can be one or two degrees off. If the engine is assembled, you can verify each mark's position as long as a degree wheel can be mounted on the sprocket shaft. In this case, you need to use a TDC locating tool, which screws into the spark plug hole to determine TDC. Crane Cams and others sell this tool.

Ignition timing is normally set according to the front cylinder's timing mark. Some ignition systems allow setting the front and rear cylinder timing separately. For stable timing, the mechanical advance unit (if present) must work smoothly and not have excessive clearances. Also, the cam's end play should be set correctly, although this is not as important as it is for the Sportster engine.



The crankcase timing hole is about 10 degrees in width. Timing the engine with the timing mark at the front edge of the hole retards the timing about 5 degrees from the mark's standard value. Conversely, placing the mark at the hole's rear edge advances timing 5 degrees. The position of all flywheel timing marks should be verified since machining tolerances can cause inaccuracies.

For accuracy the timing must be set when the engine is at full advance. This means the engine must be running fast enough for full advance to take effect, which is usually at least 2000 rpm. The timing procedure requires the use of a timing light. A timing light is a strobe light that freezes motion with a short flash of light. Timing a running engine is most easily done with two people. With the engine running, one person can aim the timing light at the inspection hole while the other person adjusts the ignition backing plate in the cam gearcase cover.

Normally the ignition is adjusted so the appropriate flywheel timing mark registers in the middle of the inspection hole. Since the inspection hole is about 10 degrees in width, adjusting the ignition with the mark positioned at the hole's back edge advances the timing five degrees. Conversely, positioning the mark at the hole's front side retards the timing five degrees. Use a good quality inductive timing light and hold a rag over the timing inspection hole to control the oil spray until it is time to flash the light. And when you're all done, buy the guy who is holding the timing light a drink because he will probably end up with a face full of oil.

Tech Products makes the EZ-Tyme degree plate that mounts inside the primary chain housing. The plate simplifies timing by eliminating the need for removing the timing inspection hole plug when the engine is running. It can also save you from getting a face full of oil.

The engine can be timed when it is stopped (static timing), but this method is not as accurate as when it is running. Some electronic ignitions, such as the Power Arc, include an LED light to simplify static timing. To time an engine statically, both spark plugs must be removed and the front cylinder must be positioned on its compression stroke with its timing mark positioned in the middle of the timing hole. The following procedures assume the timing mark is positioned in the middle of the hole. However, be sure to position it wherever is best for your engine.

You can verify the engine's compression stroke by removing the front cylinder's intake valve pushrod cover (the one closest to the carburetor) and then turning the engine forward until the pushrod starts moving downward. Always turn the engine forward — never backward. Then, very slowly turn the engine with the kick starter lever until the correct timing mark appears in the center of the hole.

If the engine only has an electric starter, place the transmission in high gear and raise the rear wheel off the ground. Turn the rear wheel by hand and follow the above procedures for centering the timing mark in the inspection hole. Someone may need to apply the rear brake to hold the flywheels from moving.

For breaker points, the Acu-Time tool is available from CCI to help hold the points cam in the fully advanced position. This tool allows the engine to be timed in the full advanced position rather than the retarded position.

When the timing mark is correctly positioned, connect a test light or ohmmeter to the points to determine when the points start to open. Rotate the points backing plate until the light goes out or the ohmmeter indicates resistance. Then lock the plate down.

For stock electronic ignitions, connect a spark plug to the front cylinder spark plug wire and ground the plug. Turn the ignition on and slowly turn the engine forward until the plug fires. You will hear it snap. The flywheel timing mark should be centered in the inspection hole when the plug fires. If the mark is not centered, adjust the ignition sensor's backing plate as necessary and then repeat the procedure.

IGNITION SUMMARY

Up to this point a great deal has been said about ignition systems, but what conclusions can be drawn and how can the information be applied towards increasing performance?

Keep in mind that the ignition system should be one of the first four components replaced for greater performance — the others being the exhaust system, carburetor and camshaft. The main reason for replacing the stock electronic ignition is to remove its built-in rpm limiter. Another purpose is to allow for a more aggressive advance curve. When the stock ignition is replaced, the stock coil should also be replaced with a high-voltage coil for a higher energy, longer duration spark. No aftermarket performance ignition can work near its potential without a high-voltage coil(s).

If you don't have the money to replace the complete ignition system, you can replace only the stock coil and spark plug wires with high-performance items at a relatively low cost. This will provide a much stronger spark for fewer misfires, smoother idle, better throttle response and stronger acceleration.

Most winning racers are running an electronic, single-fire ignition system with high-voltage coils and top quality spark plug wires. These systems use either a mechanical or electronic advance mechanism. Some systems are locked up with a fixed amount of advance. Some ignition systems include a built in rpm limiter while others require the additional cost of a separate limiter. Regardless of the ignition you use, be sure to protect your engine from over revving and missed shifts with a rpm limiter.

Evolution hydraulic valve lifters pump up at about 6300 rpm (depending on machined tolerances), so consider this when setting a rev limit.

When the lifters pump up, valve control is lost. This reduces power and can cause engine damage. RPM should be limited to 6200 for a non-serious performance engine. S&S Products makes an easily installed Evolution lifter kit that allows the lifter to work as a hydraulic at low rpm and as a solid at high rpm. This kit solves the lifter pump up problem. Some cams require full solids, so lifter pump up is not a consideration.

Ignition systems with a programmable advance capability offer a lot of flexibility, especially for the street rider and the multi-spark capability has proved to increase horsepower on many engines.

For race engines, eliminating the advance mechanism and locking up the ignition so advance is all in at startup can result in more stable timing. If cranking the engine with the electric starter motor is a problem, consider wiring the ignition to a switch separate from the starter motor circuit. Then, after the engine is turning, the ignition can be turned on.

Take note that large displacement, high compression engines may need a modified starter motor to crank the engine. Carl's Speed Shop has a high torque starter motor for 1990 and later Big Twin engines.

TUNING IGNITION ADVANCE

There are several factors that determine the optimum ignition advance for an engine combination. Some of the factors are internal to the engine while others are external. The internal factors include: combustion chamber design, chamber turbulence, piston dome shape, mechanical compression ratio, cam specifications, number of spark plugs per head, bore diameter, rod/stroke relationship, air/fuel ratios, fuel quality and engine speed. External factors include: total bike weight, gear ratios, air temperature, altitude and humidity.

Stock ignition advance for an Evolution or Shovelhead engine is 35 degrees BTDC. For a slightly modified Evolution engine, two to five degrees on either side of this setting is probably close to optimum. The Shovelhead generally needs between 35 and 45 degrees advance because of its large inefficient hemispherical combustion chamber. A small highly turbulent combustion chamber requires less advance than a large, non-turbulent design.

Large stroker engines usually work well with slightly less advance than stock stroke engines. Also, the higher the compression ratio, the faster the mixture's burn rate will be. Consequently, less timing is required. For a large displacement high compression stroker engine, start with about 30 degrees advance and work up from there. Some 100 plus cubic inch stroker engines with 14:1 mechanical compression and single plugs are running as low as 28 degrees advance. In general, the higher the compression ratio, the less ignition advance that is required for a given engine combination.

When running dual plugged heads, be sure to reduce the timing five to eight degrees from the above values, otherwise the potential benefits will not be realized. Also, if you revert to single plugs, don't forget to bump the timing back up because the retarded timing will cause excessive engine heat and possible engine damage.

Remember, these values are a general guide and should not to be construed as gospel. But they're a good place to start and should be helpful.

Before fine tuning the engine's ignition advance, the carburetor must be adjusted for the correct air/fuel ratio. This is because different air/fuel ratios burn at different rates. When tuning the engine, first tune the carburetor, then the ignition advance, gearing and exhaust system. If you don't have a baseline tune for your engine combination, start with cold plugs, rich air/fuel mixture and an ignition advance that you consider slightly less than optimum. After the carburetion is tuned, increase the ignition advance in two degree increments until the engine slows down or signs of detonation become present. Then back off the timing two degrees.

The following tuning tips are associated with ignition timing:

- For a given engine combination, the optimum ignition timing will remain the same until the combustion chamber design, piston dome shape or any variables affecting ignition timing are changed.

- Higher compression ratios require less ignition advance because the mixture burns quicker under higher pressure.

- Low gears can handle more advance than high gears.
- An engine with little or no load can take more advance than one with high load.
- A light bike with a low gear ratio (high numerically) can accept more ignition advance than a heavy bike with a high gear ratio (low numerically).
- An excessively advanced ignition will allow the engine to run strongly in low gears and turn high rpm, but the engine will not pull hard in high gear.
- Retarded timing reduces throttle response at low speeds, causes sluggish operation and won't allow the engine to turn high rpm.
- Retarding the timing a couple of degrees when in high gear can sometimes help top-end power because the engine is more heated than when it left the starting line.
- If you add more advance and the bike's 60 foot time improves but overall E.T. is hurt, the engine needs some advance taken out on the top-end. An ignition with a top-end retard can sometimes help in this situation.
- Severely retarded ignition timing exposes more cylinder wall surface to the combustion process. Therefore, engine heat is increased and the exhaust system's energy waves speed up. This condition overheats the engine, reduces power and changes the optimum tuned length for the exhaust system.
- Generally speaking, as you go up in altitude, timing can be advanced a couple of degrees over the optimum timing at sea-level.
- Since higher air density increases the burn rate, less advance may improve performance.
- Rich mixtures burn cooler than lean mixtures, so the engine can tolerate more ignition advance.
- Since different blends of fuel burn at different speeds, ignition advance must be reestablished when changing fuel blends.
- The detonation level is not fixed, but can move either up or down depending on air temperature, ignition advance, fuel octane, engine load, etc.
- A long duration cam usually requires a quicker advance curve.
- Use a projected-core spark plug whenever possible.
- If necessary, use a different spark plug heat range in each cylinder.
- A spark plug with worn or rounded electrodes requires more voltage to fire than one with sharp square edged electrodes.
- Any component that increases resistance, such as leaking spark plug wires, will limit the size of the plug gap that can be used.
- In general, any reduction in ignition voltage or increase in cylinder pressure will require a smaller spark plug gap.
- Whenever you switch to an ignition system that produces an exceptionally high amount of energy, you need to run through your tuning procedure to reestablish the optimum fuel mixture and ignition advance.
- Minimize all clearances that can affect ignition timing. This includes bushing clearances and tolerances in the cam's drivetrain and ignition advance mechanism. Loose clearances in these components will increase the ignition's hysteresis or lag time variables and affect ignition timing.
- Timing based on the front cylinder flywheel timing mark offers no guarantee that the rear cylinder is timed correctly. Therefore, if your ignition can time each cylinder separately, be sure to use it.

Building a large displacement engine today is easier than in the past. But tuning it involves a combination of art and science. Tuning is often the difference between winning and losing and reading spark plugs plays an important part in determining the tune. Joe Tud on his 115 ci Evo Pro Gas dragster. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

- A magneto's breaker cam lobe may need to be modified (indexed) for accurate rear cylinder timing. If the rear cylinder does not fire at the same time BTDC as the front cylinder, the magneto's rear cylinder cam lobe can be polished with a fine stone until both cylinders fire accurately.

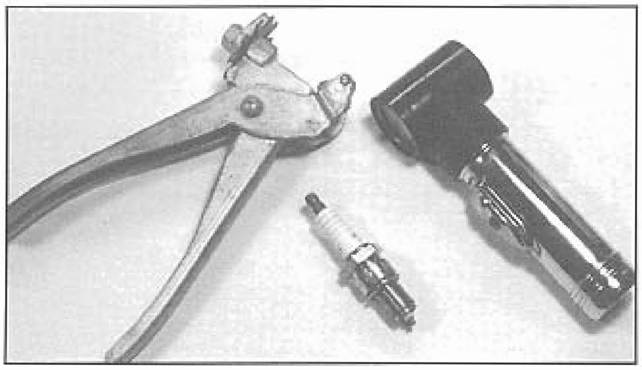
- When using a 12 volt total loss ignition system, coil saturation is sometimes enhanced by connecting three 6 volt batteries in series and using a ballast resistor to keep the voltage down to about 12 volts.

- Be sure to use a good ground from the battery to the engine and battery to the ignition system.

SPARK PLUG READING

Spark plugs play an important part in the engine's ability to make power and they are an important tool the racer can use in determining the optimum air/fuel ratio and ignition timing. Plugs also tell a compelling story about the ability of the pistons and valves to control oil. Furthermore, plugs help the tuner determine the fuel distribution to each cylinder and the cylinder's cooling. Reading a spark plug is one of three methods for determining the state of an engine's tune. The other two methods are checking the piston dome for color and the color of the exhaust header just past the port.

Reading spark plugs is actually a combination of art and science. It takes knowledge and some practice to read plugs accurately. Astute plug readers tend to be reluctant to share their plug reading knowledge because tuning is frequently the difference between winning or losing. Almost anyone can build or have built for them a large displacement engine with cutting-edge parts. However, not everyone can tune it to perform up to its maximum potential. The following information won't make you an expert at plug reading, but it will start you in the right direction.



If you want to win, you need to know how to tune your engine. Spark plugs play an important part in tuning because they are sensitive to air/fuel ratios and can provide clues about ignition timing, oil control and cylinder heat. A spark plug flashlight with magnification will help you read the plug where the ceramic attaches to the body. Also shown is a plug gapping tool that keeps the electrodes parallel to each other.

If you're working with a slightly modified engine, your starting spark plug may be close to a stock heat range and type. If you're working with a new engine combination, you need to determine a starting heat range and plug type. Plugs transfer heat by passing it from the ceramic insulator to the steel body. The longer the ceramic insulator, the farther the heat has to travel and the hotter the plug runs. Most of the plug's heat is transferred in this manner. A plug works best at about 800 degrees F and if it is much colder than this, misfires will increase or the plug will foul. When the plug runs much hotter than 1200 degrees F, it overheats and pre-ignition and detonation can set in.

For the most part, plug type is determined by the design of its ground and center electrode, the distance the ceramic insulator protrudes beyond the end of the plug's body and whether it contains a suppresser for radio frequency (RF) interference. A projected-core plug provides a long protrusion into the combustion chamber. This results in the plug running hotter and staying cleaner at low speeds while being better cooled by the fresh air/fuel mixture at high speeds. For these reasons, use a projected core plug whenever possible. Also, most electronic ignitions require the use of a resistor type plug to minimize RF interference. One exception is with the Dyna ignition, where some racers have experienced better performance with non-resistor plugs. If you're not sure where to start with plug heat range or type, consult with your engine builder or call one of the major spark plug companies.

A plug must receive some coloring from the engine's combustion process for it to indicate the engine's state of tune. Today's plugs are harder to read because they have a wider heat range than years ago. Also, unleaded gas colors a plug differently than leaded gas and different brands of race gas vary in plug coloring characteristics. Spark plugs come plated with different alloys, but zinc alloy plugs seem easier to read than other alloys.

To get a proper plug reading at the drag strip, you should adhere to the following procedures: Make a full throttle run through all the gears and cut off the ignition after passing through the timing lights. Pull in the clutch and close the throttle simultaneous to cutting the ignition and coast to a stop. Shutting the engine off cleanly is crucial for an accurate plug reading. Use a new set of plugs because a short run cannot erase the color markings of the previous run. Also, load the engine as much as possible.

To read a plug correctly you need a spark plug flash light and magnifier. However, even without one you can still pick up clues about the cylinder's air/fuel mixture and temperature. The major things to check for are: temperature, signs of detonation or pre-ignition, rich or lean conditions, ignition advance and presence of oil. The parts of the plug that are inspected include the center and ground electrodes, upper and lower part of the ceramic insulator and the threads. The color and condition of these areas will tell a story.

Normal Condition

A plug from a correctly tuned engine will result in the nose (tip) of the ceramic insulator being natural white or having a slight trace of coloring along with a light coloration of the center and ground electrodes. The center electrode should have sharp straight edges.

Overheated

Check the center electrode for early signs of overheating and pre-ignition. The electrode should not show any signs of erosion, such as rounded corners or have a pink or purple color. A slightly overheated plug will have the entire insulator bone white in color while the center electrode has no deposits, but shows signs of turning pink or purple. Also, the ground wire typically starts turning green. A severely overheated plug will include melted electrodes and glazed or blistered insulator.

Pre-ignition
Early signs of pre-ignition may look similar to an overheated plug, but advanced symptoms include extreme melting of the electrodes and blistering or possible destruction of the ceramic insulator.

Detonation
Initial signs of detonation include black or purple specks or shinny black balls on the insulator nose. Severe signs are identified by a vertically fractured ceramic insulator along with rounded electrodes.

Insulator Color
The plug's ceramic insulator can indicate a rich or lean condition. The tip of the insulator and most of it down to *almost* where it joins the plug's outer shell should be natural white or very slightly gray in color. When looking down into the plug, there should be only a hint of a light tan fuel ring at the insulator's base where it connects to the plug shell. If the insulator is entirely white (no ring), the air/fuel mixture is too lean. If the ring is dark brown or the insulator is completely colored, the mixture is too rich. The closer the fuel ring is located to the bottom of the insulator, the leaner the mixture is. Conversely, the closer it is to the insulator's nose, the richer the mixture is.

Fouled
Worn rings and cylinder bores, excessive valve guide clearances, worn valve stem seals and poor crankcase scavenging can cause plug oil fouling. The presence of oil in the chamber adds color to the plug and makes it difficult to read accurately. Also, even a slight amount of oil in the combustion chamber can significantly reduce fuel octane and bring on detonation.

Oil fouling shows up as shiny, black residue and in extreme situations it can be tar-like. Oil buildup on only one side of the plug usually indicates a valve guide seeping oil instead of the rings.

Excessively rich mixtures, too cold a plug heat range or a weak ignition can cause carbon fouling. Soft sooty debris on the plug shell, ceramic insulator or electrode indicates carbon fouling.

Thread Heat Rings
The spark plug threads can indicate the amount of heat saturation the plug is experiencing. The plating on the threads will turn to ash and discolor based on the amount of cylinder heat absorbed. The greater the number of threads blackened, the hotter the cylinder is running.

Three to five threads darkened on a long reach plug indicates a normal amount of cylinder heat for a Big Twin engine. Comparing the number of thread heat rings between each cylinder will indicate how similar the cylinders are generating heat. Remember, enriching the air/fuel mixture or retarding the ignition timing reduces heat. Also, an increase in humidity decreases the number of thread heat rings showing on a plug.

Ignition Timing
The plug's negative ground electrode can indicate, on a relative basis, the amount of ignition advance that is being run. New plugs have a nickel or zinc plating that will discolor at different points on the ground electrode based on the amount of ignition advance. The location where the dark and ashy gray portions of the electrode meet is referred to as the "ignition line" and indicates a relative amount of ignition advance.

As a general rule, the "ignition line" should be positioned approximately at the 90 degree bend on the electrode. An "ignition line" closer to the plug's shell (where the electrode is welded to the shell) indicates a greater amount of ignition advance. On the other hand, when the "ignition line" is closer to the open end of the electrode, there is less advance.

Excessive amounts of ignition advance frequently cause the ground electrode to deteriorate before the center electrode.

SPARK PLUG INSTALLATION
Correct installation is critical for proper operation and normal plug life. The plug reach (length of threads) must be matched to the cylinder

head. Plugs with a long 3/4-inch reach or short 3/8-inch reach are common. Make sure the plug does not extend too far into the combustion chamber because this will allow exposed threads. Exposed threads will easily overheat and cause pre-ignition. If necessary, an extra thick gasket can be used.

Make sure the plug clears the piston dome, especially if high dome pistons are used. If necessary, the piston can be flame slotted to provide more clearance for the plug and better flame propagation.

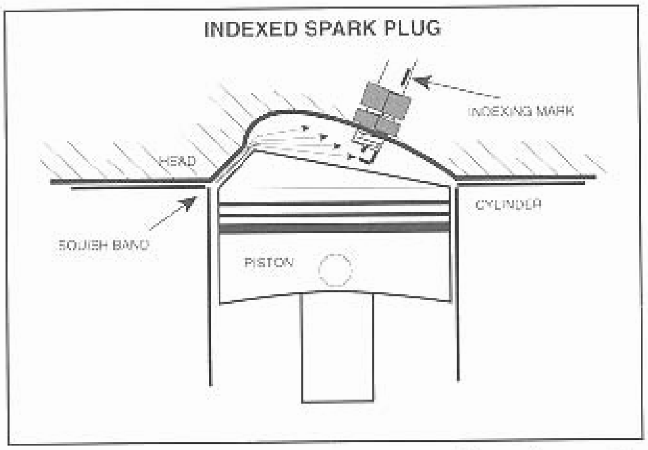
Plugs must be tightened correctly for normal heat transfer between the plug and cylinder head. Under-tightening will reduce heat transfer and may allow combustion heat to destroy the threads while over tightening disturbs the sealing barriers and interferes with proper heat transfer.

Use a torque wrench and tighten to the recommended lb. ft. If you don't have a torque wrench, tighten plugs that use a gasket 1/4-turn beyond finger tight. Also, use a very small amount of anti-seize and keep it only on the threads.

SPARK PLUG INDEXING

Indexing is the process of externally marking the plug insulator with a black felt pen at the point where the ground electrode attaches to the plug shell. Then, when the plug is torqued down in the head, the position of the ground electrode can be noted. This is frequently done for either of two reasons. First, some engines may have insufficient clearance between the plug's ground electrode and the piston dome. In this case, the plug should be

Indexing means installing the plug with the ground electrode positioned at a specific location. Indexing may be required for piston clearance and sometimes can improve combustion. The ground electrode's position is marked with a felt tipped pen before installation. Different thicknesses of indexing washers are available from various ignition system suppliers.



Air/fuel mixture is being rapidly squeezed from the squish band area toward spark plug by the angle top piston. The indexing mark indicates that the air/fuel mixture is not being shrouded from the spark by the back side of the electrode.

positioned so the ground electrode is near the top of the chamber. Second, for non-supercharged engines it is sometimes advantageous to position the ground electrode so it does not shroud the spark from the fuel squirting from the squish area.

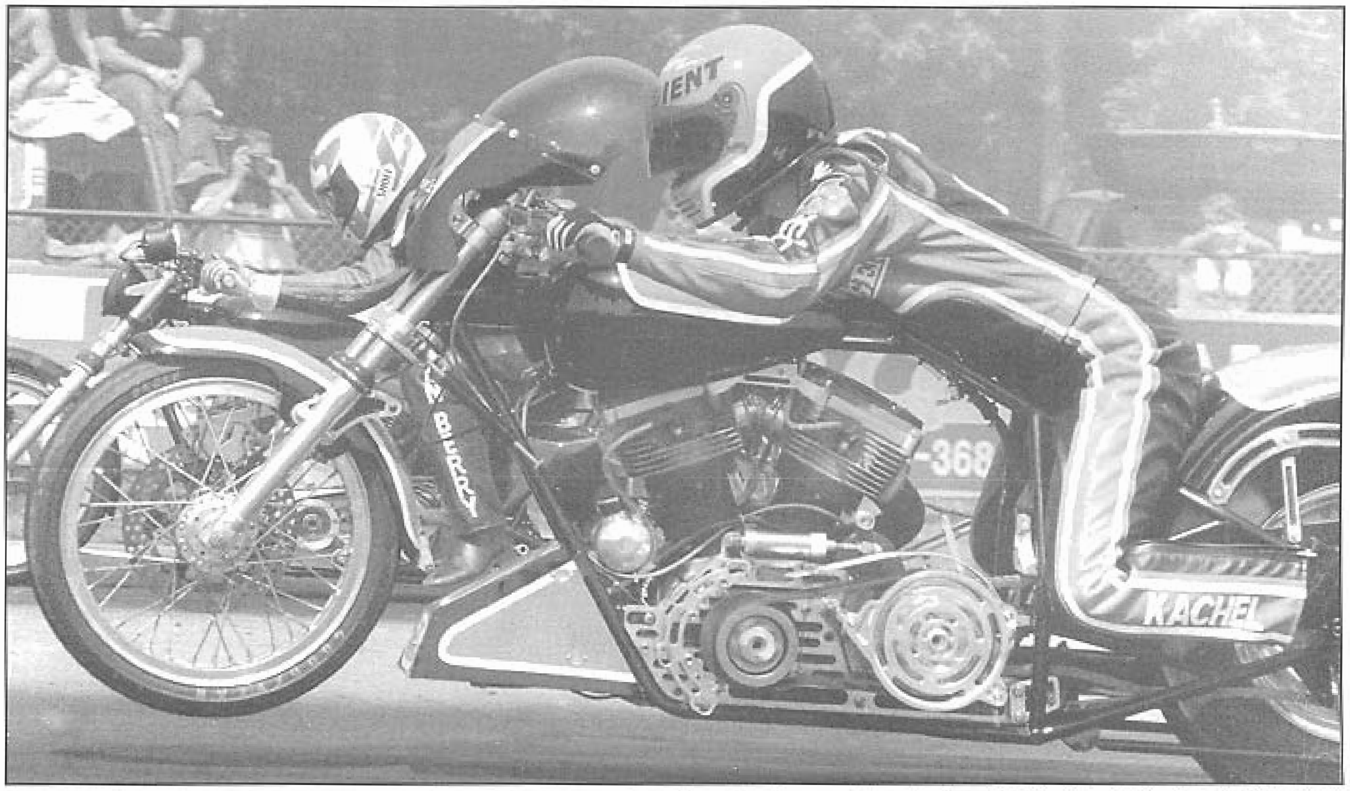
By marking the plug and experimenting with different thicknesses of indexing washers, the final position of the ground electrode can be adjusted.

SPARK PLUG ELECTRODE GAPS

The electrode gap size can vary widely based on the ignition system. In general, it is best to run as large of gap as possible as long as it is consistent with the energy capabilities of the ignition system. But remember, the wider the gap is, the higher the required voltage to jump it will be.

Increased voltage requirements stress the ignition system and the spark plug wires. Spark plug wire "crossfire" can easily develop when using wide gaps and low quality wires. Consider using silicone insulated spiral-core cable for trouble free operation. All things being equal, a wider gap can provide a superior spark for more complete combustion, but there are limits.

Magnetos require a small plug gap. For example, plugs should be gapped at .020-inch when used with the Fairbanks-Morse magneto. For breaker point ignitions, try a .028 to .030-inch plug gap. Gaps for Inductive-Discharge electronic ignitions can vary widely between different manufacturers, but .030 to .032-inch is a good place to start. More than likely, the optimum plug gap for an electronic ignition will



Preparation and total concentration help Glenn Kachel run deep into the 8 second bracket on his Pro Gas 114 ci Evolution. Photo courtesy of Larry Smith/Handcrafted American Racing Motorcycles.

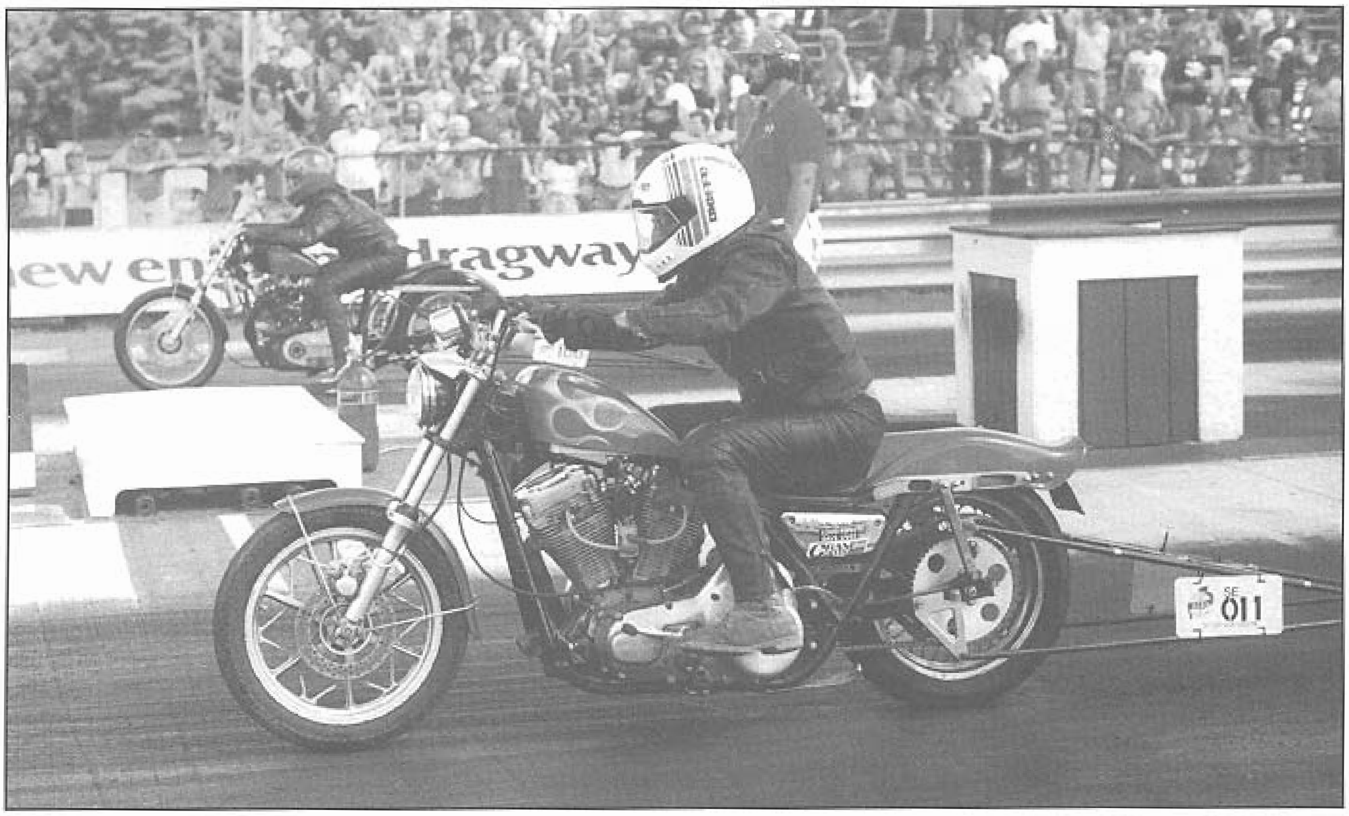
range between .030 and .040-inch. Keep in mind that the stock electronic ignition is setup with a large .038 to .043-inch gap due to emissions testing at low speed operation. Try .035 to .040-inch with the stock electronic system and less for a high compression engine. Capacitive-discharge systems, except CD magnetos, generally use .040-inch and larger gaps. As a general rule, tighten the gap when the compression goes up. For dual plug heads, the plug's gap size may need to be reduced about .005-inch from the listed sizes.

If you intend to experiment with various gap sizes, change the size in .005-inch increments to determine whether there is any performance difference. In some situations, it will be difficult to detect a change. With slightly modified engines, performance differences attributed to gap changes may be very small or even nonexistent. However, with highly modified, high compression engines, a change in plug gap size can make a notable difference.

When gapping the plug, make sure the ground electrode is perfectly parallel to the center electrode. Some racers like using side-gapped plugs because the ground electrode shrouds less of the spark from the combustion chamber. You can modify a plug for side-gapping by shortening the ground electrode with an abrasive wheel or wire cutters so that it only extends to the 90 degree edge of the center electrode. Keep in mind that worn plugs with rounded electrodes can increase the plug voltage requirement by 25 percent.

SPARK PLUG TUNING TIPS

Enrich the air/fuel mixture if the plug's insulation shows signs of glazing or there is no brown fuel ring at the bottom of the insulator. The shape and color of the ceramic insulator should remain as new and there should be no glazing or fusing. The center electrode should not be turning blue and the ground electrode should have little or no signs of turning green. The electrodes should look like new with sharp corners, no erosion and no build up of debris on the center electrode. Signs of aluminum pieces on the plug indicate the piston dome is overheating. These conditions indicate too much heat and require enriching the mixture or retarding the ignition timing.



Many racing classes, such as ECRA's Sreet Eliminator, allow you to enjoy racing without building an all out dragster. Remove the wheelie bars, quiet the exhaust, maybe add shocks and you have a streetable bike. However, preparing for race day, tuning the engine for power and the chassie for traction are just as important in a street class as they are in a high dollar dragster class. Photo courtesy of Larry Smith/Hand-crafted American Racing Motorcycles.

Get the air/fuel mixture and the ignition timing correct before changing spark plug heat range. When the fuel mixture is close to optimum, concentrate on reading the fuel ring at the bottom of the insulator. If you don't have a tuning baseline for the engine, you should start with a rich mixture, retarded ignition and cold plugs. If you have a relatively close baseline, enrich the mixture until the engine slows down and then go down one jet size. Gasoline has a narrow power range of about five percent for maximum power. However, power drops off much quicker going lean than it does going rich.

For ignition advance on an engine you're unfamiliar with, reduce it in increments of two degrees until the engine slows then add two degrees back in it. The opposite method is to increase the advance until the engine slows, shows signs of too much heat or detonation becomes present — but you need to be careful. Check the spark plug's ground electrode for the location of the "ignition line."

At this point, you can concentrate on selecting the correct spark plug heat range. It is usually best to run a plug as hot as possible without any signs of excessive heat. Heat is the major problem. Changing heat range won't make the engine go any faster, but it can help keep the engine together and make it easier to tune. Remember that a plug too hot can bring on pre-ignition and detonation, while one too cold may cause fouling or intermittent missing under excessively rich or lean conditions. You may need to run a different plug heat range in each cylinder. Sometimes the front cylinder needs a colder plug due to the uneven firing pulses of the V-Twin engine. Remember that ambient atmospheric conditions also play a part in tuning.

Up to a point, the larger the plug gap the better. But keep in mind that a large gap requires a strong ignition system and can stress the system and the plug wires. ❖

Appendix A

Camshaft Specifications

Details

This appendix lists camshafts by engine displacement. The tables should be used only as a general reference because, depending on the application, many cams are appropriate for other engine displacements.

EVOLUTION CAMS STOCK ENGINES											
Year	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
84-95	Stock-C	-15/03 03/-15	168 168		.423 .423	-30	99 99	99		Hyd	Stock
84-87	Stock-V	-06/38 25/-03	212 202		.472 .472	-9	112 104	108		Hyd	Stock
88-91	Stock-L	01/37 52/02	218 234	266 280	.495 .495	3	108 115	111.5	.091 .083	Hyd	Stock
92-96	Stock-N	-02/30 31/-09	208 202	250 242	.472 .472	-11	106 110	108	.070 .049	Hyd	Stock
95-96	Stock-Q	0/22 48/-10	202 218		.472 .472	-10	99 116	107.5		Hyd	Stock

Table A.1 Valve lift with 1.63:1 rocker arm ratio.

EVOLUTION CAMS BOLT-IN*											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	EV13	15/31 45/13	226 238	270 280	.485 .495	28	98 106	102	.161 .148	Hyd	Stock
Andrews	EV27	20/36 44/16	236 240	270 274	.495 .495	36	98 104	101	.182 .166	Hyd	Stock
Andrews	EV3	21/37 43/15	238 238	280 280	.495 .495	36	98 104	101	.197 .159	Hyd	Stock
Andrews	EV46	25/41 49/17	246 246	283 283	.495 .495	42	98 106.5	102	.207 .197	Hyd	Stock
Bartels'	BP20	18/36 45/13	234 238		.490 .495	31	98 106	102		Hyd	Stock
Bartels'	BP40	21/37 48/20	238 248		.495 .495	41	98 104	101		Hyd	Stock
Carl's Speed Shop	CM495F	19/47 50/16	246 246		.495 .495	35	104 107	105.5		Hyd	Stock
Crane	310	16/40 43/19	236 242		.490 .490	35	102 102	102	.164 .185	Hyd	Stock
Crane	H286	19/43 48/24	242 252		.490 .490	43	102 102	102	.179 .206	Hyd	Stock
Edelbrock	1740	19/43 48/24	242 252		.490 .490	43	102 102	102	.177 .204	Hyd	Stock
Head Quarters	HQ-24	20/36 52/19	236 251		.500 .500	39	98 106.5	102.2	.174 .166	Hyd	Stock
Leineweber	E-2	43/62 67.5/37		285 285	.470 .470	80 at .020"	99.5 105.2	102.4	.208 .182	Hyd/Sld	Stock
Leineweber	E3S	44/62 70/36		286 286	.502 .502	80 at .020"	99 107	103	.215 .170	Hyd/Sld	Stock
Powerhouse	500P	16/42 44/16	238 240		.500 .500	32	103 104	103.5	.174 .179	Hyd	Stock
RevTech	05	16/32 41/17	228 238		.465 .475	33	98 102	100		Hyd	Stock
RevTech	10	21/37 50/18	238 248		.475 .480	39	98 106	102		Hyd	Stock
Rivera Engineering	EV-100	27/43 59/24	250 263		.467 .467	51	98 107.5	102.7		Hyd	Stock
S&S	502	28/40 50/24	248 254		.500 .500	52	96 103	99.5	.225 .221	Hyd/Sld	Stock
Screamin' Eagle	406	16/48 51/19	244 250	282 288	.480 .480	35	106 106	106		Hyd	Stock
Sifton	143-EV	20/35 46/14	235 240		.500 .500	34	97 106	101.5	.184 .160	Hyd	Stock
Sifton	145-EV	28/42 50/20	250 250		.460 .460	48	97 105	101	.200 .176	Hyd/Sld	Stock
Sifton	140-EV	30/42 55/27	252 262		.450 .450	57	96 104	100	.222 .178	Hyd	Stock
V-Thunder	3010	15/39 39/15	234 234		.500 .500	30	102 102	102	.156 .156	Hyd	Stock
V-Thunder	3020	18/42 42/18	240 240		.500 .500	36	102 102	102	.173 .173	Hyd	Stock

Table A.2 *Pistons must have valve reliefs. Valve lift with 1.63:1 rocker arm ratio.

CAMSHAFT SELECTION

An engine's personality is directly influenced by the profile of its camshaft. For this reason, proper selection of a cam and its related valvetrain components is crucial if an engine is to perform up to expectations. The following are some factors you should consider when selecting a cam. But take note that there is no one cookbook formula for selecting a cam because too many variables are involved. For additional information, refer to the chapter Cam and Valvetrain.

APPLICATION AND OBJECTIVES

Too radical of cam can limit a race engine's power or kill low end power on a street engine. As a result, the cam you select should correspond to the engine's power range most important to you. When selecting a cam, start out by determining

EVOLUTION CAMS MODIFIED 80 CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	EV35	21/37 52/20	238 252	280 295	.495 .500	41	98 106	102	.197 .182	Hyd	Yes
Andrews	EV51	28/44 54/22	252 256	286 290	.510 .510	50	98 106	102	.233 .195	Hyd	Stock
Andrews	EV59	28/48 56/24	256 260	290 294	.560 .560	52	100 106	103	.236 .208	Hyd	Yes
Bartels'	BP60	26/42 55/23	248 258		.495 .495	49	98 106	102		Hyd	Stock
Carl's Speed Shop	CM5	19/47 50/16	246 246		.580 .580	35	104 107	105.5	.185 .176	Hyd	Yes
Crane	H290	17/43 45/23	240 248		.581 .581	40	103 101	102	.169 .195	Hyd	Yes
Crane	H296	24/48 57/25	252 262		.490 .500	49	102 106	104	.200 .206	Hyd	Stock
Edelbrock	1741	20/46 52/22	246 254		.600 .600	42	103 105	104	.187 .180	Hyd	Yes
Head Quarters	HQ-25	18/38 42/14	236 236		.550 .550	32	100 104	102	.166 .150	Hyd	Yes
Head Quarters	HQ-29	20/36 51/19	236 250		.580 .500	39	98 106	102	.193 .162	Hyd	Yes
Head Quarters	HQ-23	19/47 52/24	246 256		.600 .530	43	104 104	104	.172 .191	Hyd	Yes
Leineweber	E6S	39/56 69/26		275 275	.544 .544	65 at .020"	98.5 111.5	105	.225 .132	Hyd/Sld	Yes
Powerhouse	530G	18/44 50/16	242 246		.530 .530	34	103 107	105	.184 .184	Hyd/Sld	Yes
Powerhouse	560V	18/50 54/18	248 252		.560 .560	36	106 108	107	.180 .180	Hyd/Sld	Yes
Red Shift	535	12/42 57/4	234 241		.535 .560	16	105 116.5	110.7	.138 .098	Hyd	Yes
Red Shift	560	18/44 58/4	242 242		.560 .560	22	103 117	110	.180 .100	Hyd	Yes
Red Shift	575	25/54 63/18	259 261		.575 .575	43	104.5 112.5	108.5	.225 .162	Hyd	Yes
RevTech	20	22/46 59/19	248 258		.480 .490	41	102 110	106		Hyd	Stock
RevTech	30	21/37 52/20	238 252		.495 .530	41	98 106	102		Hyd	Yes
S&S	561	32/40 50/26	252 256		.560 .560	58	94 102	98	.252 .210	Hyd/Sld	Yes
Screamin' Eagle	400	26.5/50.5 55.5/31.5	257 267	304 314	.500 .500	58	102 102	102		Hyd	Yes
Screamin' Eagle	433	23/47 56/24	250 260	286 299	.530 .530	47	102 106	104		Solid	Yes
Sifton	144-EV	27/46 56/22	253 258		.490 .490	49	99.5 107	103	.216 .166	Hyd	Yes
Sifton	141-EV	29/41 58.5/26	250 264		.480 .480	55	96 106	101	.232 .184	Hyd/Sld	Yes
V-Thunder	3030	16/44 44/16	240 240		.530 .530	32	104 104	104	.161 .161	Hyd	Yes
V-Thunder	3040	17/45 50/22	242 252		.510 .510	39	104 104	104	.164 .191	Hyd	Yes
V-Thunder	3050	22/50 50/22	252 252		.510 .510	44	104 104	104	.191 .191	Hyd	Yes

Table A.3 Note that bolt-in cams also can be considered for this application level. Valve lift with 1.63:1 rocker arm ratio.

your objectives for the engine. Ask yourself whether you want a race only engine, a combination street and race engine or just a mild performance increase.

Next determine where you want the engine to develop its power. For example, is power more important to you between 2000 and 4000 rpm or 4000 and 6500+ rpm? Street engines spend most of their time in the 2000 to 4000 rpm range, while race engines usually stay above 4500 or

EVOLUTION CAMS 89-93 CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	EV59	28/48 56/24	256 260	290 294	.560 .560	52	100 106	103	.236 .208	Hyd	Yes
Andrews	EV81	32/60 66/30	272 276	306 310	.610 .610	62	104 108	106	.262 .244	Hyd	Yes
Carl's Speed Shop	CM5	19/47 50/16	246 246		.580 .580	35	104 107	105.5	.185 .176	Hyd	Yes
Carl's Speed Shop	CM6	36/60 68/28	276 276		.612 .612	64	102 110	106	.244 .228	Hyd	Yes
Crane	H304	24/50 55/25	254 260		.600 .600	49	103 105	104	.208 .206	Hyd	Yes
Crane	H306	28/54 69/23	262 272		.500 .510	51	103 113	108	.224 .195	Hyd	Yes
Head Quarters	HQ-23	19/47 52/24	246 256		.600 .530	43	104 104	104	.172 .191	Hyd	Yes
Head Quarters	HQ-26	20/54 53/22	254 255		.600 .530	42	107 105.5	106.2	.182 .181	Hyd	Yes
Leineweber	E-5S	39/56 69/26		275 275	.544 .544	65 at .020"	98.5 111.5	105	.225 .132	Hyd/Sld	Yes
Leineweber	E-51	39/60 66/33		279 279	.540 .540	72 at .020"	100.5 106.5	103.5	.225 .168	Hyd/Sld	Yes
Powerhouse	560V	18/50 54/18	248 252		.560 .560	36	106 108	107	.180 .180	Hyd/Sld	Yes
Powerhouse	565VP	24/54 58/20	258 258		.565 .565	44	105 109	107	.200 .200	Hyd/Sld	Yes
Red Shift	575	25/54 63/18	259 261		.575 .575	43	104.5 112.5	108.5	.225 .162	Hyd	Yes
Red Shift	625	33/58 64/28	271 272		.625 .625	61	102.5 108	105.2	.250 .216	Hyd	Yes
RevTech	40	31/55 59/27	266 266		.560 .560	58	102 106	104		Hyd	Yes
S&S	561	32/40 50/26	252 256		.560 .560	58	94 102	98	.252 .210	Hyd/Sld	Yes
Screamin' Eagle	400	26.5/50.5 55.5/31.5	257 267	304 314	.500 .500	58	102 102	102		Hyd	Yes
Sifton	141-EV	29/41 58.5/26	250 264		.480 .480	55	96 106.2	101	.232 .184	Hyd/Sld	Yes
Sifton	146-EV	30/45 56/24	255 260		.500 .500	54	97 106	101.5	.232 .200	Solid	Yes
V-Thunder	3060	24/56 61/29	260 270		.585 .585	53	106 106	106	.210 .226	Hyd	Yes

Table A.4 Valve lift with 1.63:1 rocker arm ratio.

5000 rpm. Within limits, the higher up in the rpm band you want power, the more duration you will need for a given engine displacement.

Decide if you have to run at the front of the pack or just along with the pack? If you're racing in a specific class, determine how competitive the class is. Check if your class runs a "dial in" E.T. or "heads up"? Depending on your engine combination, running at the front of the pack may require a cam with up to 30 degrees more duration and .100-inch more lift than one that gives a mild performance increase. But increased duration normally requires a tradeoff of low rpm torque for high rpm horsepower. For example, if you install a long duration cam that is designed for a large displacement engine in a small displacement engine, you will need to rev the smaller engine between 500 and 1500 rpm higher than normal to realize the potential power benefits of the increased duration. And to minimize the inherent loss of cylinder pressure (torque) at low rpm, the mechanical compression ratio of the smaller engine will need to be increased somewhere between one and four ratios. Regardless, if you want to win in a highly competitive class or any class where racing is conducted "heads up," you will generally need a cam that performs best in the higher rpm ranges.

The total weight of the bike, rider and passenger is another factor to consider. Heavy bikes

EVOLUTION CAMS 93-98 CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	EV72	30/54 60/28	264 268	298 302	.560 .560	58	102 106	104	.246 .230	Hyd	Yes
Andrews	EV79	31/55 64/32	266 276	307 314	.560 .550	63	102 106	104	.245 .250	Hyd	Yes
Andrews	EV81	32/60 66/30	272 276	306 310	.610 .610	62	104 108	106	.262 .244	Hyd	Yes
Andrews	EV84	32/64 70/30	276 280	310 314	.640 .640	62	106 110	108	.269 .246	Hyd	Yes
Carl's Speed Shop	CM6	36/60 68/28	276 276		.612 .612	64	102 110	106	.244 .228	Hyd	Yes
Crane	H304	24/50 55/25	254 260		.600 .600	49	103 105	104	.208 .206	Hyd	Yes
Crane	H310	23/63 68/28	266 276		.550 .550	51	110 110	110	.229 .229	Hyd	yes
Head Quarters	HQ-26	20/54 53/22	254 255		.600 .530	42	107 105.5	106.2	.182 .181	Hyd	Yes
Head Quarters	HQ-28	23/63 52/24	266 256		.600 .530	47	110 104	107	.199 .186	Hyd	Yes
Leineweber	E-5	45.5/70 74/41		295 295	.540 .540	86.5 at .020"	102.5 106.5	104.5	.272 .215	Hyd/Sld	Yes
Leineweber	E-4	49/74 76/39		303 295	.560 .545	88 at .020"	102.5 108.5	105.5	.297 .196	Hyd/Sld	Yes
Powerhouse	595E	24/60 64/20	264 264		.595 .595	44	108 112	110	.215 .191	Hyd/Sld	Yes
Powerhouse	620E	32/64 72/24	276 276		.620 .620	56	106 114	110	.262 .215	Solid	Yes
Red Shift	653	21/52 58/16	253 254		.653 .653	37	105.5 111	108.2	.180 .162	Hyd	Yes
Red Shift	654	21/59 57/17	260 254		.654 .654	38	109 110	109.5	.192 .162	Hyd	Yes
Red Shift	625	33/58 64/28	271 272		.625 .625	61	102.5 108	105.2	.250 .216	Hyd	Yes
RevTech	40	31/55 59/27	266 266		.560 .560	58	102 106	104		Hyd	Yes
S&S	562	34/55 60/29	269 269		.560 .560	63	100.5 105.5	103	.260 .220	Hyd/Sld	Yes
Sifton	142-EV	32/52 60/30	264 270		.540 .540	62	100 105	102.5	.235 .226	Solid	Yes
V-Thunder	3060	24/56 61/29	260 270		.585 .585	53	106 106	106	.210 .226	Hyd	Yes

Table A.5 Valve lift with 1.63:1 rocker arm ratio.

perform better at low rpm with a cam that has less duration and overlap. Lighter bikes can handle more duration and overlap while still retaining good low speed performance.

ENGINE MODIFICATIONS

Decide if you want a bolt-in cam or whether you're willing to spend the necessary time and expense to remove the cylinder heads and install stiffer springs with longer travel. Remember that most high lift, long duration cams are designed to work in conjunction with other high-performance components such as: high-flowing carburetor, free-flowing exhaust system, cylinder head porting, increased compression, stronger valvetrain components, increased engine size or a combination of these modifications.

A stiff valvetrain is imperative for achieving maximum performance from any cam. The valvetrain must remain rigid if it is to follow the cam's lobe profile accurately. Keep in mind that valvetrain rigidity is more important than light weight, although unnecessary weight is detrimental. Stiffer valve springs with longer travel are generally needed for any non-bolt-in cam. And the more radical the cam or the higher the engine's rpm, the higher the valve spring pressure generally needs to be. Also note that most bolt-in cams can benefit from stronger, high-performance springs. Valve springs must

EVOLUTION CAMS 103+ CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	EV9	36/60 64/32	276 276		.550 .550	68	102 106	104		Hyd/Sld	Yes
Andrews	EV88	34/70 76/32	284 288	318 322	.680 .680	66	108 112	110	.288 .264	Hyd	Yes
Carl's Speed Shop	CM6	36/60 68/28	276 276		.612 .612	64	102 110	106	.244 .228	Hyd	Yes
Carl's Speed Shop	CM780F	34/70 74/30	284 284		.774 .774	64	108 112	110	.291 .267	Solid	Yes
Crane	H314	26/54 65/21	260 266		.600 .600	47	104 112	108	.219 .184	Hyd	Yes
Crane	HEV0042	27/59 71/23	266 274		.600 .576	50	106 114	110	.226 .194	Hyd	Yes
Crane	HEV0034	37/59 61/35	276 276		.550 .550	72	101 103	102	.281 .263	Hyd	Yes
Crane	EVR0005	38/62 75/35	280 290	324 332	.731 .746	73	102 110	106	.277 .238	Solid	Yes
Head Quarters	HQ-28	23/63 52/24	266 256		.600 .530	47	110 104	107	.199 .186	Hyd	Yes
Head Quarters	HQ-27	26/60 70/24	266 274		.650 .575	50	107 113	110	.221 .198	Hyd	Yes
Leineweber	E-6	47/76 84/39		303 303	.560 .560	86 at .020"	104.5 112.5	108.5	.285 .196	Solid	Yes
Leineweber	E-7	43.5/73.5 77/40		297 297	.580 .580	83.5 at .020"	105 108.5	106.8	.275 .227	Solid	Yes
Powerhouse	620E	32/64 72/24	276 276		.620 .620	56	106 114	110	.262 .215	Solid	Yes
Powerhouse	620M	36/66 76/30	282 286		.620 .620	66	105 113	109	.200 .200	Solid	Yes
Red Shift	655	21/65 70/16	266 266		.655 .655	37	112 117	114.5	.197 .162	Hyd	Yes
Red Shift	625	33/58 64/28	271 272		.625 .625	61	102.5 108	105.2	.250 .216	Hyd	Yes
Red Shift	710	28/58 65/21	266 266		.705 .705	49	105 112	108.5	.232 .190	Solid	Yes
Red Shift	785	38/61 72/27	279 279		.785 .785	65	101.5 112.5	107	.315 .243	Solid	Yes
S&S	562	34/55 60/29	269 269		.560 .560	63	100.5 105.5	103	.260 .220	Hyd/Sld	Yes
S&S	563	32/64 64/32	276 276		.560 .560	64	106 106	106	.250 .220	Hyd/Sld	Yes
S&S	631	34/61 66/29	275 276		.630 .630	63	103.5 109	106	.281 .221	Solid	Yes
Sifton	142-EV	32/52 60/30	264 270		.540 .540	62	100 105	102.5	.235 .226	Solid	Yes
Sifton	147-EV	24/64 66/28	268 275		.640 .640	54	110 109	109.5	.208 .240	Solid	Yes
V-Thunder	3070	29/61 63/31	270 274		.608 .608	60	106 106	106	.234 .242	Hyd	Yes

Table A.6 Valve lift with 1.63:1 rocker arm ratio.

be matched to the cam profile and the engine's rpm range. Too little spring pressure is generally worse than too much, but excessively high pressure causes unnecessary power robbing friction, heat and wear. Consult with your cam manufacturer for recommendations.

Stock Evolution hydraulic lifters usually can be safely revved to about 6300 rpm. A low cost S&S Limited lifter kit makes an Evolution hydraulic lifter operate similarly to a solid unit and is highly recommended. Velva-Touch hydraulic lifters can be revved to about 8000 rpm and are highly recommended for serious racing or when lifter assemblies need replacement. Jim's Machine offers performance hydraulic and solid lifters and lifter blocks designed for high rpm operation. Shovelhead hydraulic lifters should be replaced with either solid, Velva-Touch or

SHOVELHEAD CAMS 74-80 CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Stock	H Front Cyl	-6/46 44/20	220 244	256 282	.390 .390	12	116 102	109		Hyd	Stock
	Rear Cyl	14/38 44/20	232 244	274 282	.390 .390	34	102 102	102			
Andrews	A	21/43 43/21	244 244	296 296	.450 .450	42	101 101	101	.156 .156	Solid	Stock
Andrews	B	26/50 50/26	256 256	298 298	.485 .485	52	102 102	102	.182 .182	Solid	Yes
Crane	288B	22/42 42/22	244 244		.450 .450	44	100 100	100	.179 .183	Hyd	Stock
Crane	308B	26/50 50/26	256 256		.490 .490	52	.182 .182	102	.184 .184	Solid	Yes
Head Quarters	HQ 20/21/22	20/54 43/21	254 244		.465 .440	41	107 101	104	.161 .160	Hyd	Yes
Leineweber	L-1	38/65 69/34		283 283	.420 .420	72 at .020"	103.5 107.5	105.5	.174 .168	Hyd/Sld	Stock
Leineweber	L-2S	42/64 73/33		286 286	.450 .450	75 at .020"	101 110	105.5	.200 .160	Hyd/Sld	Stock
Red Shift	510	25/54 63/18	259 261		.510 .510	42	104.5 112.5	108.5	.195 .142	Hyd	Yes
Rivera Engineering	RERS-10	24/49 55/25	253 260		.440 .505	49	102.5 105	103.7		Solid	Yes
S&S	514	23/43 43/23	246 246		.514 .514		100 100	100	.179 .179	Solid	Yes
Sifton	107	29/54 53/25	263 258		.440 .440	54	102.5 104	103	.193 .164	Hyd	Yes
Sifton	112	28/45 53/25	253 258		.440 .440	53	98.5 104	101.5	.200 .179	Hyd/Sld	Yes
V-Thunder	4000/1	14/42 42/14	236 236		.450 .450	28	104 104	104	.133 .133	Hyd	Stock
V-Thunder	4010/11	14/42 42/14	236 236		.485 .485	28	104 104	104	.133 .133	Hyd	Yes
V-Thunder	4020/21	17/45 45/17	242 242		.485 .485	34	104 104	104	.147 .147	Hyd	Yes
V-Thunder	4040/41	17/45 50/22	242 252		.485 .485	39	104 104	104	.147 .171	Hyd	Yes

Table A.7 Valve lift with 1.43:1 rocker arm ratio.

Jim's Machine lifters.

Chrome-moly pushrods are generally heavier than aluminum, but more ridged at high rpm. When used with solid lifters, chrome-moly pushrods are generally noisier than aluminum pushrods, but their extra rigidity is beneficial. Adjustable pushrods are recommended for most Evolution engines so hydraulic lifter pre-load can be set accurately. Maximum effort engines sometimes can benefit from lighter, one piece pushrods.

Accurate rocker arm geometry is crucial for maximum performance. When reworking your heads, be sure to have a competent head porter or engine builder properly setup the rocker arm geometry. Roller rocker arms reduce friction and valvetrain wear especially with a cam that has more than .550-inch lift. They also can improve valve control at high rpm. For accurate valve timing, the camshaft, cam bearings and rocker arm bushings must fit properly. Also, each tappet must fit correctly in its tappet block and the tappet roller must not be loose. Sloppy valvetrain components cause inaccurate valve timing.

Accurate valve timing is also dependent on accurate cam timing. All camshafts are ground to certain valve timing specifications. A Big Twin cam has a combination drive and timing gear pressed onto its shaft. The gear must be correctly positioned for accurate opening and closing valve timing. In many instances, the gear is positioned incorrectly during production. To help ensure accurate valve timing, have a capable motor shop degree the gear to the cam-

SHOVELHEAD CAMS 84-91 CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	6	32/56 56/32	268 268	325 325	.510 .510	64	102 102	102	.190 .190	Solid	Yes
Andrews	M	28/56 56/28	264 264	309 309	.590 .590	56	104 104	104	.210 .210	Solid	Yes
Andrews	S82	32/60 66/30	272 276	306 310	.590 .590	62	104 108	106	.237 .220	Hyd	Yes
Crane	310B	31/55 55/31	266 266		.525 .525	62	102 102	102	.217 .222	Solid	Yes
Crane	304B	32/56 56/32	268 268		.485 .485	64	102 102	102	.217 .217	Solid	Yes
Head Quarters	HQ-20/21/22	20/54 43/21	254 244		.465 .440	41	107 101	104	.161 .160	Hyd	Yes
Leineweber	L-3	42/68 74/36		290 290	.485 .485	78 at .020"	103 109	106	.215 .174	Solid	Yes
Leineweber	J-4	44/79 74/38		303 292	.503 .486	82 at .020"	107.5 108	107.7	.228 .182	Solid	Yes
Red Shift	510	25/54 63/18	259 261		.510 .510	42	104.5 112.5	108.5	.185 .142	Hyd	Yes
Red Shift	580	23/51 59/15	254 254		.580 .580	38	104 112	108	.190 .135	Hyd/Sld	Yes
Sifton	107	29/54 53/25	263 258		.440 .440	54	102.5 104	103	.193 .164	Hyd	Yes
Sifton	105	25/60 63/23	265 266		.475 .475	48	107.5 110	109	.180 .175	Solid	Yes
V-Thunder	4030/31	22/50 50/22	252 252		.485 .485	44	104 104	104	.171 .171	Hyd	Yes

Table A.8 Valve lift with 1.43:1 rocker arm ratio.

shaft before it is installed in the engine. But note that this procedure does not replace degreeing the camshaft to the engine after the cam is installed. For maximum performance, degreeing the cam to the engine also must be done.

COMPRESSION RATIO

There is a definite relationship between a given cam profile and the minimum compression ratio required for acceptable low rpm performance. A higher compression ratio can help regain low end power lost due to increased valve duration. However, if you're running 92 octane pump gas, too much compression will lead to power robbing detonation that will be particularly detectable on hot days.

As a general guideline, Evolution single-plug motors can run about 9.3:1 to 9.8:1 compression without encountering detonation. With dual plug heads you can get between 10:1 and 10.8:1 compression. Yet in certain cases, you sometimes can run still higher compression on pump gas, but it depends on the engine combination and atmospheric conditions along with the combustion chamber's design and setup. The maximum compression for a Shovelhead engine running 92 octane pump gas usually is about 10:1 with dual plugs and 9:1 to 9.5:1 with single plugs.

INTAKE TIMING

The intake valve's closing point is a key timing specification. A later closing intake valve improves high rpm performance, but also reduces low end power because the engine's operational compression at low speeds is reduced. Conversely, an early closing intake valve improves low rpm performance, but also reduces high rpm power since air flow is decreased at high rpm.

EXHAUST TIMING

A later opening exhaust valve improves low and midrange performance because cylinder pressure is retained longer. Conversely, an early opening exhaust helps high rpm performance because cylinder scavenging is improved. Heavy bikes and low rpm engines do better with a later opening exhaust valve, while lighter bikes and high rpm engines can benefit from an earlier

SHOVELHEAD CAMS 93-98 CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	6	32/56 56/32	268 268	325 325	.510 .510	64	102 102	102	.190 .190	Solid	Yes
Andrews	M	28/56 56/28	264 264	309 309	.590 .590	56	104 104	104	.210 .210	Solid	Yes
Andrews	C	37/61 61/37	278 278	318 318	.525 .525	74	102 102	102	.234 .234	Solid	Yes
Andrews	9	32/64 64/32	276 276	320 320	.530 .530	64	106 106	106	.222 .222	Solid	Yes
Andrews	S82	32/60 66/30	272 276	306 310	.590 .590	62	104 108	106	.237 .220	Hyd	Yes
Andrews	S84	32/64 70/30	276 280	280 284	.630 .630	62	106 110	108	.241 .223	Hyd	Yes
Crane	310B	31/55 55/31	266 266		.525 .525	62	102 102	102	.217 .222	Solid	Yes
Crane	304B	32/56 56/32	268 268		.485 .485	64	102 102	102	.217 .217	Solid	Yes
Crane	320B	36/60 64/32	276 276		.550 .550	68	102 106	104	.246 .229	Solid	Yes
Leineweber	J-4	44/79 74/38		303 292	.503 .486	82 at .020"	107.5 108	107.7	.228 .182	Solid	Yes
Leineweber	L-51	44/68 76/36		292 292	.520 .520	80 at .020"	102 110	106	.240 .180	Solid	Yes
Leineweber	L-5	41/71 73/39		292 292	.520 .520	80 at .020"	105 107	106	.225 .210	Solid	Yes
Red Shift	625	26/58 62/22	264 264		.625 .625	48	106 110	108	.198 .170	Hyd/Sld	Yes
Red Shift	550	33/58 64/28	271 272		.550 .550	61	102.5 108	105.2	.220 .200	Hyd	Yes
S&S	495	30/62 62/30	274 274		.495 .495	60	107 107	107	.220 .198	Solid	Yes
Sifton	117	32/58 56/30	270 266		.485 .485	62	103 103	103	.212 .204	Solid	Yes
Sifton	118	35/54 62/32	269 274		.575 .575	67	99.5 105	102	.233 .236	Solid	Yes
Sifton	108	34/65 66/33	279 280		.485 .485	67	105.5 106	106	.214 .240	Solid	Yes
V-Thunder	4050/51	24/56 61/29	260 270		.550 .550	53	106 106	106	.183 .207	Hyd	Yes

Table A.9 Valve lift with 1.43:1 rocker arm ratio.

opening exhaust.

LOBE SEPARATION ANGLE

Two separate cams that have identical duration and lift values, but different lobe separation angles (LSA) will have different performance characteristics. For a given duration, a narrow or smaller LSA generally improves low rpm power, while a wider LSA increases performance at high rpm. Increasing the engine's mechanical compression ratio can help restore lost low rpm power due to a narrow LSA or a large amount of valve overlap. Remember, raising an engine's compression ratio in conjunction with a performance cam installation will help maximize the cam's potential and minimize low rpm power loss.

LIFT

With stock valve springs, valve lift is limited to about .500-inch for an Evolution engine and about .450-inch lift for a Shovelhead. Ideally, a cam's lift should be matched to the cylinder head's airflow characteristics, engine displacement and rpm band. Most reworked heads benefit from higher valve lifts. With the introduction of the Evolution head and the availability of better performance parts, the trend for valve lift is moving higher.

Very high lift cams are easier to install in Evolution motors than Shovelheads. Note that higher lift cams generally require more work to install because valve spring spacing, valve-to-valve interference, cam lobe-to-crankcase clear-

SHOVELHEAD CAMS 103+ CUBIC INCH ENGINES											
Company	Grind	Valve Timing	Duration at .053"	Duration at .020"	Valve Lift	Overlap at .053"	Lobe Center	Lobe Separation Angle	TDC Lift	Lifter Type	Spring Spacing
Andrews	C	37/61 61/37	278 278	318 318	.525 .525	74	102 102	102	.234 .234	Solid	Yes
Andrews	9	32/64 64/32	276 276	320 320	.530 .530	64	106 106	106	.222 .222	Solid	Yes
Andrews	10	34/70 70/34	284 284	344 344	.580 .580	68	108 108	108	.230 .230	Solid	Yes
Andrews	S86	34/70 76/32	284 288	318 322	.660 .660	66	108 112	110	.254 .235	Hyd	Yes
Crane	320B	36/60 64/32	276 276		.550 .550	68	102 106	104	.246 .229	Solid	Yes
Crane	330B	41/65 73/33	286 286		.575 .575	74	102 110	106	.275 .235	Solid	Yes
Leineweber	L-5	41/71 73/39		292 292	.520 .520	80 at .020"	105 107	106	.225 .210	Solid	Yes
Leineweber	L-61	44/83 77/35		307 292	.550 .520	79 at .020"	109.5 111	110.2	.339 .184	Solid	Yes
Leineweber	L-6	48/79 73/39		307 292	.550 .520	87 at .020"	105.5 107	106.2	.262 .210	Solid	Yes
Red Shift	550	33/58 64/28	271 272		.550 .550	61	102.5 108	105.2	.220 .200	Hyd	Yes
Red Shift	625	26/58 62/22	264 264		.625 .625	48	106 110	108	.198 .170	Hyd/Sld	Yes
Red Shift	688	38/61 72/27	279 279		.688 .688	65	101.5 112.5	107	.272 .213	Solid	Yes
Sifton	118	35/54 62/32	269 274		.575 .575	67	99.5 105	102	.233 .236	Solid	Yes
Sifton	108	34/65 66/33	279 280		.485 .485	67	105.5 106	106	.214 .240	Solid	Yes
Sifton	111	40/66 66/39	286 285		.575 .575	79	103 103.5	103	.271 .274	Solid	Yes
V-Thunder	4050/51	24/56 61/29	260 270		.550 .550	53	106 106	106	.183 .207	Hyd	Yes

Table A.10 Valve lift with 1.43:1 rocker arm ratio.

ance and other factors must be considered for proper installation.

TUNING

A camshaft's performance is closely related to an engine's intake tract, exhaust system, compression ratio and other factors. For maximum performance, the engine's exhaust and intake systems can be tuned to the cam. Through trial and error testing, the exhaust system's length, pipe diameter and backpressure can be optimized to a specific cam. Also, the intake tract's length and volume along with the carburetor's size and design can be tailored to the cam. Furthermore, the engine's mechanical compression ratio should be optimized to the cam as well as the ignition timing and spark plug heat range.

In summary, for optimum performance, tractability and reliability, choose a cam and related valvetrain components that match your performance objectives, rpm band and engine combination. Some of the engine components and factors that should be considered in your decision making process include cubic inch displacement, cylinder head airflow characteristics, compression ratio, carburetor size, exhaust system, number of transmission gears, gear ratios and total weight of the bike. ❖

Appendix E

Additions and Updates

Update Corner

This appendix includes new performance products, updates and late breaking information.

ADDITIONS and UPDATES
CHAPTER 4
INDUCTION SYSTEM

MIKUNI HSR42 CARBURETOR

The Mikuni HSR42 is a second generation high-performance flat slide carburetor that was first introduced in early 1994. It has been on the drawing board for the last four years and retains the best inherent features of the older HS40 such as the flat slide and smoothbore design, while incorporating new features for improved performance.

The improvements start with a new body design and 42mm smoothbore venturi that flows 15 to 20 percent more air than its 40mm predecessor. These changes require a physically larger overall body size than the HS40, but one that is about 8mm shorter when measured from front to back. The shorter dimension plays a part in the airflow improvement and should allow it to tuck under large gas tanks easier.

To support its increased airflow potential and eliminate a design shortfall of the HS40, the new HSR42 includes a larger fuel bowl capacity that helps keep the fuel level from dropping below the pilot fuel jet during hard acceleration. This condition typically causes an engine to

The Mikuni HSR42 is a second generation flat slide carburetor that retains the best features of the older HS40 while incorporating new improved features. Its 42mm smoothbore venturi flows more air and its needle and seat has a greater fuel delivery rate than the HS40. Its jetting circuits and accelerator pump are almost identical to the HS40.

A high flowing intake manifold is necessary for achieving an efficient intake tract. This Ram Jett Retainer intake manifold eliminates early model Evo compliance fittings and is made to fit the HSR42 and other carburetors. Be sure to remove all machining and casting marks on its interior and finish with a 60-grit emery wheel. Remember that milled heads may require a narrower manifold while longer cylinders usually need a one wider. To keep intake tract turbulence at a minimum, it's important that the manifold mates to the heads and carb in perfect alignment. Photo courtesy of Ram Jett Retainer.

"breakup" and lose power. Additionally, a larger needle valve assembly provides a greater fuel delivery rate to the bowl during high demand situations and a new float design with greater leverage gives more positive fuel control during high vibration conditions.

The older HS40 required a stiff throttle return spring to ensure its flat slide returned to a closed position during high intake manifold vacuum conditions. The HSR42 slide uses eight stainless steel roller bearings that provide smoother operation and reduce stiction caused by high manifold vacuum. The roller bearings also allow the use of a lower pressure throttle return spring for less throttle effort.

Fuel metering is improved on the HSR42 by incorporating a stainless steel gasket into its flat slide housing. The gasket helps seal the slide against the carb body, which results in more precise fuel metering at all throttle settings and good low speed throttle response.

For large displacement engines, 45mm and 48mm HSR series carbs should be available during late 1994 or 1995.

MIKUNI HSR42 SETUP

The HSR42 is available in kit form for Evo and Shovelhead Big Twins. Early Evo kits require a 1990 or later factory manifold; however, future kits will include a high-performance manifold. Ram Jett Retainer currently offers a high-performance manifold for the HSR42 and other companies are sure to follow.

For an air cleaner assembly on the HSR42 you can use the factory 1990 and later air cleaner unit. But for higher performance, a high-flowing Screamin' Eagle kit works better. Eventually HSR42 Evo kits will include a high-flow K&N filter assembly while all Shovelhead kits initially include the K&N filter.

Custom length throttle cables can be ordered by your local parts dealer from Barnett Engineering, but note that early kits need the original factory choke cable from a 1990 and later Keihin CV carb or HS40 choke cable for the HSR42's starter system.

TUNING CIRCUITS-HSR42

The same principles and concepts used for tuning the HS40 Mikuni are also applicable to the HSR42. The HSR42 has five jet circuits, which is one less than the HS40 since the replaceable pilot air jet has been eliminated. Three of the jet circuits have replaceable jets, one is fixed (needle jet) and one is adjustable (pilot air screw). Refer to Chapter 4 for a comprehensive description of the tuning circuits. This discussion will only cover differences between the tuning circuits of the new HSR42 and the older HS40.

Early models of HSR42 carbs are shipped from the factory with the following jet combinations: #30 pilot fuel jet, #160 main jet, #8DDYO1-96 jet needle, a fixed size needle jet and the pilot air screw set two turns out from fully bottomed.

Pilot Air screw

The adjustable *pilot air screw* is externally located on the left side of the HSR42 body (toward the rear of the bike as you face the carb's inlet side). The HSR42's *pilot air screw* works exactly opposite of the pilot screw on the older HS40. With the older HS40, the *pilot screw* supplied supplemental *fuel* to the idle/low speed circuit. With the HSR42, however, the *pilot air screw* supplies *air* to the idle/low speed circuit.

The *pilot air screw* essentially does the same job as the HS40's *pilot air jet*, which has been eliminated from the HSR42. Turning the *pilot air screw* in (clockwise) enriches the air/fuel mixture while turning it out leans the mixture. The acceptable range for the screw is between 1/4 and three turns out from fully bottomed.

Pilot Fuel Jet

The *pilot fuel jet* is located inside the float bowl area and controls the maximum amount of fuel that flows through the idle/low speed circuit. The *pilot fuel jet* of the HSR42 and HS40 are identical and therefore can be interchanged.

Needle Jet

The HSR42's *needle jet* is a long tube that screws into the bottom of the carburetor's body. It can be removed with a 5/16-inch deep-well socket after removing the fuel bowl drain plug. The needle jet works in conjunction with the tapered jet needle to control the air/fuel mixture mostly between 1/16 and 1/4 throttle opening. Unlike the older HS40, the HSR42's needle jet is available in only one size, so tuning is accomplished by changing either the position or size of the jet needle.

MIKUNI HSR42 JET NEEDLES

#8DDYO1-94	RICH
#8DDYO1-95	
#8DDYO1-96	Standard
#8DDYO1-97	
#8DDYO1-98	LEAN

Table E.1

Jet Needle

The tapered *jet needle* extends down vertically from the flat slide and passes through the center of the needle jet. It is attached to the slide by a metal E-ring that fits into one of five grooves. Five different needles are available for the HSR42. Two richer and two leaner needles than the standard #8DDYO1-96 needle are available.

The five grooves at the needle's top-end are for tuning adjustment. Raising the needle (placing E-ring in lower groove) enriches the mixture while lowering it (placing E-ring in higher groove) leans the mixture. The needle is preset at the factory in its middle position (third groove).

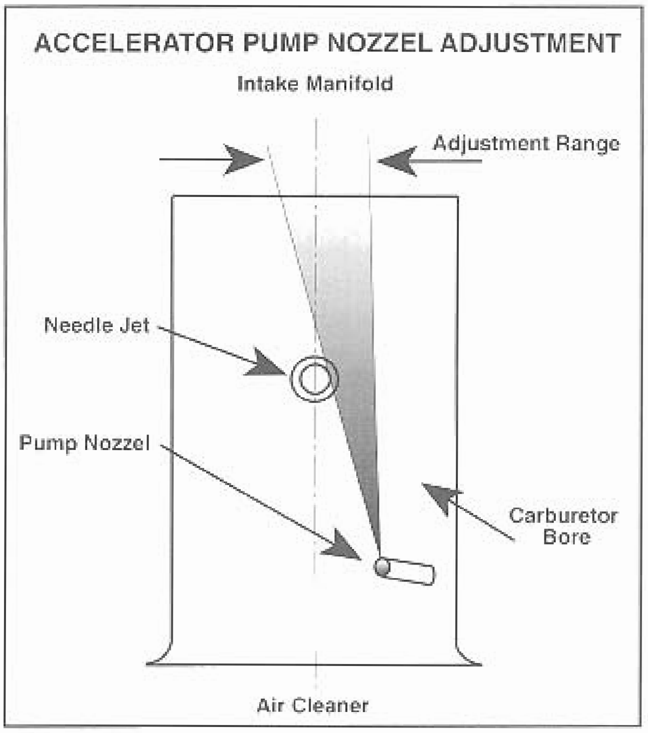
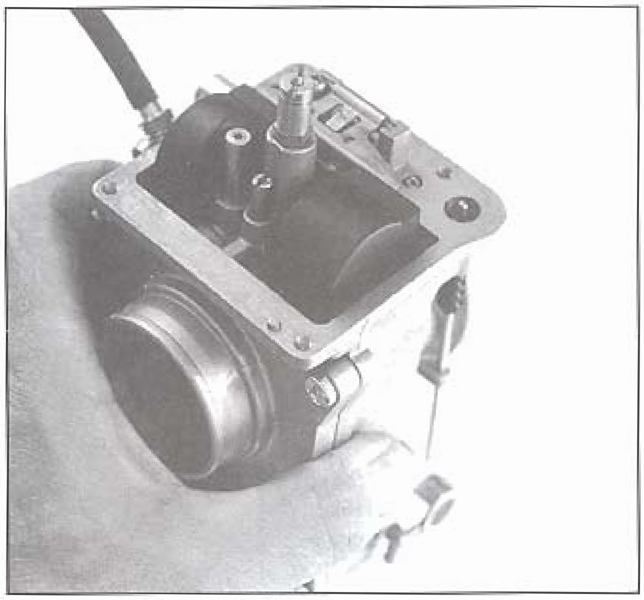


Figure E.1 *For maximum throttle response, be sure to adjust the Mikuni HSR42 accelerator pump nozzle so its spray is within the indicated range. The HS40's accelerator pump is also adjusted the same way.*



The Mikuni HSR42's fuel level is set by adjusting the float assembly. When the carburetor is held upside-down, the bottom of the float should measure about 18mm above the fuel bowl's O-ring surface. To eliminate potential problems, the float level should always be checked before installing a new carb.

Main Jet

The *main jet* screws into the bottom of the needle jet and controls the air/fuel mixture from about 3/4 to full throttle opening. The HSR42 is normally shipped with a #160 main jet. The HSR42 and HS40 main jets are identical and therefore can be interchanged.

Accelerator Pump

The HSR42 Mikuni includes an accelerator pump that is adjusted identically to the one on the older HS40 model. Be sure to position the pump nozzle correctly so its spray is within the range illustrated by Figure E.1. Use a smooth jaw needle nose pliers to gently turn the brass nozzle located in the throttle bore.

Float Level Adjustment

To check the HSR42 float level, remove the float bowl and turn the carb upside down. The float assembly's actuator tab should just begin contact with the needle valve assembly when the bottom of the float (its top side as you now view it) is 18mm (± 2mm) above the float bowl O-ring surface. A smaller measured distance above the O-ring surface results in a higher float level while a larger distance gives a lower level.

QWIK SILVER II CARBURETOR

The Qwik Silver II is a smooth bore, self-compensating, flat slide carburetor that became available in early 1993. Designed by Bill "Red" Edmonston, who also created the Lake Injector, Lectron Carburetor and Posa Fuel System, the Qwik Silver II has an aluminum and zinc die cast body that incorporates a unique flat slide design, which uses only one fuel circuit to control the fuel supply. This means there is no need for jets, air bleeds or an accelerator pump that typically have overlapping areas of operation, which can result in less than optimum air/fuel mixtures. Instead, the Qwik Silver II relies on a single needle and accurate sensing of pressure differences between the float bowl and venturi areas to meter fuel to the engine. The carburetor automatically compensates for changes in engine rpm, altitude and atmospheric conditions and is smog emissions certified by the California Air Resource Board. It's also all American-made and comes chromed plated.

The Qwik Silver II is available in 38mm, 40mm and 42mm versions for the Harley-Davidson and optionally can be ordered with components that retain use of the factory cruise control on FLH model bikes. With its high-flowing, smooth bore design and single fuel circuit, the Qwik Silver II provides excellent horsepower gains, increased throttle response and is easy to tune.

QWIK SILVER II SETUP

The Qwik Silver II is available in complete kit form or carb only versions. The kits include a high-performance intake manifold along with all necessary hardware for installation. The Qwik Silver II can be used with the factory 1990 and later Evo air cleaner assembly, the high-performance Screamin' Eagle unit, Rivera Engineering's "Smoothie" air cleaner, KüryAkyn's Hypercharger and others.

During installation, make sure the three circuit vent holes located on the carb's air intake side (air cleaner side) are not blocked by an air cleaner assembly, velocity stack or any other component. These holes allow atmospheric pressure to enter the fuel bowl area and starter enrichment circuit and must be unobstructed for correct carburetor operation. It should be noted

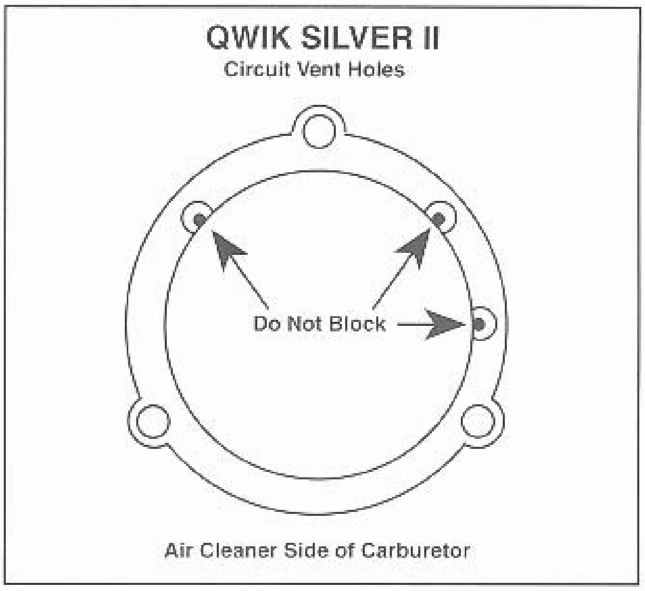
The Qwik Silver II is a high flowing flat slide carburetor that uses one fuel circuit for metering fuel in all rpm ranges. The entire circuit is controlled by one tapered needle, which is available in various sizes. This carburetor is available in 38/40/42mm sizes for Big Twins. High flowing design, simplicity, easy adjustments, and chrome plated die cast body are only some of its benefits. Carburetor kit includes manifold, air cleaner and necessary hardware. Photo courtesy of Qwik Silver II, Inc.

that a lean air/fuel mixture condition may develop at higher road speeds due to certain air cleaner backing plate designs even when the three holes are unobstructed. The lean condition is usually caused by a backing plate that creates a low pressure area around the holes even though the holes are not actually blocked. This condition requires modification of the plate.

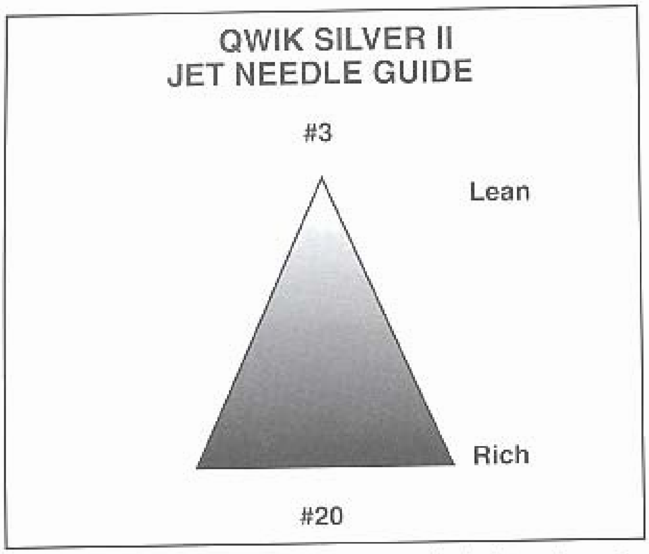
The Qwik Silver II's float level is set at the factory. To check the level, first remove the carb's fuel bowl. Notice that the fuel line connects directly to the fuel bowl and not to the carb body. This design allows checking the float level by holding the fuel bowl in your hand and then opening the fuel shutoff valve. The float is set correctly when the fuel level is between the center point and the top of the needle and seat. Bend the tab on the float either in or out to adjust the level.

QWIK SILVER II TUNING CIRCUIT

The Qwik Silver II has only one fuel circuit besides an idle speed adjustment. The fuel circuit's air/fuel mixture is controlled by either raising or lowering the needle or by replacing the needle with a thicker (leaner) or thinner (richer) version. Most carbs equipped for Harley-Davidson Big Twins are shipped with a #16 needle installed and include two extra needles — one



The three circuit vent holes allow atmospheric pressure to enter the fuel bowl area and starter enrichment circuit. To avoid a lean condition, especially at high road speeds, be sure the air cleaner or velocity stack does not block any of the holes.



The Qwik Silver II needle controls the air/fuel mixture throughout the engine's entire rpm range. Sizes are available from #3 (lean) through #20 (rich) besides a few special "-1" and "-2" needles for certain engine combinations.

richer (#17) and one leaner (#15). Needles are available in sizes: #3 (lean) through #20 (rich). Special needles with either a "-1" or "-2" added to their part number are optionally available. These needles are designed to enrich the lower throttle range and are helpful for tuning out "flat spots" with certain engine combinations.

To tune the Qwik Silver II's fuel circuit, first determine whether you want to alter the air/fuel mixture between idle and mid-throttle or mid-throttle and full throttle position. For idle to mid-throttle tuning, the carb's needle can be either raised or lowered. Black exhaust smoke, rough idle and lazy acceleration indicate a rich condition. For mid-throttle to full throttle tuning, replace the needle with either a richer or leaner version. For street riding, mid-to-full throttle tuning can be checked by quickly closing the throttle from wide open to 7/8 position when the engine's rpm is high. If the engine accelerates slightly, the needle is too lean. If the engine hesitates or misses slightly, the needle is too rich. If the engine just slows a slight amount, the needle is close to the correct size. Refer to Chapter 4 for additional carburetion tuning considerations.

To adjust the needle either up or down, first remove the 1/8-pipe plug located on top of the carburetor. Now use a small, flat screwdriver to turn the needle adjuster located in the exposed hole. The adjuster has about 30 clicks of adjustment from full rich to full lean. Turning the adjuster left (counterclockwise) leans the mixture, while turning it right (clockwise) enriches it. The adjuster is set from the factory at 14 clicks (counterclockwise) from fully bottomed.

To replace the needle, first remove the flat slide and then turn the needle's adjusting screw clockwise and count the clicks until it stops. Align the adjuster's outer and inner slots and remove the adjuster. Install the new needle into the slide along with the needle's spring, adjuster actuator pin and adjuster. Reset the new needle's position (number of clicks) to the same location as the replaced needle.

CARBURETION SELECTION GUIDE

74–86 Cu. In.	88–93 Cu. In.	96 Cu. In. & Up
Mikuni HS 40	Mikuni HSR 42	Mikuni HSR 42
Mikuni HSR 42	Qwik Silver II 40	Qwik Silver II 42
Qwik Silver II 38	S&S Super B	S&S Super G*
S&S Super B	S&S Super E	S&S Super D
S&S Super E	S&S Super G	

Table E.2 * For these engine displacements, the S&S Super G can be bored out up to .100-inch.

ADDITIONS and UPDATES
CHAPTER 8
IGNITION SYSTEM

DYNA 2000 IGNITION SYSTEM

The Dyna 2000 is a microprocessor based, programmable ignition system. It includes a control module that works in conjunction with either a Dyna S magnetic trigger or a stock Harley Hall Effect electronic trigger. The Dyna 2000 wiring harness is compatible with stock factory connectors and includes a wire for the Harley Vacuum Operated Electric Switch (V.O.E.S.).

The Dyna 2000 has four built-in ignition advance curves that are user selectable through switches located on the backside of the ignition's control module. This allows the user to tailor the advance curve to engines with various levels of modification, from stock to highly modified. Also included is a built-in programmable rpm limiter, which can be adjusted with switches from 6000 to 7500 rpm in 500 rpm increments.

This is the backside of the Dyna 2000 electronic ignition control module. Notice the panel with six switches located at the top right corner and the diagnostic LED at its right side. One switch is used to identify the type of trigger assembly, two switches designate the spark advance curve, two additional switches specify the engine's rpm limit and the last switch selects either single or dual-fire mode. Photo courtesy of Motorcycle Specialties.

The Dyna 2000 can be used in either single-fire mode (two ignition coils) or dual-fire mode (one ignition coil). A built-in tachometer correction feature is included for accurate single-fire tachometer operation along with a diagnostic LED indicator to assist in static ignition timing and trouble shooting.

DYNA 2000 SETUP

Since electronic ignitions are sensitive to radio frequency interference (RFI), carbon core, suppression spark plug wires with a resistance of at least 3000 ohms per foot must be used to minimize RFI. Be sure not to kink carbon core wires and shield them as much as possible from all other wires. Use high-performance 8mm silicone wires and periodically test them with an ohmmeter to ensure they haven't lost resistance due to usage or damage.

It's important to use ignition coils with the correct primary resistance. Use coils with 2.5 to 3.0 ohms primary resistance for either single or dual-fire operation.

DYNA 4000 Pro IGNITION SYSTEM

The Dyna 4000 pro is a high-performance ignition system designed specifically for the serious racer. By using specially designed ignition coils and a microprocessor based control module, the Dyna 4000 Pro outputs about twice the spark energy of the Dyna S ignition. It can be setup for either single or dual-fire operation and has a built-in two stage rev limiter. The Dyna 4000 Pro does not include a built-in ignition advance curve, so it must be used in conjunction with a mechanical advance mechanism.

The two stage rev limiter includes a launch limiter that helps a rider make consistent

The Dyna 4000 Pro electronic ignition is microprocessor based, offers single or dual-fire operation and includes a built-in two stage rev limiter for launch and over rev engine control. It delivers a powerful spark and smooth rpm control. Its high energy inductive ignition design is intended for use on the race track by the serious racer. The launch limiter switch (left) can be set from 3500 to 7000 rpm and the over rev switch (right) from 6500 to 10,000 rpm. Photo courtesy of Morocco Racing Products.

The Dyna 2000 and 4000 Pro ignitions both must be used with carbon core suppression plug wires and can use the Dyna S pickup as a trigger assembly. The 4000 Pro system also requires a special dual magnet rotor attached to the camshaft. Be sure to periodically test the resistance of the carbon core wire with an ohmmeter. Photo courtesy of Morocco Racing Products.

IGNITION FEATURES GUIDE

Manu-facturer	Single-Fire	Dual-Fire	Magnetic Sensor	Optical Sensor	Mechanical Advance	Electronic Advance	Adjustable Advance	RPM Limiter	Separate Cyl. Timing	Multi-Spark	High RPM Retard	VOES Switch
Accel		Yes	Yes			Yes	Yes	*	Yes			
Carl's CDI Magneto	Yes	Yes	Yes					*				
Compu-Fire	Yes	Yes	Yes			Yes	Yes	Adjust-able				Yes
Crane HI-4	Yes	Yes	Yes			Yes	Yes	Adjust-able	Yes			Yes
Dyna-S	Yes	Yes	Yes		Yes			*	Yes			
Dyna 2000	Yes	Yes	Yes			Yes	Yes	Adjust-able				Yes
Dyna 4000 Pro	Yes	Yes	Yes		Yes			**Adjust-able			***Yes	
MC Power Arc	Yes			Yes		Yes	Yes	Adjust-able	Yes	Yes	Yes	Yes
Screamin' Eagle		Yes	Yes			Yes		*Fixed at 8000				Yes
1200cc XL Module		Yes	Yes			Yes		Fixed at 6250				Yes
Stock Electronic		Yes	Yes			Yes		Fixed at 5250				Yes

Table E.3 *External RPM Limiter Can Be Added. **Two-stage RPM Limiter. ***Requires separate Retard Module.

launches from the starting line by accurately holding the engine at a preset rpm just before launch. The launch limiter is activated by a clutch switch and the launch rpm limit is selected by adjusting a rotary switch located on the control module. The switch can be set from 3500 to 7000 rpm in 250 rpm increments. The rev limiter's second stage is an over rev limiter, which is adjustable by setting a second rotary switch. The over rev limiter is adjustable from 6500 to 10,000 rpm in 250 rpm increments.

The Dyna 4000 Pro uses the Dyna S as a trigger mechanism and a special dual magnet rotor. The 4000 Pro automatically shuts off when the engine is not running and includes a diagnostic LED indicator to assist in static ignition timing and trouble shooting.

A Dyna Programmable Retard Module can be purchased separately and connected to the Dyna 4000 Pro system. The retard module has a single phase function that can retard the engine's ignition advance at any time during a run. The amount of retard can be set to any value between 2 degrees and 20 degrees in 2 degree increments.

Retarding ignition advance a few degrees at the end of a run can sometimes improve top-end performance because the engine is more heated than when it left the starting line. By slightly backing off the advance under such high heat, high load conditions, potential detonation may be eliminated, which can allow the engine to develop more power. Furthermore, 60 foot times also may be helped because the engine now may tolerate more total advance.

DYNA 4000 Pro SETUP

The Dyna 4000 Pro ignition must be used with Dynatek 0.7 ohm blue ignition coils, Dyna S ignition trigger, special Dyna S dual magnet rotor and 8mm carbon core suppression spark plug wires. Because of the low ohm ignition coils, this system is intended only for the race track where usage is limited to short time periods.

For maximum performance with single-fire operation and dual plug heads, the Dyna 4000 Pro should be setup to use four ignition coils with one coil driving each spark plug. Using one coil per spark plug allows true single-fire operation and delivers two coils worth of energy when generating each spark. Keep in mind that the same rules for carbon core suppression spark plug wire as discussed with the Dyna 2000 ignition system also apply here. During installation, be sure that the magnets located on the rotor and pickup assembly are in alignment. If necessary, shim the rotor so the magnets are aligned. ❖

About The Author

D. William Denish grew up in the Midwest and as a teenager was introduced to Harley-Davidson dirt track racing by his uncle. The roar of the engines and the excitement of racing turned his cursory interest in motorcycling into a deep passion for owning a Harley. In 1958 as a 14 year old, he reluctantly bought his first motorcycle—a 10 horsepower Mustang. Reluctantly he says, because the Mustang was not exactly a Harley. But at that time a 14 year old was restricted to a five horsepower engine. One had to be 16 to own the real thing—a Harley. Well, a 10 horsepower scooter disguised as a five horsepower model had to do for now. Before the summer ended the Mustang had a big Amal carb, high compression head and racing cam.

As it turned out, D. William never got a Harley at 16. Instead, it took until 1964, at age 21 to realize the dream. Officially the Harley was called an XLCH Sportster. The "C" stood for competition. Unofficially, it was called the Corvette of the motorcycle world—rugged, super fast and all 55 cubic inches bulging with torque and delivering 55 horsepower at 6300 rpm—good for a 14.75 E.T. and 92 MPH in the quarter mile.

Although the Sportster's engine idle alone discouraged challengers, the only way to fly was with a stroker motor. So, during the winter of 1964, out went the stock engine parts and in went long stroke KH flywheels, "Doc" Dytch two ring stroker pistons, Sifton "minus-plus" cams, big valves, bored-out Linkert carb with velocity stack, gutted mufflers and close-ratio transmission gears. The reworked 65 inch engine pushed quarter mile performance to 12.10 E.T. at 110 MPH.

The winter of 1967 saw the addition of Dytch big bore cylinders, Sifton "minus minus" cams, stronger S&S flywheels and beefed up crankcases. Now with 74 cubic inches, performance improved to 11.40 E.T. at 117 MPH. One of the first new S&S carburetors was installed in 1968. Eventually the engine was bumped up to 86 cubic inches with Trock cylinders and Branch heads. The increased displacement, stronger transmission parts, more radical cams and better flowing heads improved performance to 11.10 E.T. at 121 MPH with a narrow rear tire and no wheelie bar.

Another project was a bored and stroked 98 cubic inch Shovelhead that included an S&S Sidewinder kit, Baisley heads, Red Shift cam and S&S carb. The bike turns low 11s at about 118 MPH in street trim. For long distance touring, an Evolution "dresser" sits waiting to hit the open roads. An Andrews cam, Mikuni carb, MC Power Arc ignition and high flowing heads make it easy to pass an eighteen wheeler riding two up.

D. William's motorcycle experience spans more than 37 years of which the last 31 have been devoted to building, riding and racing Harleys. He has been published in various motorcycle magazines and also has written a companion book entitled *The V-Twin Tuner's Handbook*. His enthusiasm and dedication in authoring *The Big Twin High-Performance Guide* affirms his interest in sharing his knowledge with Harley enthusiasts searching for more performance. ❖

Dedication

To all of the pioneers

inventors and designers
engineers and machinists
engine builders and fabricators
tuners and racers

who have shown the way