DeskTop DYNOS

COMPLIMENTARY MINI-GUIDE

Engine Builder's Guide To Filling & Emptying Simulation And High-Performance Design

INCLUDES:

- Details Of All On-Screen Menu Choices/Selections
- Motion's Filling & Emptying Simulation Basics
- Cylinderhead Flow And Discharge Coefficients
- Understanding And Calculating Camshaft Timing
- Ram Tuning And Pressure Wave Dynamics
- Expose Fallacies About Engine Components
- Complete Glossary And Much, Much More!

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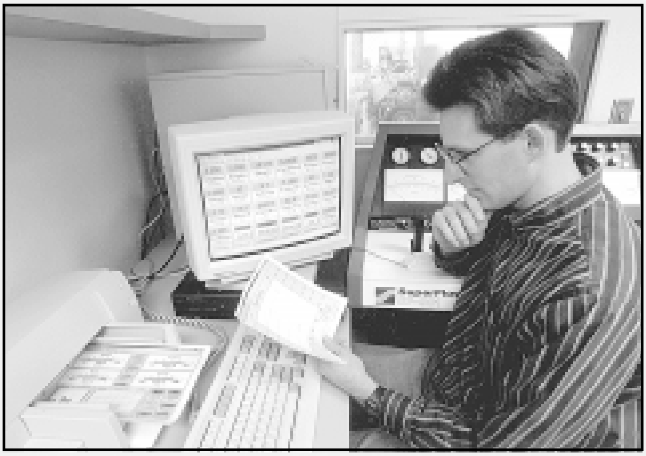
BY:

LARRY ATHERTON



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INTRODUCTION

This is not a typical "hot rodding" book. Yet it is written specifically for you: The amateur or professional automotive enthusiast interested in both engines and computer simulations. This book, and it's "big-brother" the complete DeskTop Dynos book, may reveal more to you about high-performance and racing engines than you've discovered in all the other "enthusiast" books and magazines you've read. I know this to be true, because I have been an engine and book enthusiast all my life, and the nearly two-year process of writing these books taught me more about high-performance engines than my thirty years of engine building, tinkering, and racing.

Most books, like many products aimed at the consumer, begin with a publisher (a manufacturer) "discovering" a need in the marketplace. Every once in a while, though, a book is published that falls outside of the typical form. It is born from a unique collaboration—the result of fortuitous events—that could never have been planned. This is one of those books.

Two years ago, the DeskTop Dynos book and Mini Guide were simply to be a comprehensive guide for Motion Software users of Filling-And-Emptying engine simulation software. What the author failed to realize at first was that there is very little difference between understanding engine simulations and understanding engines themselves. In retrospect, it's obvious: If a simulation is a good one, it must mimic the physics at work inside the IC engine. Understanding engine simulations is simply another approach to understanding IC engines.

But engine simulations have been around for years, and there are many talented engineers, developers, and university professors that understand the physics and math that make the IC engine tick. *What's very rare is the individual who has this knowledge and understands racing engines and the needs of the high-performance engine builder.* That individual is Curtis Leaverton, the developer of Motion Software's engine simulation and my associate in the development of these books. This Mini Guide is, to a great extent, a transcription of a small portion of his vast knowledge, with my contribution being one of organizer and presenter.

Despite the inauspicious beginnings and its odd evolution, these books have become true "hot rodders" engine guides. Engine books—the better ones, at least—are a compilation of the learned experiences of the author/racer. They describe what components and procedures are known to work, and they conjecture about what is less understood or what seems to defy logic. This "outside-in" approach is completely understandable. It is a direct result of the most common method of engine development used by engine builders and racers: Trial-and-error. The DeskTop Dynos books follow a new path. They describe engine function from the "inside-out." You won't find a list of bolt-on parts that work on only a few engines, rather you'll discover information that will help you understand

WHY engines "need" parts of a specific design to optimize horsepower.

Admittedly, these books only "scratch the surface" of the vast and complex fields of engine-pressure analysis, thermodynamics, and gas dynamics, but that's actually what makes these books "work." There are many other publications that delve deeply into these subjects, the *Bibliography* on page 108 lists a few of these tomes, and many are not especially "approachable."

Hopefully, what you find here will be the right mix of theory and practical application. If you are able to gain a deeper understanding of the IC engine from these pages and from using the DeskTop Dyno software, I will consider this project a success.

Larry Atherton

Acknowledgments

This book simply would not have been possible without the technical assistance of Curtis Leaverton. Over the last few years I have spent many enjoyable days with Curtis discussing engines and simulation science. He is one of only a handful of people I've encountered in my life who can explain complicated subjects with such enthusiasm that they not only become clear to the listener, but the sheer joy of being guided toward the understanding becomes an indelible memory! In short, Curtis is a terrific teacher (and I like to think, my friend). Occasionally, he conducts seminars throughout the country, and I encourage every reader of this book to avail themselves of the unique opportunity to be "taught by one of the masters."

I also wish to thank the many manufacturers that provided photos and information for this book, especially Harold Bettes of SuperFlow Corporation. Harold, a lot like Curtis Leaverton, has no shortage of enthusiasm and excitement about engines and performance. He has also been extremely generous with his valuable time, and I have learned a great deal from the many hours of conversation we have had (over some great Chinese dinners!). Thanks Harold.

A heartfelt thanks to all the manufacturers that helped with this publication:

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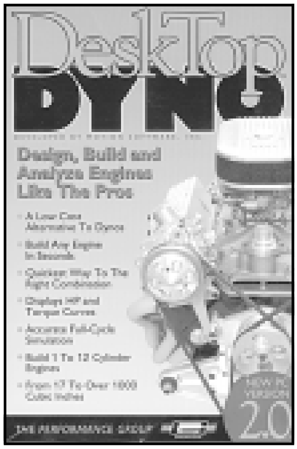
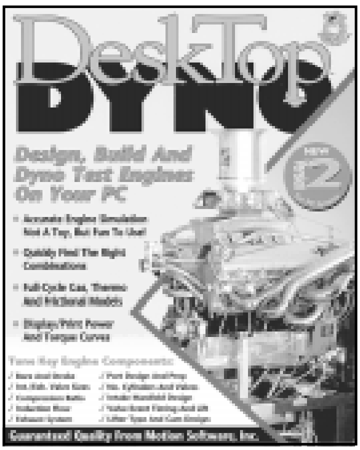
Another special thanks to Jim Dennewill for the many fine drawings throughout this book. Jim's talents and patience are both rare and boundless. Thank you for all your help and your friendship.

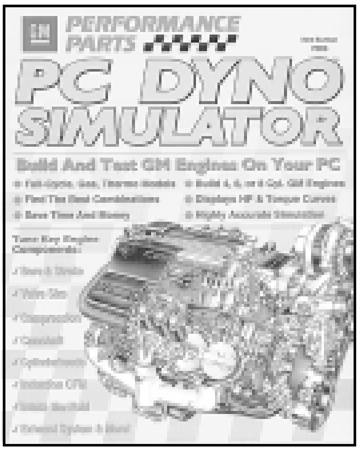
Finally, a sincere thanks to Paul Hammer. I can't imagine what this book project would have been like without Paul's dedicated help in "putting out the fires" around me (and I can imagine some pretty awful stuff!).

Filling & Emptying By Any Other Name

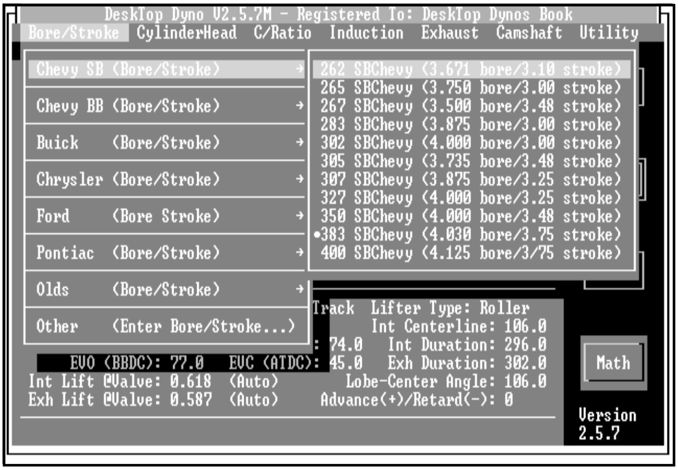
Motion Software recently released engine simulation software for IBM-compatible "PC" computers. These simulation programs are based, as of the writing of this book, on the *Filling-And-Emptying* method of full-cycle analysis as applied to the 4-stroke IC (internal combustion) engine. These software packages are available under various names, including the *DeskTop Dyno* (available from Mr. Gasket Corporation and from Motion Software, Inc. and its distributors) the *GM PC Dyno Simulator* (available from GM Performance Parts) and others. Each of these packages has individual functional differences, but all of these products use the same simulation methodology and many of the menu choices are comparable. Because of these similarities, the information in this guidebook applies equally to version 2.0, 2.5, and later simulation software releases. Minor functional differences will be discussed as required. Most screen illustrations in this book depict the latest version of the software (version 2.5.7—in the final stages of development when this book was published). This guide is intended to help you obtain a clearer understanding of overall program function, what is covered by each menu group, the implications of individual choices within the menus, and how to interpret simulation results. In addition, we will illustrate what can and cannot be modeled and why, the drawbacks of the current simula-







Motion Software, Inc., has recently released engine simulation software for IBM-compatible "PC" computers. These programs are based on the *Filling-And-Emptying* method of full-cycle analysis. They are available under various names, including the *DeskTop Dyno*, the *GM PC Dyno Simulator* and others.



Motion's Filling And Emptying simulation offers an inexpensive and rapid way predict power and torque to within 5% of actual dyno figures. The various choices in the Bore/Stroke menu are shown here (version 2.5.7—in development when this guidebook was published—includes expanded menu choices and several other enhancements, including a *Cam Math Calculator*).

tion software, and what advances may be possible in the near future.

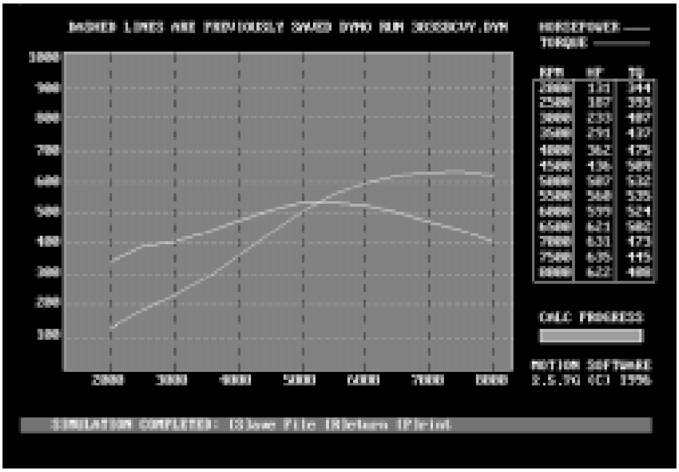
Motion Engine Simulation Basics

Motion engine simulations have been optimized for rapid "what if" testing. Engine component parts are grouped in menus along the top of the screen. One or more selections from each menu group effectively "builds" a test engine (called a "paper engine" in simulation parlance). Then, simply pressing "RUN" begins a full-cycle analysis of cylinder pressures. The simulation plots a graphic, on-screen representation of horsepower and torque for easy visual analysis. It is a simple procedure to save the test, select a new part or component dimension from one of the menus, and rerun the simulation to perform back-to-back testing. The changes in power and torque that would occur if a "real" engine were built and tested with the components at hand are clearly displayed in the graphic curves. Remarkably, this entire process often takes less than one minute on most computer systems.

Ease of use has always been an important design element. Without it, many of the tens of thousands of automotive enthusiasts that have successfully "assembled and tested" engines would not have been able to use our simulation programs. Imagine if one of the necessary inputs requested: "Enter the intake port flow at each 0.010-inch of the valve lift up to the maximum valve lift." Yet this information is absolutely necessary to perform a true engine simulation. To make this software technology available to as wide an audience as possible, the information requested by the program is straightforward and generally known by most enthusiasts. The program uses this "basic" information to derive flow curves, cam profiles, frictional models, induction characteristics, and other "technical details." While the widely understood terms in the menus have made these programs accessible, their less-than-exact wording has left some experienced users scratching their heads. Engine builders and other advanced users are often very aware of the "internal complexities" of the IC engine and may not realize at first glance how Motion's simulations handle many of these important considerations.

Furthermore, some performance enthusiasts are disappointed when they realize that engine simulations do not include some of the components they were expecting to test. A short list of these might include: block and head metallurgy, piston types and dome shapes, head-gasket thickness, combustion-chamber shapes, oils, oil-pan designs, ignition timing, bolt torque loads, and many more. Modeling many of these elements would require very complex techniques (with concomitant complex inputs from the user, not to mention extend calculation times) and would reveal only relatively small power differences. These limitations are discussed throughout this guide,

The simulation described in this guide is easy to use and provides a remarkable level of predictive power. If you have a tendency to "dismiss" this simulation because it seems to consist only of simple pull-down menu choices, we encourage you to take a second look. We believe this software, at under $50, offers terrific value!



but a good example of complexity vs. practicality can be found in oil pan testing. In order to simulate the conditions inside an oil pan, here are a few of the inputs that the user would have to address: dimensions of pan and lower crank/block contours, position of oil pump, positions and sizes of baffles, trays and screens, oil viscosity and temperature, level of oil in pan, acceleration and directional vectors (and how these vectors change over time), and more. Simply gathering together the needed information would be quite a project!

While it would be wonderful if an inexpensive computer program could simply and quickly zero-in on the optimum combination of all components for any intended application, that time has not yet come. However, most of the engine components that play a major role in power production _are_ modeled in Motion's Filling-And-Emptying simulations. Cam timing, compression ratio, valve size, cylinderhead configuration, bore and stroke, induction flow, and manifold type are just some of the modeled elements that have major effects on engine power.

The simulation discussed in this guide is easy to use and provides a remarkable level of predictive power. The designers have purposely avoided complex areas that would either make data entry difficult or greatly extend computational times. If you have a tendency to "dismiss" these simulations because they seem to consist only of simple pull-down menu choices, we encourage you to take a second look. Motion's Filling And Emptying simulation offers an inexpensive and rapid way to select component combinations that produce power and torque curves often within 5% of optimum for applications that lie within the range of the simulation. We believe that's quite an accomplishment for a software program you can load in your PC for under $50!

That brings us to the main purpose of this guidebook. The information presented here revolves around detailed descriptions of every item listed in the on-screen component menus. You'll discover the assumptions made by Motion programmers with respect to each of the possible choices and combinations. Within sections that discuss each menu category, you'll also find substantial background information that can be helpful for both your simulated and real-world engine building projects. This information was compiled from the feedback of thousands of users, hundreds of beta testers, and countless hours of testing and exploration. We are confident that what you find here will make using Motion engine simulation software easier and your engine analysis more productive.

THE ON-SCREEN MENU CHOICES:

After starting Motion's engine simulation, the user can follow two paths of engine testing: To recall a previous test (using the

UTILITY menu) or to “build” an engine from scratch. Assuming the latter selection, a common choice is to start with the leftmost **Bore/Stroke** menu then work, menu-by-menu, from left to right. While there are no restrictions dictating the order in which menus must be opened, the upcoming sections in this guide follow the “natural” left-to-right progression taken by most “engine builders.”

THE BORE/STROKE MENU

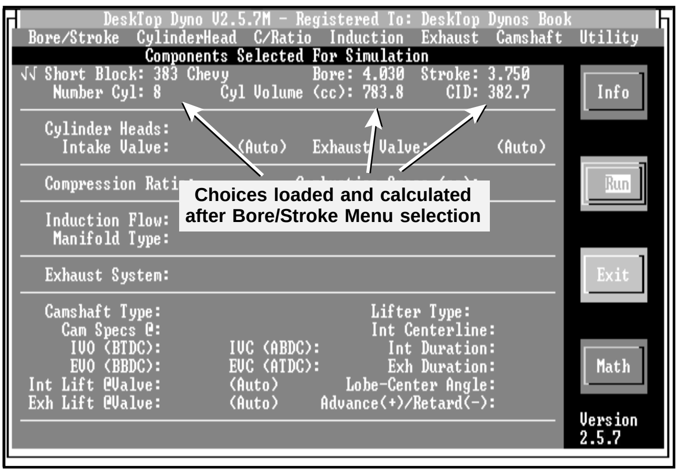
The **Bore/Stroke** menu is located on the left end of the menu bar. By opening this menu, you are presented with a variety of “pre-loaded” engine configurations. If any one of these choices is selected, the appropriate bore, stroke, and number of cylinders will be loaded in the SHORTBLOCK section of the on-screen Component Selection Box. In addition to selecting an existing engine configuration, you can scroll to the bottom of the Bore/Stroke menu and choose the “Other...” option. This closes the menu and positions the cursor in the Component Selection Box, permitting direct entry of bore, stroke (in inches to three decimal places) and number of cylinders. At each data input position, a range of acceptable values is displayed at the bottom of the screen. As with all numerical input, only values within the range limits will be accepted by the program.

What’s A “Shortblock”

When a particular engine combination is selected from the Bore/Stroke menu, the bore, stroke, and the number of cylinders are “loaded” into the on-screen Component Selection Box. These values are subsequently used in the simulation process of predicting horsepower and torque. The Bore/Stroke menu choices should be considered a “handy” list of common engine cylinder-bore and crankshaft-stroke values, not a description of engine configurations (e.g., V8, V6, straight 6, V4, etc.), material composition (aluminum vs. cast iron), the type of cylinderheads (hemi vs. wedge) or any other specific engine characteristics. The Bore/Stroke menu only loads **bore**, **stroke**, and the **number of cylinders** into the program database.

Bore, Stroke, & Compression Ratio

After making a selection from the Bore/Stroke menu, or when the individual bore, stroke, and number of cylinders have been entered manually, the swept cylinder volume (in cubic centimeters) and the total engine displacement (in cubic inches) will be calculated and displayed in the on-screen Component Selection Box. The swept cylinder volume measures the volume displaced by the movement of a single piston from TDC (top dead center) to BDC (bottom dead center). This “full-stroke” volume is one of the two essential elements



When a selection is made from the Bore/Stroke menu, the bore, stroke, and the number of cylinders are “loaded” into the on-screen Component Selection Box. In addition, the swept cylinder volume (in cubic centimeters) and the total engine displacement (in cubic inches) will be calculated and displayed.

required in calculating compression ratio. We'll discuss compression ratio in more detail later in this guide, but for now it's helpful to know that compression ratio is determined with the following formula:

Compression Ratio =

$$\frac{\textbf{Swept Cyl Vol + Combustion Space Vol}}{\textbf{Combustion Space Vol}}$$

In other words, the total volume that exists in the cylinder when the piston is located at BDC (this volume includes the Swept Volume of the piston and the Combustion Space Volume) is divided by the volume that exists when the piston is positioned at Top Dead Center. For the time being, it's important to keep in mind that your selections of bore and stroke dimensions greatly affect compression ratio. When the stroke, and to a lessor degree the bore, is increased while maintaining a fixed combustion-space volume, the compression ratio will rapidly increase. And, as is the case in Motion's simulation software, if the compression ratio is held constant—because it is a menu selection and, therefore, fixed by the user—the combustion space volume must increase to maintain the desired compression ratio. This may seem more understandable when you consider that if the combustion-space volume did not increase, a larger swept cylinder volume (due to the increase in engine displacement) would be compressed into the same final combustion space, resulting in an increase in compression ratio.

One example of how bore and stroke can have a significant affect on compression ratio is demonstrated in a destroked racing engine. Engine designers have realized for many years that shorter-stroke engines waste less power on pumping-work (more on this later), leaving more horsepower available for rotating the crankshaft. Furthermore, if engine displacement is held constant (by a concomitant increase in bore size), a larger cylinder diameter will accommodate larger valves, and increasing valve size is one of the most effective ways to improve breathing and horsepower. All of these changes taken together can produce considerable power increases, *providing the compression ratio is maintained or increased*. Unfortunately, it is difficult to maintain a high compression ratio in short stroke engines because: 1) Overall swept volume is often reduced, 2) combustion space volume is proportionally larger, and 3) short stroke engines move the piston away from the combustion chamber more slowly requiring increased valve-pocket depth to maintain adequate piston-to-valve clearance.

It is easy to use your PC to simulate a 1-inch stroke, 5-inch bore, 8-cylinder engine of 157 cubic inches with an 11:1 compression ratio that produces over 500hp at 8000rpm (with the power continuing to climb rapidly!). Unfortunately, it may be nearly impossible to build this engine with much more than 9 or 10:1 compression because the volume needed for the combustion chamber, valve pockets, and head gasket on a 5-inch bore is large compared to the swept volume produced by the short 1-inch stroke. If we installed typical small-block race heads on this short-stroke configuration and great care was used in machining the valve pockets, a total combustion space of about 55cc's might be possible. Unfortunately, this would only produce 7:1 compression. The entire combustion-space volume would have to be less than 32 cubic centimeters to generate a compression ratio of 11:1 or higher!

Bore And Stroke Vs. Friction

You know that selecting stroke length will determine, in part, the cubic-inch displacement of the engine. And from the previous section you now realize that obtaining high compression ratios with shorter stroke engines can be quite difficult. What you may not realize is that setting the stroke length determines, to great extent, the amount of power lost to friction. The stroke fixes the length of the crank arm, and that determines how fast the piston and ring packages "rub" against the cylinderwalls at any given engine speed. And here's another rub (pardon the pun): *70% to 80% of all IC engine frictional losses are due to piston and ring-package "drag" against the* *cylinderwall!* If a 1-inch stroke engine running at 8000rpm looses 10 horsepower due to cylinderwall friction, the same engine will loose 25hp with a 2-inch stroke, 40hp with a 3-inch stroke, 70hp with a 4-inch stroke, 90hp with a 5-inch stroke, and 120hp with a 6-inch stroke at the same 8000rpm crank speed. That means this 6-inch stroke engine consumes 110 more horsepower than its 1-inch stroke counterpart just to drive the pistons and rings up and down their bores!

Try this simulation on your PC to demonstrate frictional losses. Build a 603.2 cubic-inch engine, we'll call it a very "Long-Arm" smallblock, with the following components:

Bore:	**4.000 inches**
Stroke:	**6.000 inches**
Cylinderheads:	**Smallblock/Stock Ports And Valves**
Valve Diameters:	**2.02 Intake; 1.60 Exhaust (Valve diameters must be manually selected to disable the "Auto Calculate" function. This keeps the valve size fixed when the bore size is changed—required for the next test.)**
Compression Ratio:	**10.0:1**
Induction Flow:	**780 CFM**
Intake Manifold:	**Dual Plane**
Exhaust System:	**Small-Tube Headers with Open Exhaust**
Camshaft:	**Stock Street/ Economy**
Lifters:	**Hydraulic**

This combination produces the power and torque curves shown in the accompanying chart (Long-Arm). Note that the power drops to zero above 5500rpm. Translated, this means that at 5500rpm the engine is using all the power it produces to overcome internal friction!

Now build the same displacement engine, but change the bore and stroke combination to:

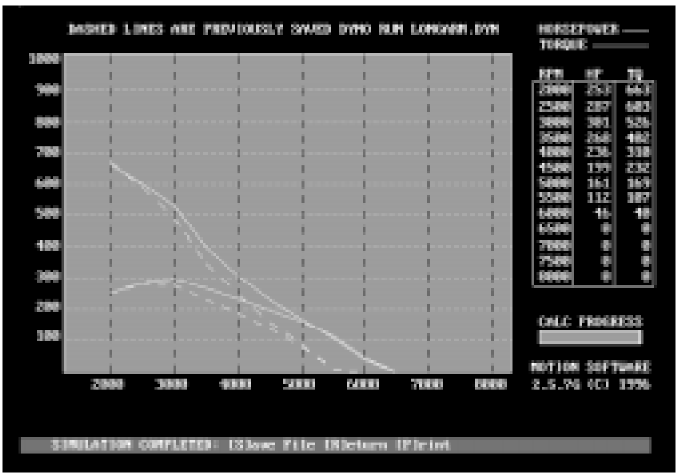
Bore:	**5.657 inches**
Stroke:	**3.000 inches**

This "Short-Arm" configuration displaces the same 603.2 cubic inches, but it produces over 100hp at 5500rpm and maintains a "non-zero" power level until about 6500rpm. Where was the "extra" horsepower hiding? The majority is "freed" by lower piston speeds and reduced bore-wall friction from the 3-inch shorter stroke.

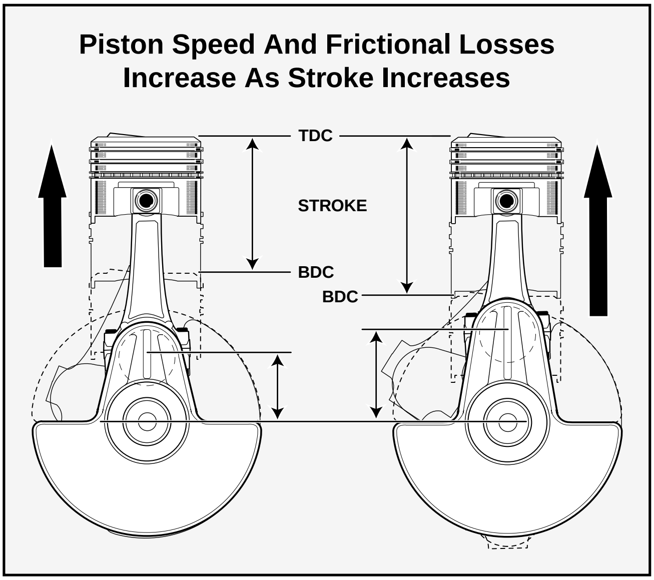
Motion simulations use an empirical equation to calculate frictional mean effective pressure (Fmep):

Fmep = (12.964 + (Stroke x (RPM / 6))) x (0.0030476 + (Stroke x (RPM / 6) x 0.00000065476))

Once the Fmep is known, the horsepower consumed by friction can be calculated with



This "Short-Arm" configuration (solid lines) displaces the same 603.2 cubic inches as the "Long-Arm" test (dotted lines), but it produces nearly 100hp more at 5500rpm and maintains a "nonzero" power level until about 6500rpm. Where was the "extra" horsepower hiding? The majority is "freed" by lower piston speeds and reduced bore-wall friction.



The stroke length determines, to great extent, the amount of power lost to friction. *70% to 80% of all IC engine frictional losses are due to piston and ring-package "drag" against the cylinderwall!* A longer stroke increases the length of the crank arm, and that increases the speed of piston and ring contact with the cylinderwall.

the following equation:

Friction HP =
(Fmep x CID x RPM) / 792,000

Applying these equations to the two previous test engines reveals the following:

Power consumed by friction with the 6-inch stroke engine @ 5000rpm: **120.7hp**

Power consumed by friction with the 3-inch stroke engine @ 5000rpm: **44.8hp**

So far we've discussed how stroke length affects displacement, compression ratio, and frictional losses within the engine. In addition to these effects, bottom-end configuration can have a measurable effect on power consumed by the processes of drawing fresh fuel/air mixture into the engine and forcing burned exhaust gasses from the cylinders (a process called *pumping,* consuming what is termed *pumping work*). However, the IC engine is a remarkably complex mechanism that can sometimes fool nearly anyone into believing that they have discovered subtle, "secret" power sources. The logic behind the discovery may seem to make perfect sense, but eventually turn out wrong or misleading. This is the case with the belief purported by some "experts" that: 1) Shorter-stroke/larger-bore engine configurations have the potential to be more efficient high-speed air pumps, and 2) Shorter strokes are the key to producing high-speed horsepower. Let's take an in-depth look at how bore, stroke, and rod length affect piston speed and pumping work.

Fallacy One: Stroke Length Vs. Pumping Work

The concept of pumping work is more easily understood when you remember that the IC engine is, at its heart, an air pump. Air and fuel are drawn into the cylinders during the intake cycle, and burned exhaust gasses are pumped from the cylinders during the exhaust cycle. The power required to perform these functions is called *pumping work* and is another source of "lost" horsepower.

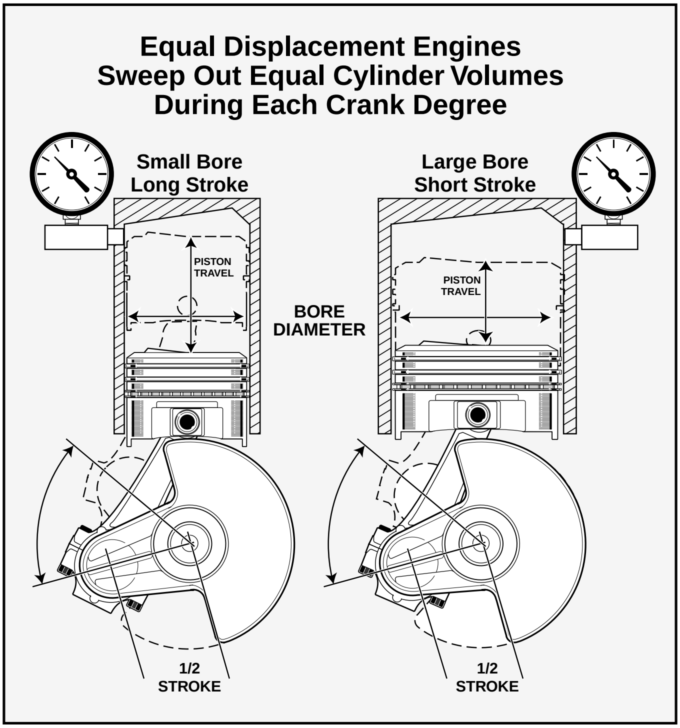
The intake stroke begins with the piston accelerating from a stop at TDC. As it begins to move down the bore, the swept volume within the cylinder increases, creating a lower pressure. This drop in pressure causes an inrush of higher pressure air and fuel from the intake system to compensate or "fill" the lower pressure in the cylinder. As the piston continues to accelerate down the bore, the speed of the induced charge also increases, until at about 70-degrees after top dead center, the piston reaches maximum velocity. At this point, the greatest pressure drop exists within the cylinder, drawing outside charge with the greatest force. It is this “force” that is the key element in understanding pumping work. The differences in pressures that cause the inrush of air and fuel are not a freebie from nature; they consume work. The greater the difference in pressure between the intake manifold and the cylinder, the more pumping work will be required to move the piston from TDC to BDC. Any increase in pumping work directly reduces the power available at the crankshaft.

Let’s consider various techniques that can reduce pumping work on the intake stroke and potentially increase power output. First of all, pumping work can be substantially reduced by lowering cylinder volume. A smaller cylinder takes less work to fill it. Unfortunately, reducing cylinder volume also reduces the amount of air and fuel that can be burned on the power stroke, lowering power output more than any possible gains from a reduction in pumping work. So if power is the goal, reducing pumping work must be accomplished without decreasing the swept volume of the cylinder.

The question is how can we fill the same space using less work? This can be translated to read: How can we move the same or more air/fuel volume into the cylinder while inducing less pressure drop between the cylinder and the intake system? Solutions to this problem are widely used by racers and include larger intake valves, freer-flowing ports and manifolds, larger carburetors or injector systems, and tuned-length intake runners. All these techniques allow the same or more air/fuel mixture to flow into the cylinder while consuming less pumping work.

While it’s true that shorter-stroke engines consume less frictional horsepower to drive the pistons up and down the cylinders (as described in the last section), it may also seem logical that a shorter stroke engine, with its slower piston speeds, can reduce pumping work. Here’s how this concept is often described and how the underlying thread of truth often remains undiscovered: First, picture a 6-inch stroke, small-bore engine. When the piston begins to move

While it’s true that shorter-stroke engines consume less frictional horsepower, it is not true (as believed by some) that shorter-stroke engines, with their slower piston speeds, generate less pumping work. For equal displacements, equal cylinder volumes are swept out at each increment of piston travel from BDC to TDC. Short-stroke/large-bore engines pump just as much air as long-stroke/small-bore engines.

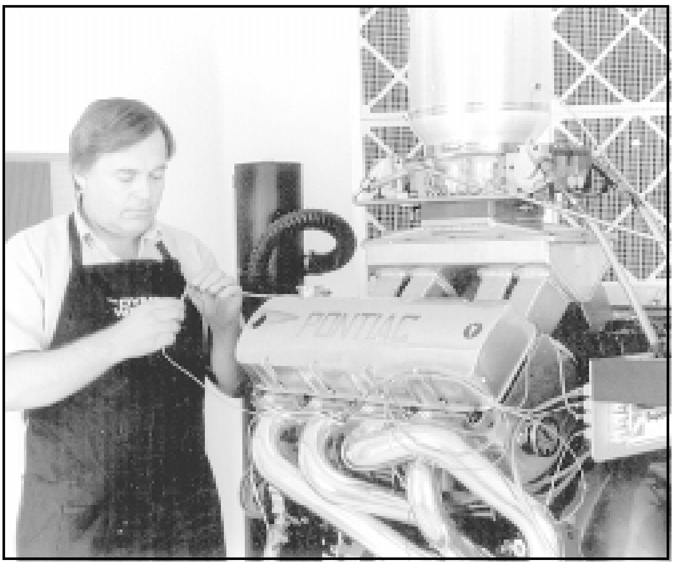


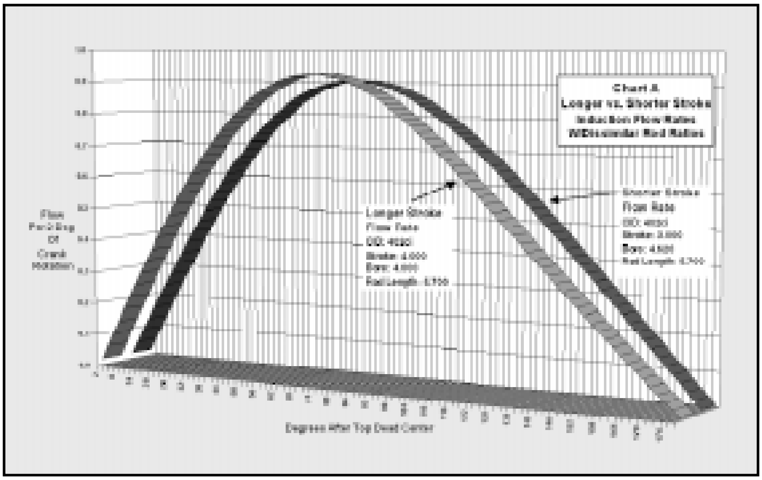
away from TDC, the long stroke sends it quickly towards a maximum velocity (maximum volume change) that occurs about 70 degrees ATDC. At this point, the long stroke has accelerated the piston to very high speeds that generate a strong pressure drop in the cylinder. It is claimed that this low pressure generates flow velocities so high that the cylinderheads, valves, and the induction system become a significant restriction to flow. The rapid buildup in flow and high peak flow rates lower pumping efficiency, and the overall picture gets worse as engine speed and piston speed increase. Now consider a 1-inch stroke, large-bore engine. Because the stroke is much shorter, peak piston speed—and therefore peak pressure drop—are believed to be considerably lower. The reasoning continues that induction flow never reaches as high a rate but is spread out more evenly across the entire intake cycle. The shorter stroke is believed to allow the induction system to more efficiently fill the cylinder, consuming less pumping work. As engine speed increases, this improved efficiency lets the engine produce higher power levels at greater speeds than a longer-stroke engine.

The accompanying **Chart-A** (drawn from a simulation based on the underlying physics) illustrates the actual differences in incremental flow between an engine with a long stroke and the same displacement engine with a "destroker" crank and a larger bore. Notice how the shorter stroke engine develops less peak flow and spreads the same total flow volume slightly more evenly across the entire intake stroke. This reduction in peak flow allows the induction system to operate more efficiently, slightly improving cylinder filling and reducing pumping work. So that would seem to prove it, right? The chart shows that a shorter stroke pumps more efficiently. While that may seem to be the case, take a look at **Chart-B**. Here's a plot of the same two engines, but this time the rod lengths have been adjusted so that they have exactly the same rod ratio, that is, the length of all connecting rods are now equal to 1.7 times the length of the strokes. With identical rod ratios, the cylinders in both engines pump exactly the same volume at each degree of crank rotation. The "proven benefits" of a short stroke disappear.

The misunderstanding that longer stroke engines are less efficient breathers probably occurs because during dyno tests of various combinations, identical rod ratios are not maintained when stroke lengths are changed. So measured differences in power are mistakenly attributed to changes in stroke, rather than the real cause: Variations in rod ratio. If you have any doubts about this, take a look at the simulation

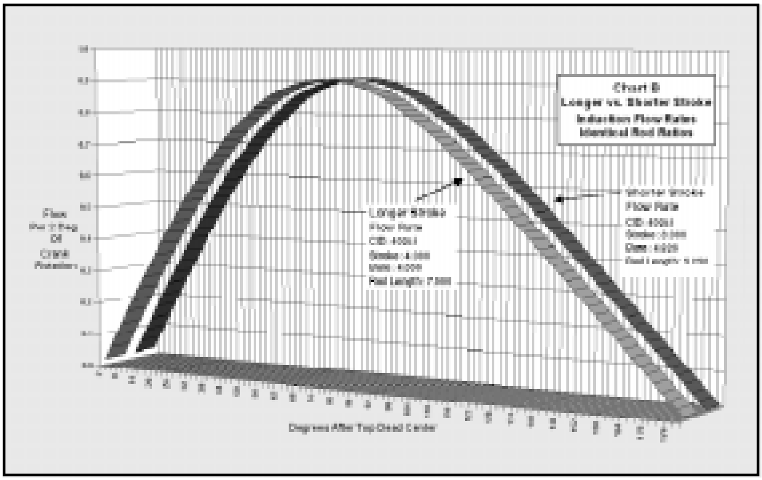
While pumping work is not directly affected by bore or stroke, larger engines, like this 600cid big block, generate tremendous pumping work at high engine speeds. The pumping losses are due to restrictions at the valves, ports and runners, and throughout the intake and exhaust systems. At some point in engine speed, there no longer exists enough <u>time</u> to move the gasses through passages of fixed sizes. When this happens, power takes a nose-dive.

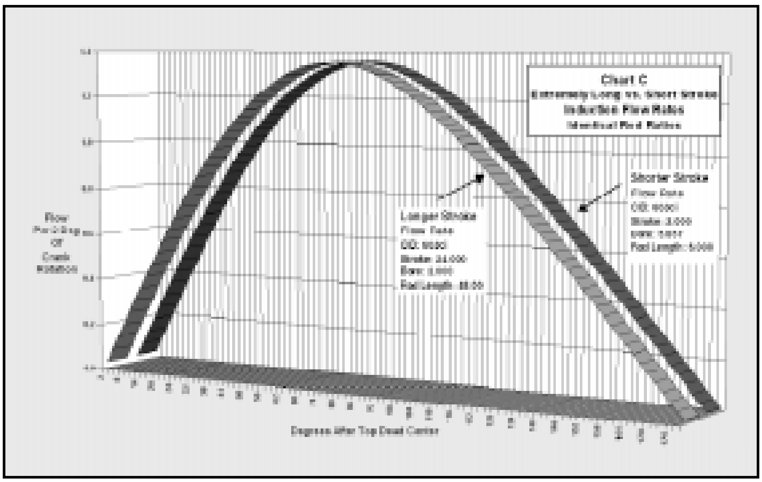




This graph illustrates the differences in flow as stroke changes. Notice how the shorter stroke engine develops less peak flow and spreads the same total flow volume slightly more evenly across the entire intake stroke.

Here's a plot of the same two engines, but this time the rod lengths have been adjusted to produce the same 1.7 rod ratio. With identical rod ratios, the cylinders in both engines pump exactly the same volume at each degree of crank rotation.





The graph shows the flow per crank degree for 3-inch stroke vs. 24-inch stroke engines. Both engines have the same 603cid and sweep out the same volume at each 2-degree increment of crank rotation throughout the entire intake stroke, despite the *21-inch difference in stroke lengths*!

depicted in **Chart-C**. Here's a test of a 3-inch stroke, 5.67-inch bore engine compared to a very long 24-inch stroke engine having a 2-inch bore. Both engines have the same total displacement of 603ci and the same rod ratios (2.0 times the stroke length). The graph shows that the pistons in both engines sweep out the same volume at each 2-degree increment of crank rotation throughout the entire intake stroke, despite the *21-inch difference in stroke lengths*!

So it's not stroke, but rather rod ratio, that has subtle, measurable effects on pumping work and peak flow rates. The longer the rod ratio, the more spread out the induction flow and, potentially, the greater the high-speed power. The shorter the rod ratio, the higher the peak flow and the earlier in the intake cycle the flow peak is reached. Considering these flow changes, short-rod combinations could show potential gains on engines that benefit from a strong carburetor signal at lower engine speeds, like short-track and street engines. But even rod ratio cannot be considered in isolation. When the rod length is changed it also affects lateral loads on the pistons, and it can change the way the pistons "rock" in their bores and the way the rings "flutter" or seal against the cylinder walls. So, depending on the lubricants used, the cylinder block and piston configurations, and several additional factors, the "theory" of rod ratio and the power changes that are predicted may or may not be found on the dyno. At least you now have an understanding of the underlying theory, even though it makes its predictions in isolation of many other interrelated variables.

While we're on the subject of rod ratio, many readers may be wondering why this variable was not included in one of the pull-down menus in Motion Software's engine simulation. There are two reasons for this: 1) As just stated, the effects of varying rod length are very subtle and often masked by other variables within the engine (such as piston side loads, borewall friction, etc.) making accurate modeling extremely difficult, and 2) The subtle changes in swept volume primarily require wave-action dynamics for rigorous analysis (a process discussed in the complete *DeskTop Dynos* book available from Motion Software). So rather than add a component that could not be modeled as accurately as the other variables in the program, rod length is not presented as a "tunable" element (although it is included within the simulation process).

Fallacy Two: Long-Stroke Torque Vs. Short-Stroke Horsepower

Another remarkably widespread belief has to do with stroke length vs. engine speed and torque vs. horsepower. A great many performance enthusiasts believe that long-stroke engines inherently develop more low-speed torque and short-stroke engines generate more high-speed horsepower. This understanding (probably as the result of reading various magazine articles over the years) is almost completely incorrect, and some enthusiasts are quite defiant when confronted with the fact that stroke, by itself, has little to do with high-speed or low-speed power potential. But, like the fallacy that short-stroke engines are more efficient air pumps, there is a thread of truth lying hidden under the surface.

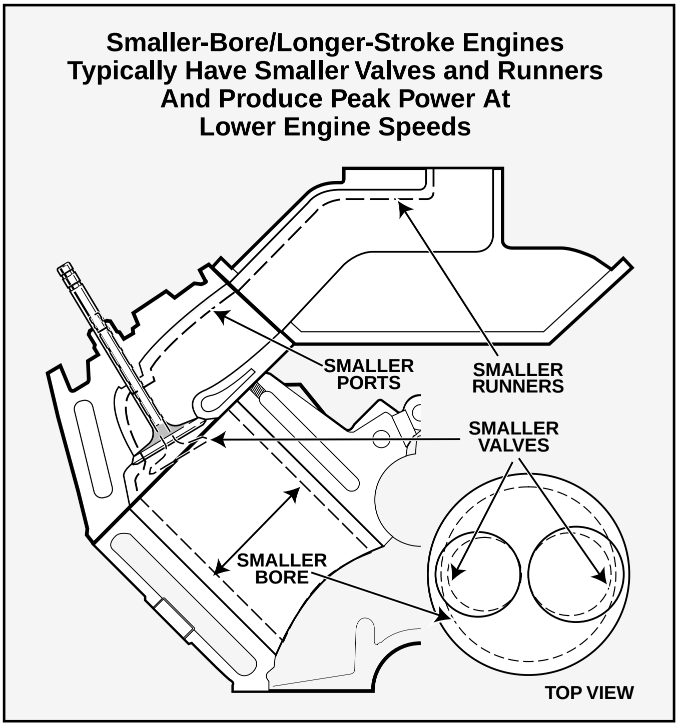
To start off, long- and short-stroke engines of equal displacement will produce the same cylinder pressures during the power stroke for the same swept volumes (assuming that they have consumed equal quantities of air and fuel; and we've already demonstrated that stroke, by itself, has almost nothing to do with induction efficiency). While the pistons in a longer-stroke engine are smaller in diameter and, therefore, experience less force from the same cylinder pressure (force on the piston is directly related to the surface area of the piston and pressure in the cylinder), the crank arms are longer and have a greater mechanical advantage. The result, believe it or not, is that *equal displacement engines of unequal stroke lengths—experiencing the same cylinder pressures—will produce the same torque at the crankshaft*. Stroke, however, is not an isolated variable; it affects the design of many other engine components. It's in this interrelatedness that we find the roots of misunderstanding about stroke vs. horsepower.

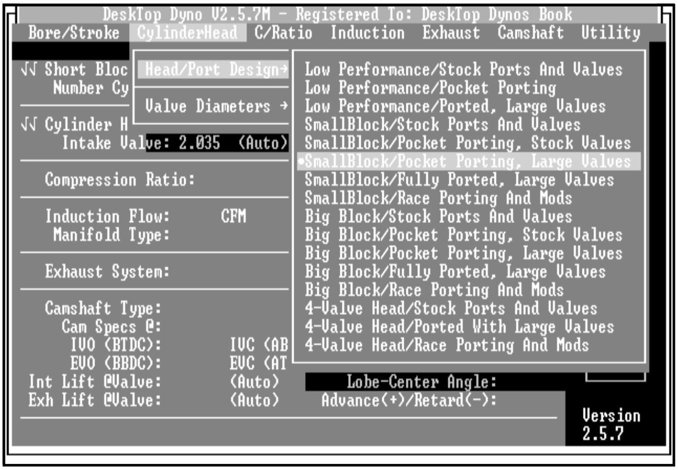
Probably the most direct reason for the belief that longer-stroke engines are lower-speed "torque generators," not capable of producing as many horsepower per cubic inch, is that many longer-stroke engines have smaller bores. An engine with smaller bores almost always has smaller combustion chambers, and smaller combustion chambers have smaller valves. So most longer-stroke engines are forced, by the design of the cylinderheads and the entire induction system, to produce less power at higher engine speeds. It's not the length of the stroke that limits power potential, its reduced flow from smaller valves, ports, and runners.

For more concrete proof, refer back to the simulation we performed in the earlier section on stroke vs. friction. In that test, we compared identical displacement engines of 603ci, one with a 6-inch stroke (dotted lines on graph) to one with a 3-inch stroke (solid lines). Both engines used the same size valves and the same 780cfm induction flow capacity. As previously demonstrated, the increase in horsepower from the shorter-stroke engine is due to a reduction in bore-wall friction, adding about 100 horsepower at 5500rpm. But look at the *shape* of the solid-line curves. They match the *shape* of the dotted-line curves for the longer-stroke engine. There is no noticeable boost in low-speed torque from the engine with a 6-inch stroke. A steadily-increasing power loss from additional cylinderwall friction is clearly visible, but notice that *the short-stroke engine produced no less torque between 2000 and 2500rpm*, a range that many believe the longer-stroke engine should easily out-torque its short-stroke counterpart.

From a "real-world" mechanical standpoint, longer-stroke engines have drawbacks that limit their high rpm potential. As can be seen in our simulation, bore-wall friction becomes a substantial power robber. Ring seal also becomes a serious problem. As the stroke increases, higher and higher piston speeds cause the rings to "flutter" against the cylinderwalls decreasing their sealing ability. It's also possible

Many people believe that longer-stroke engines are lower-speed "torque generators," not capable of producing as many horsepower per cubic inch as shorter-stroke engines of equal displacement. What is often overlooked is that most engines with smaller bores almost always have smaller combustion chambers, smaller valves, and smaller runners. It's not the length of the stroke that limits power potential, its reduced flow through the induction system.





The Head/Port Design menu, under the Cylinderhead main menu, lists general cylinderhead characteristics, including restrictive ports, typical small- and big-block ports, and even 4-valve cylinderheads. Each category includes several stages of port/valve modifications from stock to all-out race.

that at very high piston speeds during the power stroke, the piston moves so quickly that the rings can't maintain a seal between the bottom of the ring and the ring land in the piston, increasing blowby and further decreasing horsepower. The mechanical loads on the piston and rod assembly in long-stroke engines also become a serious factor. As the piston is accelerated from TDC down the bore, extremely high tension loads are imparted to the rods and pistons, and added component weight to compensate for the additional loads further increases stress.

While there are good reasons why longer-stroke engines are ill suited to high-rpm applications, don't confuse these ancillary problems with the basic design relationships between bore and stroke. And don't fall into the trap of believing, like thousands of enthusiasts, that selecting a longer stroke will automatically boost low-speed power.

THE CYLINDERHEAD AND VALVE DIAMETER MENUS

The **Cylinderhead** pull-down menu, located just to the right of the Bore/Stroke menu, contains two sub-menus that allow the simulation of various cylinderhead designs and a wide range of airflow characteristics. The first submenu, **Head/Port Design**, lists general cylinderhead characteristics, including restrictive ports, typical small- and big-block ports, and even 4-valve cylinderheads. Each category includes several stages of port/valve modifications from stock to all-out race. A selection from this menu is the first part of a two-step process that Motion simulations use to accurately model cylinderhead flow characteristics. This initial selection determines the airflow restriction generated by the ports. That is, a selection from the first submenu fixes *how much less air than the theoretical maximum peak flow will pass through each port*. What determines peak flow? That's selected from the second *Cylinderhead* submenu: **Valve Diameters**. Valve size fixes the theoretical peak flow (called *isentropic* flow—more on that later). Most cylinderheads flow only about 50% to 70% of this value. The Valve Diameter submenu allows the direct selection of valve size or **Auto Calculate Valve Size** may be chosen, directing the simulation to calculate the valve diameters based on bore size and the degree of cylinderhead porting/modifications.

While it may seem as if the various *Cylinderhead* menu choices simply refer to ranges of airflow data stored within the program, that is not the case. If each menu selection fixed the flow capacity of the cylinderheads to a specific range of values (like typical flow bench data measured at a standardized pressure drop), the simulation would be severely limited. While a simulation based on this technique might adequately calculate mass flow for engines that used nearly identical cylinderheads, accuracy would diminish rapidly for engines with even modestly larger or smaller ports and valves.

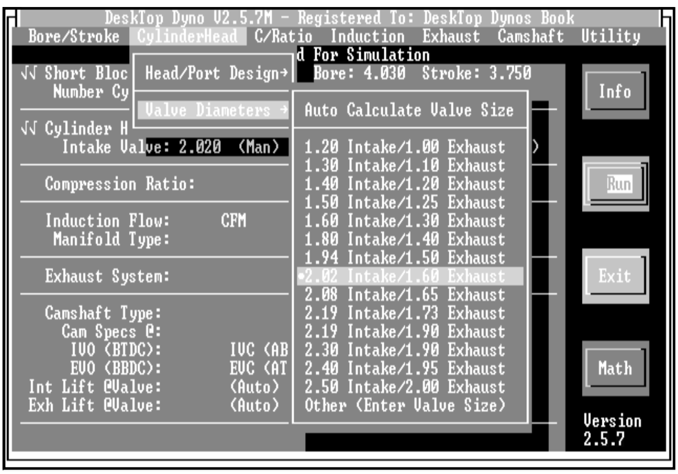
There are several reasons why the determination of port flow in sophisticated engine simulations can not be based strictly on flow-bench data. First of all, flow generated in the ports of a running engine is vastly different than the flow measured on a flow bench. Airflow on a flow bench is a steady-state, measured at a fixed pressure drop (it's also dry flow, but a discussion of that difference is beyond the scope of this book). A running engine will generate rapidly and widely varying pressures in the ports. These pressure differences directly affect—in fact, they directly cause—the flow of fuel, air, and exhaust gasses within the engine. An engine simulation program calculates these internal pressures at each crank degree of rotation throughout the four-cycle process. To determine mass flow into and out of the cylinders at any instant, you need to calculate the flow that occurs as a result of these changing pressure differences. Since the variations in pressure, or pressure drops, within the engine are almost always different than the pressure drop used on a flow bench, flow bench data cannot directly predict flow within the engine.

Before we delve deeper into the differences between flow-bench data and the mass flow calculated by an engine simulation program, let's permanently put to rest the idea of *storing* the needed airflow data within the program. Consider for a moment an engine simulation program that's based on internally-stored airflow data. If you assume that the program is capable of modeling a wide a range of engines (like Motion Software simulations), then it must also store a wide range of cylinderhead flow data. Flow would have to be recorded from a large number of engines, starting with single-cylinder motors and working up to 1000+ cubic-inch powerplants. If it were possible to accumulate this much data, there are additional shortcomings in a lookup model. Since the pressures inside the IC engine are constantly changing, a lookup program would have to contain <u>ranges of flow data</u> measured from zero to the maximum pressures differentials developed within the running engine. Even if this much data could be assembled and filed away, additional data for each engine and cylinderhead would be needed to predict the flow changes when larger valves were installed; and even more data would be needed to model port modifications. It soon becomes clear that the sheer bulk of flow data needed for a comprehensive lookup model would probably consume more space than exits on your hard drive (and that's considering that most of us now own gigabyte and larger drives)!

Cylinderhead Choices And Discharge Coefficients

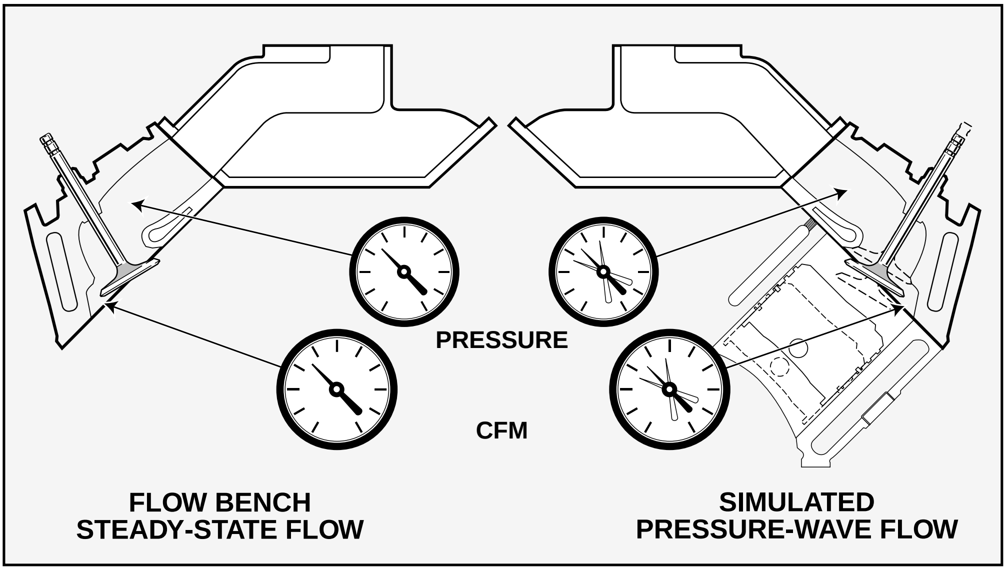
While it is impractical to <u>base</u> an engine

The Valve Diameter menu allows the direct selection of valve size or you can select *Auto Calculate Valve Size*, directing the simulation to calculate the valve diameters based on bore size and the degree of cylinderhead porting and other modifications.



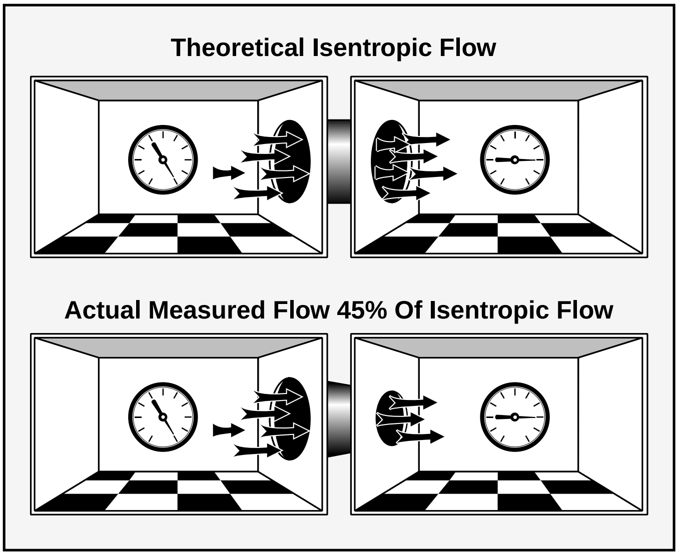
simulation on extensive flow-bench data, measured cylinderhead flow figures are, nonetheless, commonly used in sophisticated engine simulations. Rather than stored in extensive "lookup" tables within the programs, flow-bench data can be used as a means to compare the measured flow of a particular port/valve configuration against the calculated isentropic (theoretical maximum) flow. The resulting "ratio," called the **discharge coefficient**, has proven to be an effective link between flow-bench data and the simulated mass flow moving into and out of the cylinders throughout a wide range of valve openings and pressure drops. Furthermore, the discharge coefficient can be used to predict the changes in flow for larger or smaller valves and for various levels of port modifications. In other words, it's the discharge coefficient, not flow bench data, that provides a practical method of simulating mass flow within a large range of engines. Since this is such a powerful and often misunderstood concept, the following overview should prove helpful in understanding what's happening "behind the scenes" when various choices are made from the Cylinderhead pull-down menu.

The choices in the Cylinderhead menu are purposely generic. This can be frustrating if you are interested in modeling only one engine or a single engine family. In these cases it would be ideal to list the exact components you wished to test, in name or part-number order. But if you are interested in simulating different engines, including popular 4-, 6-, and 8-cylinder powerplants, the choices provided in the Cylinderhead menu (and many of the other menus) offers considerable modeling power. The menu choices move from restrictive heads to smallblock, big block, and finally to 4-valve-head configurations. If each of these choices loaded a higher absolute flow curve into the simulation, they would cover only a very narrow range of engines. Instead, each of the menu choices model an increasing flow capacity and reduced restriction. This makes it possible to accurately simulate a lawnmower engine, a high-performance motorcycle engine, an all-out Pro Stock big block, or a mild street driver, each having different port and/or



Airflow on a flow bench is steady state, but Motion's engine simulation program calculates internal pressures and mass flow at each degree of crank rotation throughout the four-cycle process. Since the variations in pressure, or pressure drops, within the engine are rarely equal to the pressure drop used on a flow bench, flow bench data cannot directly predict flow within the engine.

The absolute maximum flow rate through an orifice with no losses from turbulence, heat, or friction is called the isentropic flow. This is the flow rate that "nature" will never exceed for the given pressure drop and hole size. If we measure the actual flow through the hole and divide it by the calculated isentropic flow, we will have determined the flow efficiency or *discharge coefficient*.



valve sizes, using selections from the same menu!

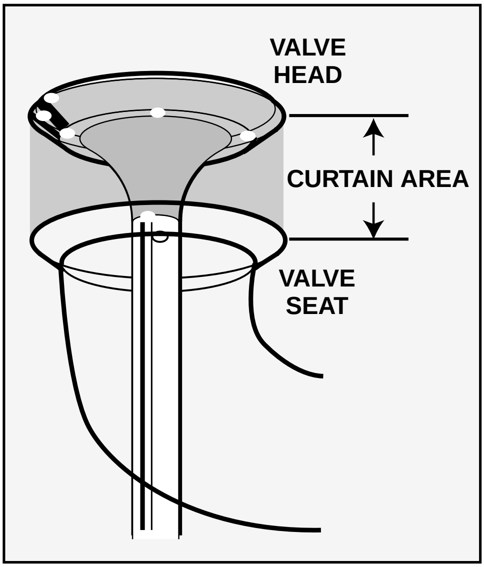
The basis for this "universality" is that each menu selection uses a different <u>discharge-coefficient curve</u> rather than airflow curve. While the discharge coefficient data is *derived* from flow-bench data, it is dimensionless (has no length, weight, mass, or time units) and is applicable to any cylinderhead of similar flow efficiency. To clarify this concept, picture two large rooms connected by a hole in the adjoining wall. When pressure is reduced in one room, air will flow through the hole at a specific rate. It is possible to calculate what the flow would be if there were no losses from heat, turbulence, etc. This flow rate, called the isentropic flow, is never found in the real world, but is, nevertheless, a very useful term. It's the rate of flow that "nature" will never exceed for the given pressure drop and hole size. If we measure the actual flow through the hole and divide it by the calculated isentropic flow, we will have determined the flow efficiency or *discharge coefficient* of the hole:

Discharge Coefficient (always less than 1)
= Measured Flow / Calculated Flow

Since every orifice has some associated losses, the discharge coefficient is always less than 1. If the calculated discharge coefficient for our hole in the wall was 0.450, this would mean that the hole flows 45% as much air as an ideal hole. In other words, it's 45% efficient.

Let's see how this concept can be applied to cylinderheads. When an intake valve of a specific size is opened a fixed amount, it exposes a flow path for air called the *curtain area*. Through this open area, measured in square inches just like the hole in the wall, air/fuel mixture moves at a specific rate depending the pressure drop across the valve (as you recall, the pressures are calculated by the simulation software at each degree of crank rotation). If the valve and port were capable of perfect isentropic flow, the simulation equations could calculate the precise mass flow that entered the cylinder during this moment in time. But real cylinderheads and valves are far from perfect, and it's flow bench data that "tells" the simulation how far from perfect the real parts perform. By dividing the isentropic flow by the flow-bench data (both at the same pressure drop) of a comparable head/port configuration at each increment of valve lift, *the simulation software creates a discharge coefficient curve that it can use as a correction factor for port flow at all other pressure drop levels*.

The true power of this method lies in the fact that the cylinderhead tested on the flow bench—used to develop the discharge-



When a valve of a specific size is opened a fixed amount, it exposes a flow path called the *curtain area*. Through this open area (measured in square inches) gasses move at a specific rate depending on the pressure drop across the valve.

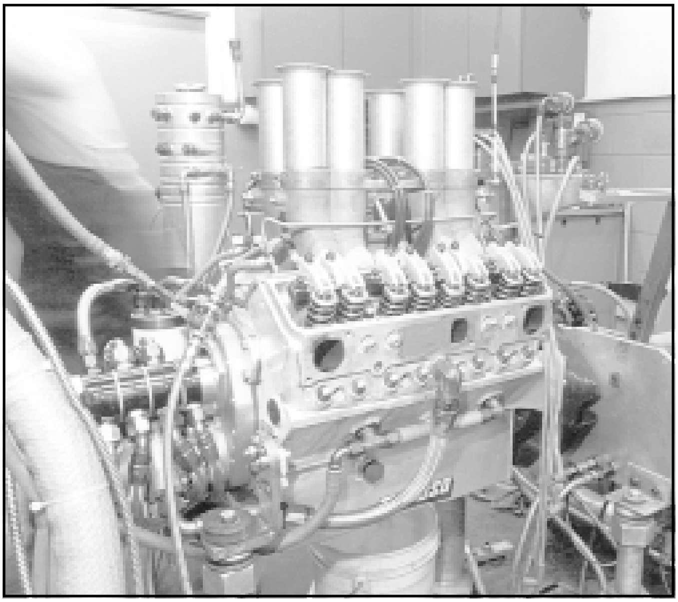
coefficient curve—may have been designed for an entirely different engine and used different valve sizes. Nevertheless, as long as the head configuration matches the simulated cylinderhead, in other words, the ports have similar flow efficiency, the discharge-coefficient curve will closely adjust isentropic flow to real-world corrected flow. So a specific choice in the Cylinderhead menu, say "Pocket Porting With Large Valves" may accurately model a factory performance head for a smallblock Chevy, a cylinderhead on a 4-cylinder Toyota engine, or even a motorcycle head. <u>With discharge coefficient corrections, it's not the absolute flow numbers that are important, it's how well the valve and port flow for their size that really counts.</u>

As you made selections from the Cylinderhead menu, you may have wondered whether the flow data used to "correct" the isentropic flow in any of the menu items would match the published flow of cylinderheads that you would like to simulate. Considering what you now know about the relationships between cylinderheads of similar efficiencies and how flow data is used by the program, it becomes clear that knowing the flow-bench data used in the program would probably not be helpful. The flow may not have been obtained from the same valve sizes or even the same brand cylinderheads. On the other hand, it is entirely feasible for the simulation to substitute user-entered flow data from tests conducted on a specific set of cylinderheads. This data could then be used to develop custom discharge-coefficient curves that would, in turn, closely model cylinderheads. While this is not currently supported, it will be incorporated in upcoming versions of Motion simulation software (if you haven't already, send in your registration card; you'll receive information on upgrades and new releases).

Ram Tuning And Pressure Waves

Up to this point our discussion has centered around airflow, valve size, and discharge coefficients. The assumption has been that reducing restriction and increasing flow efficiency will allow more air/fuel mixture to enter the cylinders and produce more horsepower. Initially this is true, but when port and valve sizes are increased, power begins to fall off at lower speeds, then at higher speeds, and finally very large passages reduce power throughout the entire rpm range! At first, this may seem quite mysterious. If minimizing restriction was the key to improved airflow and power, this phenomenon would indeed be inexplicable. But as we've discovered, the IC engine does not function by simply directing the flow of air and fuel as a hose pipe directs the flow of water. Powerful wave dynamics take control of how air, fuel, and exhaust gasses move within the inlet and exhaust passages. Because of these phenomena, there is an optimum size for the ports and valves for any specific application, from street economy to all-out drag racing.

Consider what happens after the intake valve opens and the piston begins moving



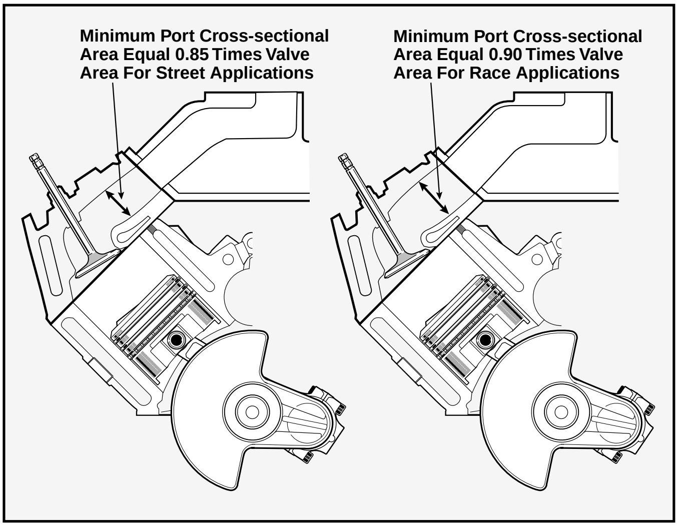
When the piston moves from TDC to BDC on the intake stroke, "the bigger the better" is the rule for ports and valves. After the piston reaches BDC and begins to travel back up the bore, it tries to push charge back out of the cylinder. Now the rule of thumb for power is "smaller ports and high airflow speeds." A compromise must be found. This race engine finds that critical balance with large runners that generate 700-ft/sec peak port velocity at very high engine speeds.

down the bore on the intake stroke. As pressure drops, air/fuel mixture enters the cylinder. During this portion of the induction cycle, any restriction to inlet flow reduces cylinder filling and power output. So, while the piston moves from TDC to BDC, the rule of thumb for the ports and valves is "the bigger the better." After the piston reaches BDC and begins to travel back up the bore on the early part of the compression stroke, the intake valve remains open and the cylinder continues to fill with air/fuel mixture. This "supercharging" effect, caused by the momentum built up in the moving column of air and fuel in the ports and inlet runners, adds considerable charge to the cylinder and boosts engine output. However, as soon as the piston begins to move up the bore from BDC, it tries to push charge back out of the cylinder. Larger ports and valves not only offer low restriction to incoming charge, but they make it easier for the piston to reverse the flow and push charge back out of the cylinder. Furthermore, a low restriction induction system has a large cross-sectional area and allows the incoming charge to move more slowly (in feet per second), so the charge carries less momentum and, once again, is more easily forced back out of the cylinder. So the rule of thumb to optimize power for the period of time between BDC and intake-valve closing is "smaller ports that produce higher airflow speeds are better." Since it's not possible to rapidly change the size of the ports as the engine is running, a compromise must be found that minimizes restriction and optimizes charge momentum.

As it turns out, optimum port and valve sizes for performance applications at a specific engine speed must be small enough to allow the charge to reach speeds of about 700 feet per second during the intake stroke. Then, as the piston moves from BDC to the intake valve closing point, the momentum generated by these speeds continues to force air and fuel into the cylinder, optimizing charge density. Typically, the best port size and cam timing for performance allows a slight "reversion" of fresh charge just before the intake valve closes.

Unfortunately, the smaller port cross-sectional areas required to generate optimum flow velocities create a restriction to airflow and increase pumping work. If port cross-sectional areas were smaller and flow velocities increased much beyond 700 feet/second, the added restriction and increased pumping work would reduce overall cylinder filling and engine output would suffer.

Despite the interrelatedness of engine speed, cam timing, exhaust system configuration, and more, there are some "rules of thumb" in selecting a workable port for common applications. Cylinderheads will probably provide good performance if the area of minimum port cross-section is equal to 0.9 times the valve area for race applications and 0.85 for street use.

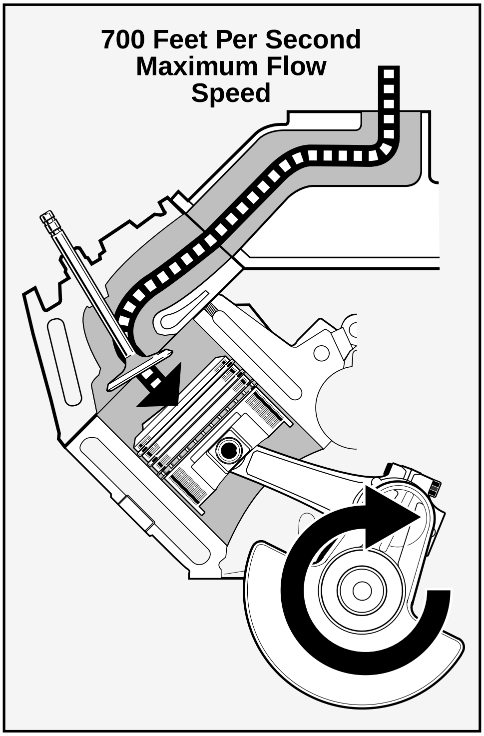


The "magic" balance found at about 700 feet per second between charge momentum, inlet restriction, and pumping work allows the cylinders to fill with the greatest mass of air/fuel mixture throughout the complete induction cycle from IVO (intake valve opening) to IVC (intake valve closing).

What is the best cross-sectional area for any specific engine; the area that generates about 700 feet/second peak flow velocities? That is a very tough question to answer. In fact, optimum port shapes and cross-sectional areas are so interrelated with other engine variables, like engine displacement and rpm, cam timing, exhaust system configuration, intake manifold design, compression ratio, and more, that engine simulation software is needed to sort through the complexity and find the "magic" combinations. Furthermore, a full wave-action modeling program, like *Dynomation* discussed in the complete *DeskTop Dynos* book, is required for this task since it's the dynamics of finite-amplitude wave motion that are responsible for the recorded changes in horsepower.

Despite these complexities, there are some "rules of thumb" that can be helpful in selecting a workable port for common

The best cross-sectional area for any specific engine, one that generates about 700 feet/second peak flow velocity, is difficult to find. Optimum port shapes and cross-sectional areas are so interrelated with engine displacement and rpm, cam timing, exhaust system configuration, intake manifold design, compression ratio, and more, that engine simulation software is needed to sort through the complexity and find the "magic" combinations.

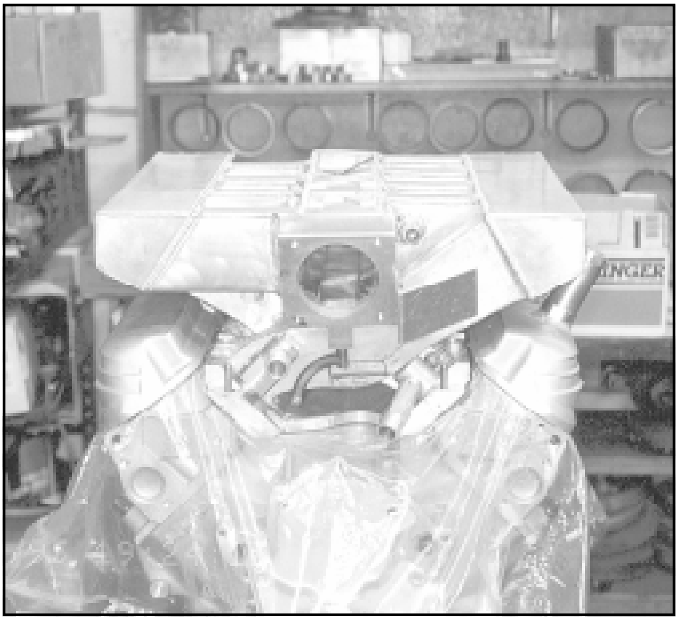


applications. If the area of minimum cross section in the ports is equal to 0.9 times the areas of the valves for race application and 0.85 times the valve areas for street use, the heads will probably provide good performance. This rule applies to typical automotive engines, not to high-rpm motorcycles or low-rpm aircraft. However, for typical powerbands between 5000 and 8500rpm, this rule should be reasonably close, providing the valve diameters are not too small or too large for the application.

These same relationships between port cross-sectional areas and valve diameters are used in Motion Software's Filling And Emptying simulation. As you move down the Cylinderhead menu, choices within each section—say within the smallblock or big block categories—the minimum port cross-section increases from 0.85 to 0.9 times the selected valve areas. However, since the current simulation does not incorporate as robust a wave-action analysis as the Dynomation program (discussed in the complete DeskTop Dynos book), port cross-sectional areas are not tunable by the user.

The phenomenon of cylinder filling by charge momentum is often called **ram tuning**, and is commonly thought to be an independent property of induction tuning, separate from the complex theory of wave dynamics that most professional racers only vaguely recognize. Because the concept of ram tuning is easy to grasp, it has been widely "applied" for many years. Long injector stacks, extremely large ports and valves, and other measures have been used to facilitate the free flow of additional charge into the cylinder after BDC on the intake stroke. But now that we have uncovered the fine balance needed between port cross-sectional area, charge momentum, airflow velocity, pumping work, cam timing, and engine speed, it is more obvious that large ports and valves, by themselves, are not the answer. Unfortunately, the insatiable desire to have large ports and valves seems to drive the cylinderhead market. Most customers simply want larger ports and valves. So head manufacturers and porters turn out droves of heads with ports that are too large for the application. While they reduce restriction during the intake stroke, low charge momentum during the majority of the rpm range of the engine reduces overall cylinder filling and power output.

One important thing to learn from this book should be to permanently discard the notion that "bigger is better" when it comes



The phenomenon of cylinder filling by charge momentum is often called ram tuning, and optimizing this effect means finding a balance between port cross-sectional area, airflow velocity, pumping work, cam timing, and engine speed. This custom manifold was designed—with the help of simulations, including Dynomation (discussed in the complete DeskTop Dynos book) to find that illusive balance on smallblock Fords.

The "Low Performance" cylinderhead choices are intended to model cylinderheads that have unusually small ports and valves. Heads of this type were often designed for low-speed, economy applications, with little concern for high-speed performance. Early 260 and 289 smallblock Ford and to a lessor degree the early smallblock Chevy castings fall into this category.

to ports and valves. The right combination is the key.

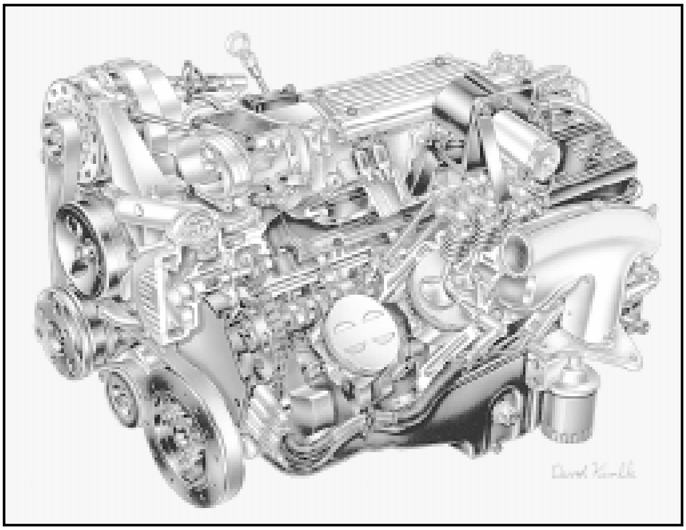
Sorting Out Cylinderhead Menu Choices

Now that we have covered some of the basic theory behind the choices in the Cylinderhead menus, here's some practical advice that may help you determine the appropriate selections for your application.

Low Performance Cylinderheads— There are three "Low Performance" cylinderhead selections listed at the top of the Head/Port Design submenu. Each of these choices is intended to model cylinderheads that have unusually small ports and valves relative to engine displacement. Heads of this type were often designed for low-speed, economy applications, with little concern for high-speed performance. Early 260 and 289 smallblock Ford and to a lessor degree the early smallblock Chevy castings fall into this category. These choices use the lowest discharge coefficient of all the head configurations listed in the menu. Minimum port cross-sectional areas are 85% of the valve areas or somewhat smaller and, if Auto Calculate Valve Size has been selected (more on this feature in the next section), relatively small (compared to the bore diameter) intake and exhaust valve diameters will be used.

The first low-performance choice models an unmodified production casting. The second choice "Low Performance/Pocket Porting" adds minor porting work performed below the valve seat and in the "bowl" area under the valve head. The port runners are not modified. The final choice "Low Performance/Ported, Large Valves" incorporates the same modifications and includes slightly larger intake and exhaust valves. Valve size increases vary, but they are always scaled to a size that will generally install in production castings without extensive modifications.

The low-performance choices have some ability to model flathead (L-head & H-head) and hybrid (F-head) engines. While the ports in these engines are considerably more restrictive than early Ford smallblock engines, by choosing Low-Performance and manually entering the exact valve sizes, the simulation will, at least, give you an approximate power output that you can use to evaluate changes in cam timing, induction flow, and other variables. Also, it's not essential that your model produce an absolutely accurate horsepower



The "Smallblock Cylinderhead" menu choices model cylinderheads that have ports and valves sized with performance in mind, like the heads on this LT1. The stock selections are not excessively restrictive for high-speed operation, and overall port and valve-pocket design offers a good compromise between low restriction and high flow velocity.

number. A great deal of useful information can be found by simply looking at the *changes in power* that result from various combinations of parts.

Smallblock Cylinderheads—The smallblock and big-block choices comprise the two main cylinderhead categories in the Head/Port Design submenu. Choices from these two groups apply to over 90% of all performance engine applications from mild street use to all-out competition.

The basic smallblock selections model heads that have ports and valves sized with performance in mind. Ports are not excessively restrictive for high-speed operation, and overall port and valve-pocket design offers a good compromise between low restriction and high flow velocity. The stock and pocket-ported choices are suitable for high-performance street to modest racing applications.

The next step is "SmallBlock/Fully Ported, Large Valves" and this cylinderhead moves away from street applications. This casting has improved discharge coefficients, greater port cross-sectional areas, and increased valve sizes. Consider this head to be an extensively modified, high-performance, factory-type casting. It does not incorporate "exotic" modifications, like raised and/or welded ports that require custom-fabricated manifolds. "SmallBlock/Fully Ported, Large Valves" heads are high-performance castings that have additional modifications to provide optimum flow for racing applications.

The last choice in the smallblock group is "SmallBlock/Race Porting And Mods." This selection is designed to model state-of-the-art, high-dollar, Pro-Stock type cylinderheads. These custom pieces are designed for one thing: Maximum power. They require hand-fabricated intake manifolds, have excellent valve discharge coefficients, and the ports have the largest cross-sectional areas in the smallblock group. This head develops sufficient airflow speeds for good cylinder filling only at high engine rpm.

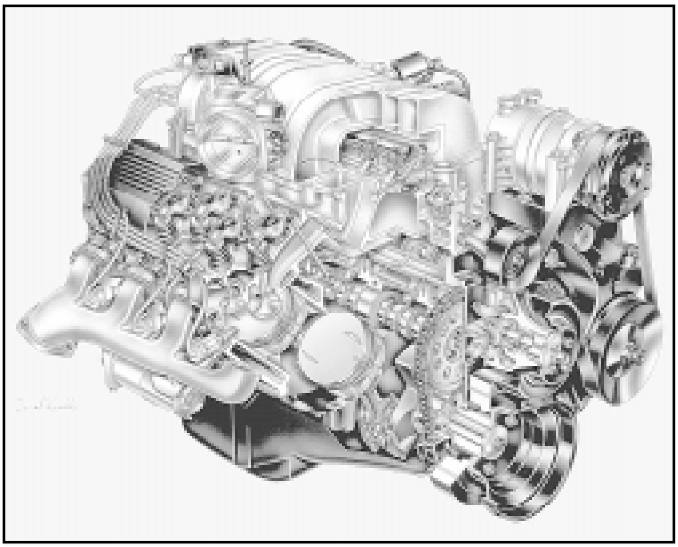
Big Block Cylinderheads—All big-block selections are modeled around heads with "canted" valves. That is, the valve stems are tilted toward the ports to improve the discharge coefficient and overall airflow. All ports have generous cross-sectional areas for excellent high-speed performance.

The first three choices are based on an <u>oval-port</u> design. These smaller cross-sectional area ports provide a good compromise between low restriction and high flow velocity for larger displacement engines. The stock and pocket-ported selections are suitable for high-performance street to modest racing applications.

The final two selections simulate extensively modified rectangular-port heads. These choices are principally all-out, big-block Chevy heads, however, they closely model other extremely aggressive high-performance racing designs, like the Chrysler Hemi head. As with the smallblock category, the "Big Block/Fully Ported, Large Valves" heads are not suitable for most street applications. These castings have high discharge coefficients, large port cross-sectional areas, and increased valve sizes. This head is basically a factory-type casting but extensively improved. However, it does not incorporate "exotic" modifications, like raised and/or welded ports that require custom-fabricated manifolds.

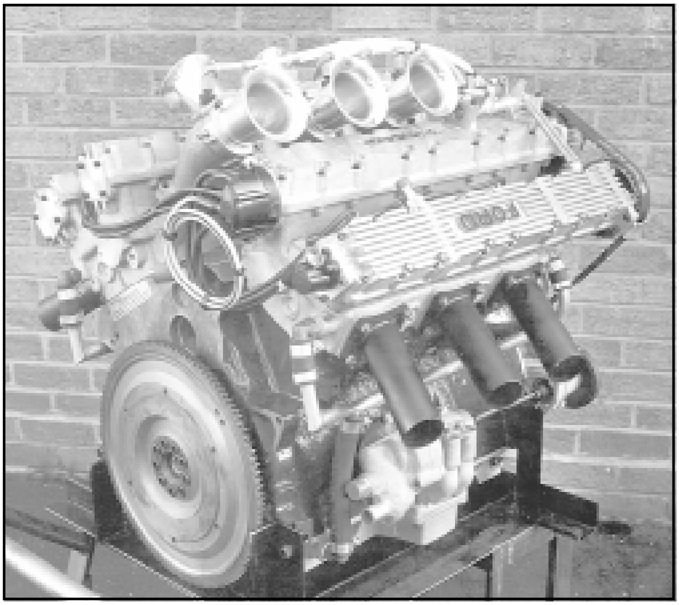
The last choice in the big-block group is "Big Block/Race Porting And Mods." This selection is designed to model state-of-the-art, high-dollar, Pro-Stock cylinderheads. These custom pieces, like their smallblock counterparts, are designed for maximum power. They require hand-fabricated intake manifolds, have optimum valve discharge coefficients, and the ports have the largest cross-sectional areas in the entire Head/Port Design submenu, except for 4-valve heads (discussed next). These specially fabricated cylinderheads only develop sufficient airflow for good cylinder filling with large displacement engines at very high engine speeds.

4-Valve Cylinderheads—The last three selections in the Head/Port Design submenu model 4-valve cylinderheads. These are very interesting choices since they simulate the effects of very low-restriction ports and valves in stock and performance applications. The individual ports in 4-valve heads begin as single, large openings, then neck down to two Siamesed ports, each having a small (relatively) valve at the combustion chamber interface. Since there are two intake and two exhaust valves per cylinder, valve curtain area is considerably larger than with the largest single-valve-per-port designs. In fact, 4-valve heads can offer more than 1.5 times the curtain area of the largest 2-valve heads. This large area, combined with high-flow, low-restriction ports greatly improves air and fuel flow into the cylinders at high engine speeds. Unfortunately, the ports offer an equally low restriction to reverse flow (reversion) that occurs at low engine speeds when the piston moves up the cylinder from BDC to Intake Valve Closing (IVC) on the final portion of the intake stroke. For this reason, 4-valve heads, even when fitted with more conservative ports and valves, can be a poor choice for small-displacement, low-speed engines. On the other hand, the outstanding flow characteristics of the 4-valve head put it in an-



The "Big-Block Cylinderhead" selections are modeled around heads with canted valves. Ports have generous cross-sectional areas. The first three menu choices model oval-port designs. The final two selections simulate modified rectangular-port heads. The appropriate selection for this L29 big-block Chevy would probably be the second or third menu choice—the fourth menu choice models a head with flow capacity beyond the stock L29.

The "4-Valve Cylinderhead" menu selections model cylinderheads with 4-valves per cylinder. These heads can offer more than 1.5 times the curtain area of the largest 2-valve heads. This large valve area, combined with high-flow, low-restriction ports greatly improves air and fuel flow into the cylinders at high engine speeds. These Cosworth heads were designed for the English Ford V6. When they were raced in England several years ago, they regularly beat Chevy V8s.



other "league" when it comes to high horsepower potential on large engines at high engine speeds.

The first choice in the 4-valve group is "4-Valve Head/Stock Ports And Valves." This simulates a 4-valve cylinderhead that would be "standard equipment" on a factory high-performance or sports-car engine. These "mild" heads offer power comparable to high-performance 2-valve castings equipped with large valves and pocket porting. However, because they still have relatively small ports, reasonably high port velocities, and good low-lift flow characteristics, they often show a boost in low-speed power over comparable 2-valve heads.

The next choice, "4-Valve Head/Ported With Large Valves" represents a mild rework of "stock type" 4-valve heads. Larger valves have been installed and both the intake and exhaust flow has been improved by pocket porting. However, care has been taken not to increase the minimum cross-sectional area of the ports. These changes provide a significant increase in power with only slightly slower port velocities. Reversion has increased, but overall, these heads should show a power increase throughout the rpm range on medium to large displacement engines.

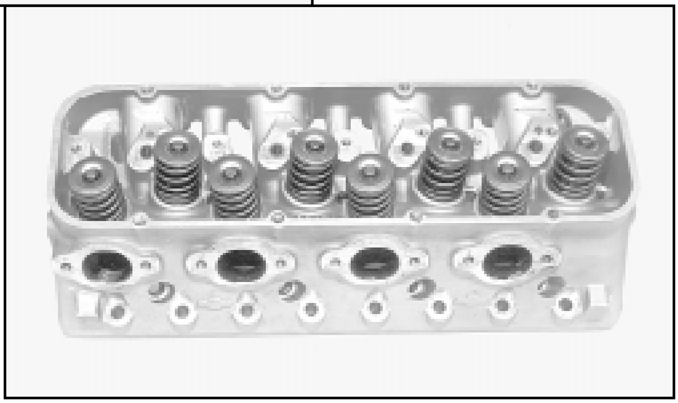
The final choice, "4-Valve Head/Race Porting And Mods," like the other "Race Porting And Mod" choices in the Head/Port Design submenu, models an all-out racing cylinderhead. This selection has the greatest power potential of all. The ports are considerably larger than the other choices, the valves are larger, and the discharge coefficients are the highest possible. These heads suffer from the greatest reversion effects, especially at lower engine speeds on small displacement engines. (These heads, like all head choices, are "scaled" to engine size, so that smaller engines automatically use appropriately smaller valves—providing the Auto Calculate Valve Size option is selected—and smaller ports.) If you would like to know what "hidden" power is possible using any particular engine combination, try this cylinderhead choice. It is safe to say that the only way to find more power, with everything else being equal, would be to add a supercharger, nitrous-oxide injection, or use exotic fuels.

When To Choose Smallblock Or Big-Block Cylinderheads

Making appropriate Head/Port Design choices may be easier now that you are aware of some of the less-obvious issues.

This smallblock Chevy head from Brodix incorporates canted valves and a spread-port intake design, meeting the criterion for a "big-block" for simulation purposes. The second or third menu choices (oval-port) are probably the best models for these heads.

This is another example of a smallblock Chevy with canted valves and spread-port intakes. Developed for the limited IMSA GTP racing program in the early '90s, these Chevrolet splayed-valve heads should be correctly modeled by the third or fourth big-block menu choice.



However, there are additional aspects of menu selections that should be emphasized to avoid confusion when modeling specific cylinderheads. Perhaps the most "confusing" issue surrounds the selection of heads from the smallblock group for engines that are commonly recognized as big blocks.

Many big-block engines use cylinderheads that are simply "scaled-up" smallblock designs. The valve and port sizes on these heads may be 10 to 20% larger than their smallblock counterparts, but that's about the only difference. The ports remain a tall, rectangular shape, valve stem angles relative to the port centerline are identical or nearly identical, and combustion chamber shapes are often the same. When proportionally larger heads are installed on engines of proportionally larger displacement, overall flow efficiency remains very similar. So for engines like the big-block Chrysler wedge, Olds, Pontiac and several Ford big blocks, plus many other engines, a selection from the smallblock group, combined with entering (or having the program calculate) larger valve diameters, will closely simulate the power levels of these "big-block" engines.

Some big blocks have cylinderheads of substantially different design. These "true" big block heads are built for higher performance levels and have visibly different ports, valves, and/or combustion chambers. Big-block Chevy heads fall into this category with two common configurations. The first is a milder, "street," oval-port version and the second uses larger rectangular-ports. The larger head is, without question, a high-performance, high-rpm design, even on large-displacement engines. Both of these heads "cant" the valve stems toward the centerline of the port. This shifted valve position improves the discharge coefficient and allows for slightly larger valves.

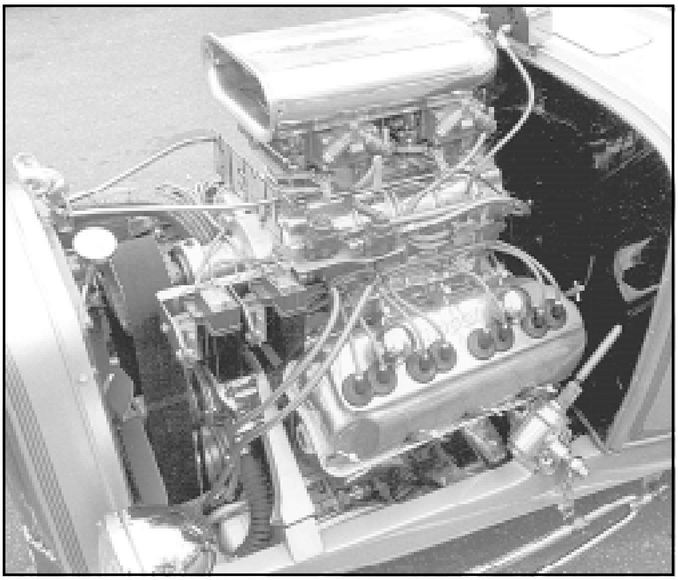
This same design philosophy is used in Chrysler's Hemi heads. An all-out racing design, the Hemi "cants" the valves even further and uses combustion chambers shaped in a true hemisphere. The ports have large cross-sectional areas with high discharge coefficients. These heads, like all big-block heads with the most "aggressive" designs, come up short on low-speed performance because of poor cylinder filling and high reversion, but produce exceptional horsepower at high engine speeds on large displacement engines.

Finally, there are engines that have smallblock displacements but use cylinderheads that look like they came straight off of a big block. Many 4- and 6-cylinder "sports-car" engines fall into this category, using single- or double-overhead cams, canted valves, and even "hemi" combustion chambers. More rarely, smallblock Chevys even turn up with big-block heads. Chevrolet's maximum performance "splayed-valve" heads for the smallblock Chevy IMSA racing program were released in the early nineties. These cylinderheads are simply a "mini" big-block design, and modeling of this head should be done using the big-block choices in the Head/Port Design submenu.

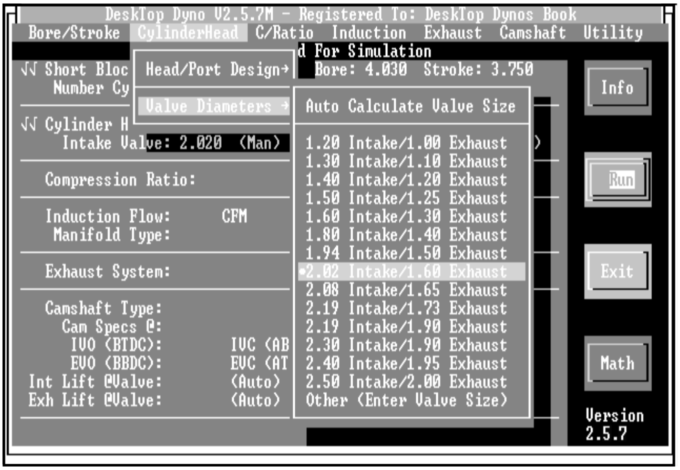
Valve Diameters And The "Auto Calculate" Feature

The **Valve Diameters** submenu is located in the lower half of the CylinderHead menu. The first selection is "Auto Calculate Valve Size." This powerful feature instructs the simulation software to determine the most likely valve sizes to be used with the current engine based on an assessment of the bore diameter and the Head/Port Design selection. The Auto-Calculate function is active when selected and by default when the simulation program is started or whenever "Clear Current Component Selections" is chosen from the **Utility** menu. Auto Calculate is especially helpful if you are experimenting with a several different bore and stroke combinations or comparing different engine configurations. Auto Calculate will always select valves of appropriate size for the cylinderheads under test and it will never use valve sizes that are too large for the current bore diameter. If you are using version 2.5 or greater of Motion Software's simulation, the program will display the calculated valve diameters in the Component Selection Box along with "(Auto)," indicating that the valve sizes were automatically determined. Earlier versions simply displayed "(Auto)"; calculated sizes were not shown.

While the Auto-Calculate feature is helpful during fast back-to-back testing, it may not "guess" the precise valve sizes used, and therefore, not simulate power levels as accurately as possible. In these situa-



The last two choices in the big-block menu will also model Chrysler's 426-Hemi cylinderhead. The last menu choice will simulate all-out ProStock configurations.



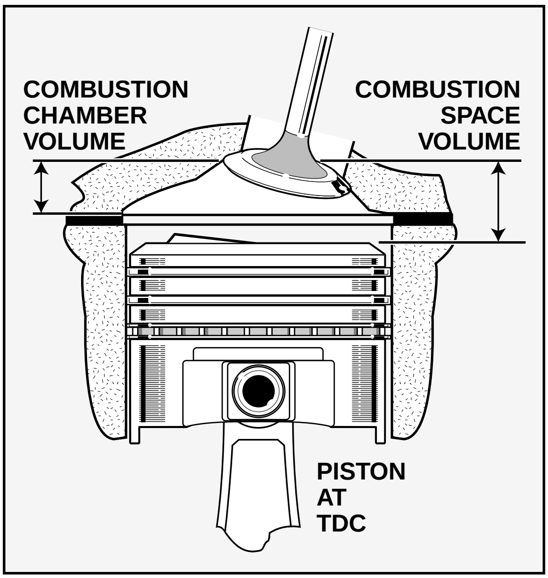
Auto Calculate will always select valves of appropriate size for the cylinderheads under test. While this feature is helpful during fast back-to-back testing, it may not "guess" the precise valve sizes. In these situations refer the list of exact valve sizes, or you can choose "Other..." from the bottom of the menu and directly enter valve diameters.

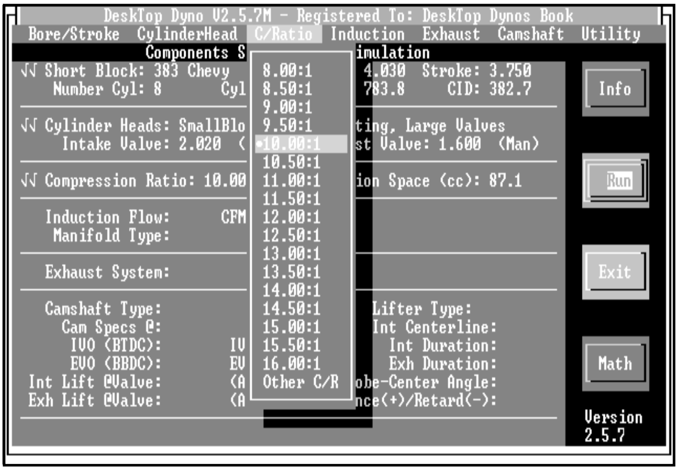
tions refer to the second part of the Valve Diameters menu. Here you will find a list of exact valve sizes consisting of common intake and exhaust combinations, or you can choose "Other..." from the bottom of the menu and directly enter any valve diameters within the acceptable range of the program (approximately 0.800- to 2.75-inches). When exact valve sizes are selected by either of these methods, the diameters are displayed in the Component Selection Box along with "(Man)," indicating that the sizes were manually entered. When "(Man)" is active, the program disables auto-calculation of valve diameters and will not automatically change the displayed values, regardless of the cylinderheads or cylinder bore diameters chosen for the test engine. You can change valve diameters at any time by simply choosing different sizes from the menu or by selecting "Other..." again. You can also re-enable the Auto Calculate feature at any time by re-selecting it from the Valve Diameter submenu.

THE COMPRESSION RATIO MENU

C/Ratio is the third pull-down menu located in the main menu bar. A selection

While combustion-chamber volume is simply the volume in the cylinderhead, the combustion-space volume is the total enclosed volume when the piston is located at TDC. This space includes the volume in the combustion chamber, plus any volume added by the piston not rising to the top of the bore, less any volume due to the piston or piston dome protruding above the top of the bore.





The *C/Ratio* establishes the compression ratio for the simulated engine. The compression ratio is a comparison of the geometric volume that exists in the cylinder when the piston is located at BDC (bottom dead center) to the "compressed" volume when the piston reaches TDC (top dead center). Changing compression ratio has a pronounced effect on engine power.

from this menu establishes the compression ratio for the simulated engine from 8:1 to 16:1 (version 2.5.7—under development when this book was published—supports a compression ratio range of 6:1 to 18:1 using the "Other" selection). The compression ratio is a comparison of the geometric volume that exists in the cylinder when the piston is located at BDC (bottom dead center) to the "compressed" volume when the piston reaches TDC (top dead center). As you recall, compression ratio is calculated with the following formula:

$$\textbf{Compression Ratio} = \frac{\textbf{Swept Cyl Vol + Combustion Space Vol}}{\textbf{Combustion Space Vol}}$$

Let's take a close look at this relationship to discover exactly what compression ratio is and how compression ratio works inside the IC engine.

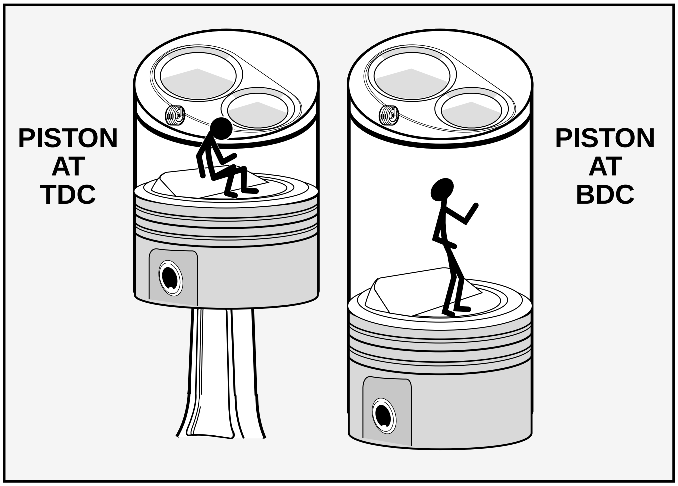
Compression Ratio Basics

The above compression-ratio equation contains two variables: 1) swept cylinder volume, and 2) combustion space volume. These volumes are the only two variables that affect compression ratio. However, each of these variables is made up of multiple elements, so the first step in exploring compression ratio must be to "dissect" these variables.

Swept cylinder volume is the most straightforward of the two. As you discovered in the previous **Bore/Stroke** section of this guide, swept volume is calculated by Motion's Filling And Emptying simulation—and displayed in the Component Selection Box—as soon as the bore and stroke have been selected for the test engine. Swept volume is simply the three-dimensional space displaced by the piston as it "sweeps" from BDC to TDC, and is determined solely by the bore diameter and stroke length.

The other variable in the compression-ratio equation is combustion-space volume. This is the total volume that exists in the cylinder when the piston is located at TDC. This space includes the volume in the combustion chamber, plus any volume added by the piston not rising fully to the top of the bore, less any volume due to the piston protruding above the top of the bore. The complexity involved in combustion-space volume can be a stumbling block for some enthusiasts. You may find it helpful to refer to the accompanying drawings for illustrations of these concepts.

A good way to visualize these variables is to imagine yourself a "little man" wandering around inside the engine. Let's take a walk inside the combustion space. Picture what it would look like in the cylinder with the piston at TDC. You would see the combustion chamber above you like a ceiling. Your floor would be the top of the piston. If the piston (at TDC) didn't rise completely to the top of the cylinder, around



Picture yourself as a "little man" inside the cylinder. With the piston at BDC, the very same volumes that exist at TDC are still there, but now the swept-volume of the cylinder has been added. The ratio between these two volumes is the compression ratio.

the edges of the floor you would see a bit of the cylinderwall, with the head gasket sandwiched between the head and block, like a low molding around the room. There may be notches in the top of the piston just under your feet (don't trip!). If the piston had a dome, it might look like a small retaining wall rising from the floor, to perhaps knee high. The combustion space would be larger if the piston was positioned lower down the bore or if the notches under your feet were deeper, and it would be smaller if the retaining wall (dome) volume was larger. This entire space is "home" for the compressed charge when the piston reaches TDC. This is the volume that makes up the denominator of the compression-ratio calculation. Now let's continue our "ride" in the cylinder, but this time picture what it looks like when the piston is positioned at BDC. The very same volumes that we just described (chamber, dome, notches, gasket, etc.) are still there, but are now located well above our head. It looks like the room has been stretched, like the elevator ride in the Haunted House at Disneyland. This "stretched" volume is what is described in the numerator of the equation. It's simply the original combustion volume plus the volume added by the "sweep" of the piston as it traveled from TDC to BDC. The ratio between these two volumes is the compression ratio.

Changing Compression Ratio

A quick look at the compression-ratio equation reveals that if engine displacement (swept volume) is increased, either by increasing the bore or stroke, the compression ratio will rise. In fact, with everything else being equal, a longer stroke will increase compression ratio much more effectively than increasing bore diameter. This is due to the fact that a longer stroke not only increases displacement, but it tends to <u>decrease</u> combustion space volume since the piston moves higher the bore (in our "little man" example, raising the floor closer to the ceiling). This "double positive" effect results in rapid increases in compression ratio for small increases in stroke length. On the other hand, increasing cylinder bore diameter also increases compression ratio but much less effectively. This is caused by the <u>increase</u> in combustion volume that accompanies a larger bore (again, using our "little man," a larger bore adds more floor space by increasing the diameter of the room), partially offsetting the compression-ratio increase that occurs from increasing cylinder displacement.

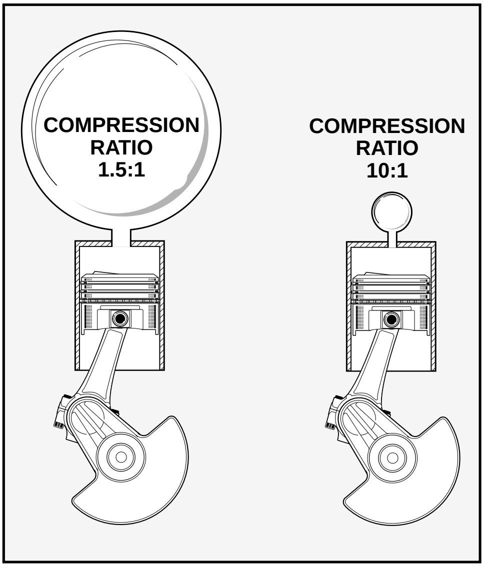
Changing combustion space, the other element in the equation, will also alter the compression ratio. Anything that reduces the combustion volume, while maintaining or increasing the swept volume of the cylinder, will increase the compression ratio. Some of the more common methods are decreasing the volume of the combustion chambers (by replacing or milling the heads), using thinner head gaskets, changing the location of the piston-pin or rod length to move the piston closer to the combustion chamber, installing pistons with larger domes, and others. All of these modifications will increase compression ratio. And that increases horsepower, right? Well, surprisingly, the answer to that question is often “Yes!”. But do you know why?

Why Higher Compression Ratios Produce More Horsepower

Most automotive enthusiasts believe that a higher compression ratio directly raises horsepower output. Before you read the next few paragraphs, see if you can explain to yourself why compressing the charge before combustion is necessary. Or is it necessary? If it is, what causes a boost in power? Does it burn the fuel more completely or efficiently? What about the power that’s needed to compress the charge? Is it simply “lost” energy?

The answers to these questions lie in the laws of thermodynamics. Luckily, we don’t have too delve to deeply into this complex subject to gain an insight into how compression before ignition works. Let’s take a simplified look at what happens inside the engine when the compression ratio is increased. Picture a spherical combustion space containing twice as much volume as the cylinder that’s attached to it. This configuration produces an engine with a very low 1.5:1 compression ratio. Just before the intake valve closes, the piston is positioned at BDC, and the cylinder and the spherical volume are exposed to atmospheric pressure of about 14psi. As the piston moves up the bore, the valve closes and the charge is compressed. When the piston reaches TDC the pressure in the cylinder will rise to about 21psi. At this point the spark plug fires and drives the post-combustion pressure to about 250psi and the piston is pushed back down the bore. About 12 to 14 degrees after TDC, the cylinder pressure driving the piston down the bore will be about 230psi.

Now picture the same engine, except this time the spherical combustion space has been reduced in size to about 1/10th of the volume of the cylinder. The compression ratio is now 10:1. As the intake valve closes at BDC, the cylinder and the volume are once again at atmospheric pressure. When the piston reaches TDC on the compression stroke, the small volume has driven the compression pressure to about 400psi, or 18 times as high (the new volume is about 18 times smaller). When the piston reaches 12 to 14 degrees after TDC, the pressure on the piston is about 1500psi. The higher compression ratio produces much higher cylinder pressures throughout the first half of the piston’s travel from TDC and BDC on the power



A spherical combustion space containing twice as much volume as the cylinder produces a 1.5:1 compression ratio. Peak cylinder pressures (see text) will be about 250psi. With a combustion space about 1/10th of the volume of the cylinder, the compression ratio is now 10:1. Peak pressures now reach about 1500psi. The higher compression ratio produced much higher cylinder pressures throughout the first half of the piston’s travel from TDC and BDC on the power stroke. This additional pressure generates a much larger force across the surface of the piston, and that increases torque and horsepower.

stroke. This additional pressure generates a much larger force across the surface of the piston and that increases torque and horsepower.

While it may now be clear that higher post-ignition pressures result in increases in torque output, you may still be wondering how much power is *consumed* to compress the charge to higher initial pressures. This is more easily understood if you picture what happens when the spark plug doesn't fire. It certainly takes power to drive the piston up the bore to compress the charge, but without ignition nearly identical pressures drive the piston back down the bore on the "power" stroke. The net result (forgetting about friction, mechanical, and heat losses) is zero. In other words, under ideal circumstances it take no more net power to compress a charge to a lazy 21psi with a 1.5:1 compression ratio than it does to raise it to 400psi with 10:1 compression because the consumed work is returned on the power stroke. In a real engine, higher compression ratios increase losses from charge heating and, especially, from increased ring-to-cylinderwall friction. But the power lost is usually smaller than the power gained. Testing done by GM many years ago indicated that, for gasoline as the fuel, power will continue to increase until compression ratios reach about 17:1. Considering that many racing engines are now hovering around this level, GM may have been right.

Other Effects Of Increasing Compression Ratio

As mentioned above, the greatest losses from high compression and high cylinder pressures (except when detonation is induced) occur from ring/cylinderwall friction and heat losses into the pistons and water jackets. Beyond these well-known effects, changing the compression ratio has other subtle consequences, some of which are largely unexplored.

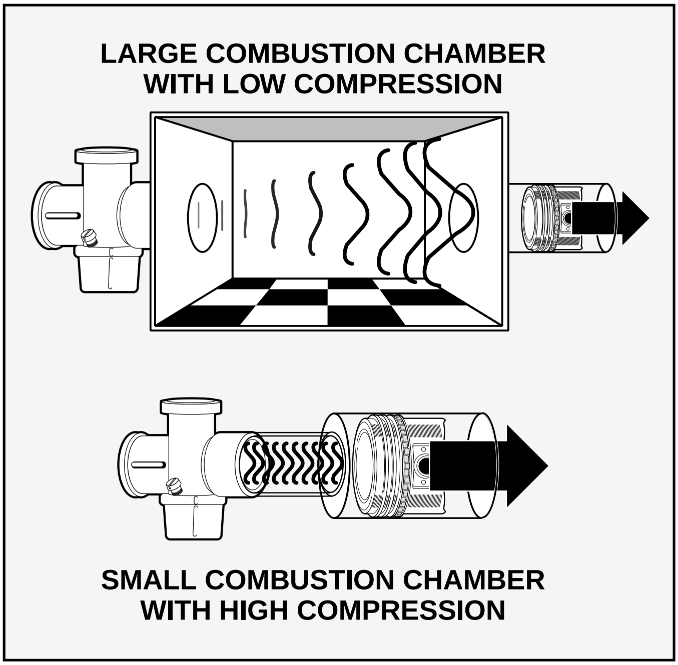
Let's examine a potential source of "hidden" power. The combustion space volume above the piston tends to act as an absorber, slightly smoothing the pressure pulses in the cylinder. This effect can dampen the peaks of the negative pressure waves created when the piston moves down the bore on the intake stroke. When the compression ratio is increased, the combustion space decreases. With less volume to absorb pressure pulses, the pressure drop on the intake stroke may be more "directly linked" to the induction system, improving cylinder filling. You can visualize this effect by picturing a small cylinder attached to a large room (the room exaggerates the effects of the large combustion space used in low compression ratio applications). As the piston moves down the bore, any drop in pressure is dissipated in the room, so that the carburetor—attached at the adjacent wall—receives an extremely weak signal and virtually no air/fuel mixture flows into the room or cylinder. Now reduce the size of the room to a small space, more like the real-world conditions in a high-compression engine. As the piston moves down the bore, the drop in pressure is almost directly "linked" to the induction system, instantly drawing air and fuel into the cylinder. These same positive effects, though not as pronounced, should occur anytime the compression ratio is increased by decreasing the combustion space.

Compression Ratio Assumptions

Changes in compression ratio have hundreds of consequences throughout the engine. However, Motion's Filling And Emptying simulation program analyzes only some of these changes. Changes that occur because of alterations in the wave dynamics within the engine are not well predicted by the current program. Thermodynamic effects, however, are very accurately modeled in the simulation. In order to understand the variations in power and torque curves produced as the compression ratio is changed, it is helpful to understand some program assumptions.

The program assumes that ignition timing is always optimum. That is, based on the gasoline burn model used by the program, ignition occurs at a point that produces peak cylinder pressures between 12 to 14 degrees after top dead center. Tests

Here's a source of "hidden" power. A large combustion space that accompanies a low compression ratio (illustrated here by the large box attached to the carburetor) tends to act as an absorber, smoothing pulses in the cylinder. This dampens the negative peaks created when the piston moves down the bore on the intake stroke. When the compression ratio is increased, the combustion space decreases. With less volume to absorb pulses, the intake stroke is more "directly linked" to the induction system, improving cylinder filling.



have indicated that these same conditions reproduced in the real world often optimize power output.

The limitations of the combustion model prevent changing the ignition point. The simulation of varying ignition timing—or performing sophisticated combustion analysis to reveal detonation, preignition, or provide emissions analysis—requires advanced techniques. While these models do exist, they not only need full three-dimensional maps of the constantly-varying combustion space, but they consume, literally, days of computational time. Obviously, this is not practical approach for a quick-response "what-if" program.

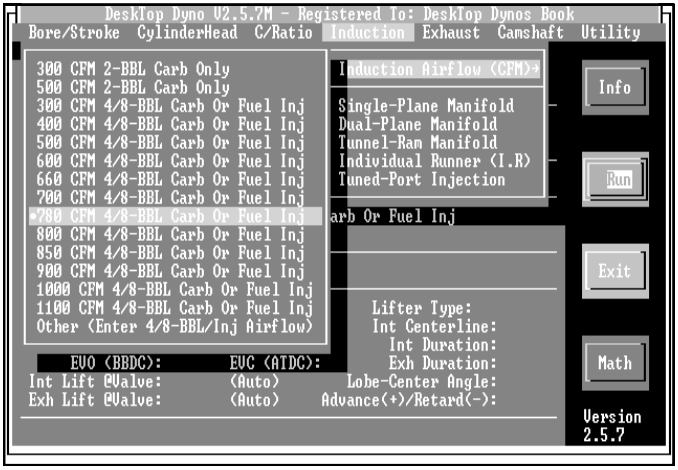
THE INDUCTION MENU

The fourth main component menu establishes an **Induction** system for the simulated test engine. An induction system, as defined in Motion's simulation, is everything upstream of the intake ports, including the intake manifold, common plenums (if used), carburetor/throttle main body, venturis (if used), and opening to the atmosphere. The Induction menu is divided into two submenus: an Airflow menu to select the maximum-rated airflow that can pass through the induction system, and the Manifold submenu to establish an intake-manifold configuration. One choice from each of these two groups fixes a specific induction system from among thousands of possible combinations.

Airflow Selection

The first Induction submenu is used to select the rated airflow for the induction system. This menu consists of two 2-barrel carburetor selections, twelve 4-barrel carburetor/fuel injection choices, and an "Other..." selection in which you can directly specify the rated airflow from 100 to 3000cfm. The first two selections "install" either a 300- or 500-cfm 2-bbl carburetor on the test engine. These are the only 2-barrel choices directly available in the menu. The remaining choices range from 300 to 1100cfm on 4- or 8-barrel carburetor and fuel injection applications.

In order to perform "apple-to-apple" comparisons among the Airflow selections, it is important to realize that the ratings for 2-barrel carburetors are measured at a pressure drop twice as high as the pressure



The first Induction submenu is used to select the rated *Airflow* for the induction system. This menu consists of 2-barrel and 4/8-barrel carburetor/fuel injection choices, and an "Other..." selection by which you can directly specify the rated airflow from 100 to 3000cfm.

used to rate 4-barrel carburetors and fuel-injection systems. Rated airflow for 2-barrels is typically measured at a pressure drop of 3 inches of mercury (the pressure differential maintained across the carburetor during airflow measurement at wide-open throttle). This is often written as "3-in/Hg" ("Hg" is the symbol for mercury used in the periodic table of elements). The higher pressure drop increases the measurement resolution for smaller carburetors and "shifts" the flow numbers toward the range commonly found in automotive applications (roughly, 100 to 700cfm).

Knowing the differences in rating methods, it is a simple task to convert any 2-bbl flow into it's 4-bbl equivalent. Here's the formula:

$$\textbf{4-bbl Flow} = \frac{\textbf{2-bbl Flow}}{\textbf{1.414}}$$

Using this conversion, it is possible to simulate virtually any 2-barrel carburetor induction system. For example, a custom 2-barrel that flows 650cfm at 3-in/Hg, would flow 460cfm if measured at 1.5-in/Hg (you can confirm this using the above formula). By manually entering 460cfm into the Component Selection Box of the simulation, the program will accurately model this custom 2-barrel.

The remaining choices in the Induction Airflow menu are labeled with "4/8-BBL Carb Or Fuel Inj." This means that the selections below the first two designate airflow ratings that were measured at 1.5-in/Hg. The "4/8-BBL" indicates that the induction system can consist of single or multiple carburetors that, combined, produce the rated airflow. For example, the menu choice "1000 CFM 4/8-BBL Carb Or Fuel Inj" can indicate one 1000cfm 4-bbl carburetor or two 500cfm 4-bbls. The same 1000cfm selection could even indicate three 470cfm 2-bbls (i.e., 3 x 470 = 1410cfm; converting to 4-bbl flow: 1410/1.414 = 997cfm). The important thing to remember about airflow selection is that the program *makes no assumption about the type of restriction* that makes up the carburetor or injection system. The airflow is simply a rating of the total restriction of the induction system.

Motion's Filling-And-Emptying model uses this restriction value to calculate a critically important variable needed by the simulation to accurately determine mass flow into the cylinders: manifold vacuum. Here's how the process works:

1) The simulation runs through an entire cycle (all four Otto cycles) to initially determine the total mass flow into the cylinders. This determination—because it is based on a degree-by-degree, crank-angle analysis—takes the entire range of engine variables into consideration, including displacement, engine speed, valve size, cam timing, compression, and assumptions about the intake manifold, exhaust system, and more.

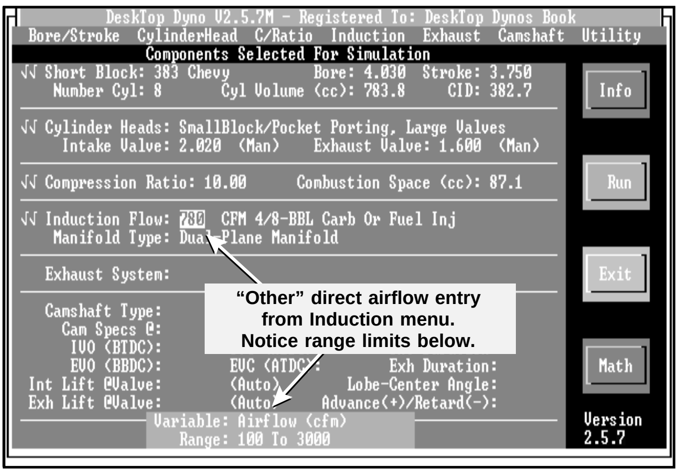
2) Since there can never be mass accumulation or loss within the engine, the same mass that flows into the cylinders must also flow through the induction (and exhaust) system.
3) Since atmospheric pressure exists on one side of the carburetor or throttle body, using the selected induction restriction (chosen from the airflow submenu), the program can calculate the vacuum generated in the manifold that will produce the same predicted rate.
4) When the manifold vacuum has been determined, the charge density can be calculated. Knowing the charge density is the final step used by the program to "iterate," or home-in on, a precision determination of the air/fuel mass entering the cylinder.

Throughout these simulation steps, the carburetor creates a restriction that produces a vacuum in the manifold. Larger carburetors produce less restriction and decrease vacuum. As vacuum drops and manifold pressure approaches atmospheric levels, the density of the charge becomes greater. When the density of the air/fuel charge increases, a greater mass of air and fuel can be drawn into the cylinders, resulting in higher power output. Seemingly, the conclusion here is that the greater the flow capacity of the carburetor, the more power the engine can produce. While this may hearken back to the old cliché "the bigger the better," in theory, this trend is absolutely correct. The carburetor can be thought of as a density-regulating device, and any restriction generated by the carburetor (or injection system) increases manifold vacuum and decreases charge density and power.

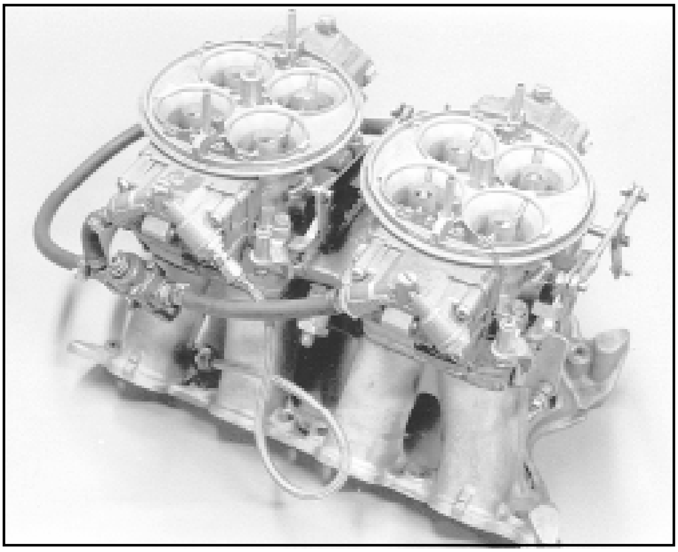
In the real world, however, carburetors must generate a pressure drop across the venturis in order to atomize fuel and air in the proper proportions. In fact, most carburetors must generate at least 0.5-in/Hg pressure drop at the lowest airflow levels to function properly (e.g., during part throttle transition to full throttle at low engine speeds). This same requirement does not exist for modern fuel-injection systems. Many high-performance electronic and mechanical injection systems maintain precise air/fuel mixtures throughout the rpm range while generating negligible restriction and manifold vacuum at full throttle. While there are practical limitations to "the bigger is better" rule, in theory, low restriction and high charge density clearly produce more power.

Airflow Menu Assumptions

As higher airflow levels are selected from the Induction menu, the simulation lowers the restriction within the induction system. This decrease in restriction increases charge density. Along with this concept, the simulation assumes that *the air/fuel ratio is always at the precise proportion for optimum power*. While optimum air/fuel ratios are more achievable with fuel injection



Choosing "Other" from the Induction Airflow menu allows direct entry of any airflow rating for the simulated induction system, providing it falls within the acceptable range limits. Airflow values entered using this technique are assumed to be rated at a pressure drop of 1.5-in/Hg, the standard of measurement for 4-barrel carburetors.



Larger carburetors produce less restriction and decrease manifold vacuum. Seemingly, the greater the flow capacity, the more power the engine will produce. In theory this trend is correct. The carburetor is a density-regulating device with any restriction increasing manifold vacuum and decreasing charge density, resulting lower power. In the real world, however, carburetors must generate a pressure drop of about 0.5-in/Hg to function properly.

systems, a carefully tuned carburetor also can come remarkably close to ideal fuel metering. Regardless of whether the simulated engine uses carburetors or fuel injection, the power levels predicted by the program can be considered optimum, achievable when the engine is in "peak" tune and the induction system is working properly.

The airflow (in Cubic Feet per Minute, or CFM) selected from the Induction Airflow menu is also assumed to be the *total rated airflow into the engine*. On dual-inlet or multiple-carburetor systems, the total airflow is the sum of all rated airflow devices. So a manifold equipped with twin 1100cfm Holley Dominators would have a rated airflow of 2200cfm. If an air cleaner is used, total airflow must be adjusted to compensate for the increase in restriction (contact the element manufacturer or flow test the carburetor/air-cleaner as an assembly).

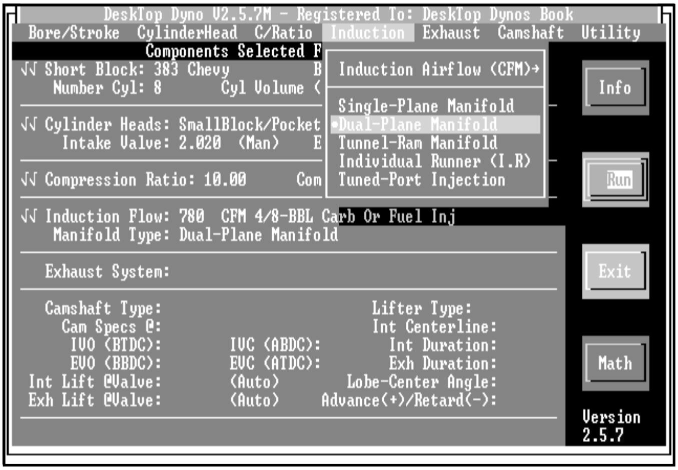
Also keep in mind the unique way airflow capacities are handled on Individual Runner (I.R.) manifolds. On these induction systems, each cylinder is connected to a single "barrel" or injector stack with no connecting passages that allow the cylinders to "share" barrels. The total rated flow for these induction systems is divided among the number of cylinders. For example, a smallblock V8 equipped with 4 Weber carburetors (having 8 barrels) may have a total rated flow of 2000cfm. To properly model this system, 2000cfm is directly entered into the simulation by choosing "Other..." from the Airflow menu. When an "I.R." manifold is selected from the second part of the Induction menu (more on manifolds next) the airflow is divided into all 8 cylinders, allotting 250cfm to each cylinder.

Induction Manifold Basics

The lower half of the Induction menu consists of five manifold choices. Each of these manifolds applies a unique tuning model to the induction system, but before we cover the particulars of each selection, it is helpful to review induction tuning and how wave-dynamic models are used to reveal manifold function. For a more in-depth look at induction tuning and wave dynamics, refer to the complete *DeskTop Dynos* book available from Motion Software, Inc.

The flow of air and fuel within engine passages is influenced by waves generated by rapidly changing pressures within the induction and exhaust systems. These pressure "pulses" arise from the release of high pressure exhaust gasses into the ports and headers during the exhaust cycle and by the "pumping action" of the piston dur-

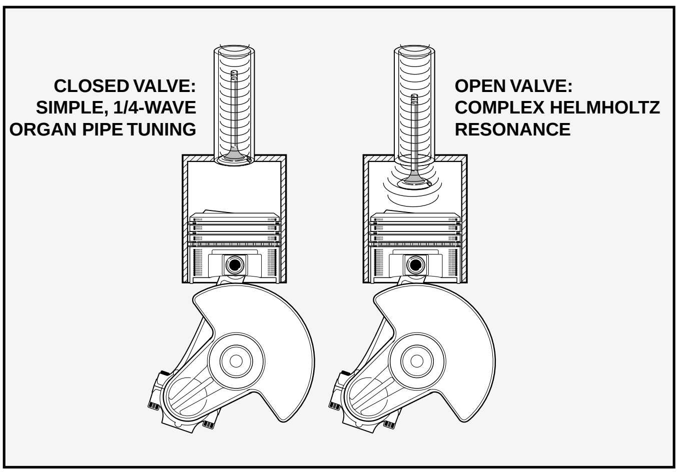
The lower half of the Induction menu consists of five *Manifold* choices. Each manifold applies a unique tuning model to the induction system. Refer to the text for information on the design of each manifold, an overview of how the manifold boosts power or torque, and finally, a description of the assumptions and recommendations associated with each menu selection.

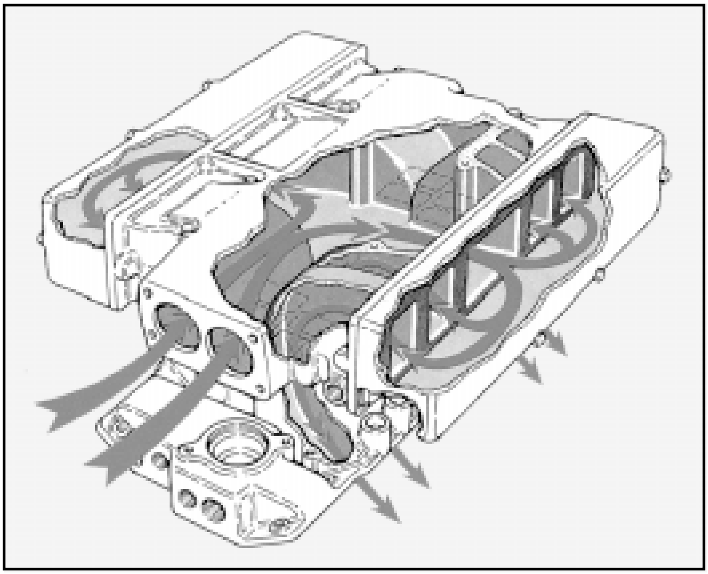


ing the intake cycle. These pressure waves are thousands of times stronger than waves we are familiar with in everyday life: common acoustic or sound waves. At these high energy levels, IC engine pressure waves, now referred to as *finite-amplitude waves*, alter their shape as they travel through passages, and they interact with each other in ways considerably more complex than simple acoustic waves. The combination of these phenomena make the "solutions" to finite-amplitude wave analysis extremely difficult.

This complexity is multiplied by the configuration of the ports, valves, manifold runners and plenum shapes. The levels of complexity continue to grow as the intake and exhaust valves open and close. Initially, with the intake valve closed, the port/runner system forces pressure waves through a basic oscillation cycle, bouncing back and forth off of the closed valve at one end of the runner and the open manifold plenum at the other. This "reasonably simple" process, called *1/4-wave* or *organ-pipe tuning*, delivers a series of decaying-amplitude pressure pulses to the closed intake valves. When the intake valves are open, however, the system switches from organ-pipe tuning to a much more complex Helmholtz resonator. This is the same resonance that you can duplicate by blowing into the neck of a jug, creating a deep "whirring" sound. Not only does pressure-wave analysis have to deal with this more complex resonance, but the Helmholtz

When the intake valve is closed, the port/runner system forces pressure waves through a basic oscillation cycle called *1/4-wave* or *organ-pipe tuning*. When the intake valve is open, the system switches to a much more complex Helmholtz resonator with <u>changing volume</u> as the piston moves in the cylinder and <u>varying restriction</u> as the valve opens and closes.





The five manifolds included in Motion's simulation are, by no means, a comprehensive list of all the intake manifolds available for IC engines. However, the five discrete designs within the menu can be applied to a wide range of manifolds. This SLP TPI manifold can be simulated by applying a large airflow rating to the Tuned-Port Injection model.

resonator is <u>changing its volume</u> as the piston moves in the cylinder, and the <u>restriction at the "neck" is also varying</u>, as the valve moves through its lift curve. Combining all these factors with the already complex interaction of finite-amplitude waves gives a glimpse of the mathematical sophistication needed to analyze pressure waves inside the IC engine.

After all this, you may be wondering how much of an influence this morass of complex pressure waves has on engine performance. The answer is a lot! They can either aid or restrict cylinder filling depending on the design of inlet ducting. A carefully constructed Pro-Stock induction system will use these invisible pressure waves to gain hundreds of horsepower. Even street engines can use induction tuning to improve throttle response, fuel economy, and add "seat-of-the-pants" power.

Induction-tuning power benefits come from several techniques, but the most direct is to harness the suction wave created during the intake stroke. Optimum runner lengths will return a reflection of this negative pressure wave to the intake valve on the next intake cycle when the piston reaches maximum velocity, about 70 degrees after TDC. The returning suction wave combines with the low pressure created by the piston rapidly moving down the bore to produce a powerful draw of air and fuel into the cylinder. Then, again, just before the intake valve closes as the piston is beginning to move up the cylinder, the induction system returns a positive pressure pulse to minimize or prevent "reversion" of charge back out of cylinder.

These tuning effects can be adjusted for specific applications by changing runner length, volume, passage interconnections, and plenum configuration. In other words, installing intake manifolds of different design can have dramatic effects on how the pressure waves are used to assist cylinder filling and control engine power. The following section details each of the manifold choices provided in the Induction menu and explains the assumptions and limitations associated with each design.

Manifold Selection Advice And Design Assumptions

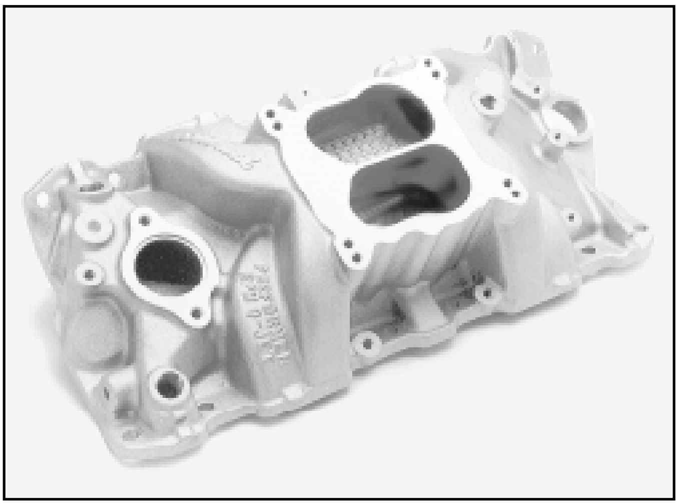
The complex interaction of pressure waves within the induction system require a rigorous mathematical analysis, involving the Method Of Characteristics discussed in the complete *DeskTop Dynos* book. This advanced technique uses considerable computational time and does not lend itself to simulations designed for a rapid "what-if" interaction with the user. Furthermore, extensive analysis of intake manifold configurations would require the input of many detailed variables, such as runner lengths, volumes, taper angles, plenum configuration and dimensions, and much more. These technical inputs would further shift the emphasis away from ease of use toward a dedicated research tool. However, in order to evaluate manifold differences, any program—including Motion's Filling-And-Emptying simulation—must incorporate some pressure-wave analysis to make accurate estimations of power and torque differences. Motion's program uses a "mini-wave action" model that offers three advantages for most users: 1) rapid calculation times allowing fast back-to-back testing, and 2) the mini-model does not require dimensional data entry for the ports and runners, making component selection extremely simple, and 3) overall accuracy is quite good (within 10%) and trends in power differences between the five manifold types are reliable. For those individuals involved in engine development programs or intake manifold research, the mini-model will not provide sufficient data resolution to make subtle design changes. However, for nearly everyone else, the mini-model should provide a good compromise between speed, ease of use, and predictive accuracy.

The five manifolds included in Motion's simulation are, by no means, a comprehensive list of all the intake manifolds available for IC engines. The list of five should, instead, be interpreted as five discrete designs that approach the limits of resolution of the mini-model within the program. If you are interested in a manifold with a design that falls in between two menu selections, you can often use the "trend" method to estimate power for a hybrid design. For example, run a test simulation using manifold Type A, then study the differences in power attributed to manifold Type B. The changes will indicate trends that should give you insight into how a hybrid manifold Type A/B *might* perform. Because a rigorous analysis of pressure waves is not performed by the current program, keep in mind that the data you obtain might not match real-world dyno data with some combinations. In general, however, the trends and overall accuracy should be within 10%.

For each manifold described below, you will find information about its basic design, an overview of how the manifold boosts power or torque, and finally, a description of the assumptions and recommendations associated with that individual design.

Dual-Plane Manifold—Remarkably, the well-known and apparently straightforward design of the dual-plane manifold is, arguably, the most complex manifold on the list. An intake manifold is considered to have a dual-plane configuration when 1) the intake runners can be divided into two

The Edelbrock Performer Q-Jet represents the current state-of-the-art in dual-plane manifold designs. This manifold is said to have a 2nd degree of freedom. This powerful resonance multiplies the force of the pressures waves, simulating the effects of long runners, boosting low- and mid-range power.

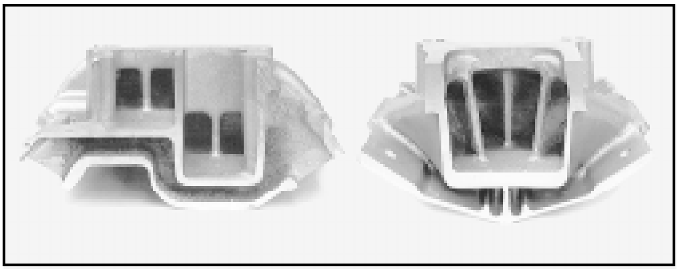


groups, so that 2) each group alternately receives induction pulses and 3) the pulses are spaced at even intervals. If all of these criteria are met, the manifold is said to have a 2nd degree of freedom, allowing it to reach a unique resonance causing the entire manifold and all the runners oscillate in unison. During this period, pressure readings taken throughout the manifold will be in "sync" with one another. This powerful resonance multiplies the force of the pressures waves, simulating the effects of long runners. Since longer runners typically tune at lower engine speeds, not surprisingly, the dual-plane manifold is most known for its ability to boost low-end power.

The divided plenum is another common feature of dual-plane manifolds that further boosts low-end power. Since each side of the plenum is connected to only one-half of the cylinders (4-cylinders in a V8), each cylinder in the engine is "exposed" to only half of the carburetor. This maximizes wave strength and improves low-speed fuel metering (these effects are much less pronounced with throttle-body fuel-injection systems). However, the divided plenum can become a significant restriction at higher engine speeds and limit peak horsepower.

The main benefits of the dual-plane are its low-speed torque boosting capability, compact design, and wide availability for use with both carburetors and injection systems. However, not all engines are capable of utilizing a dual plane. Typically, engines that do not have an even firing order or have too many cylinders to generate a resonance effect will not benefit from a dual-plane manifold. While there are some exceptions, engines having 2 or 4 cylinders work best with this manifold. Since most V8 engines are basically two 4-cylinder engines on a common crankshaft, even-firing V8s also benefit from the resonance effects of the dual-plane manifold. Motion's simulation does not prevent choosing a dual-plane manifold on engines that will not develop a full resonance effect. For example, you can install a dual-plane manifold on a 5-cylinder engine, but the results—a low-end power boost—are not reproducible in the real world, since an effective dual-plane manifold cannot be built for this engine. The simulation is best utilized by modeling dual-plane manifolds combinations that already exist rather than testing theoretical fabrications.

Many dual-plane manifolds are hybrids incorporating facets of other manifold designs. Especially common is the use of an undivided or open plenum typically associated with single-plane manifolds. These hodgepodge designs are attempts at harnessing the best features while eliminating the worst drawbacks of various designs. Sometimes, the combinations are successful, adding more performance without much of a sacrifice in low-speed driveability. With



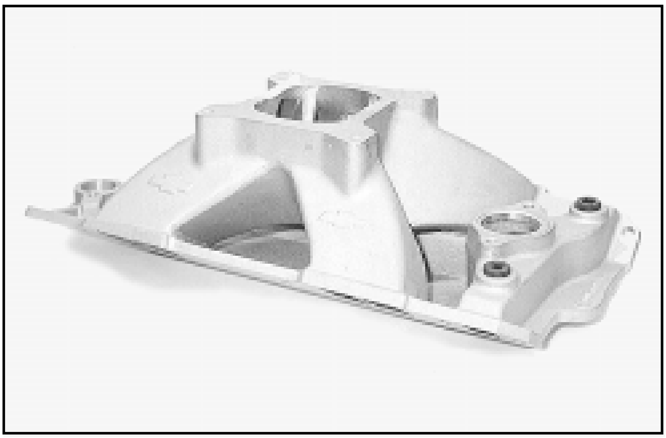
The basic difference between single- and dual-plane manifolds are clearly illustrated here. The dual-plane (left) divides the plenum in half, with the runners grouped alternately by firing order. Each cylinder "sees" only one-half of the carburetor, transferring a strong signal to the venturis. This manifold design is said to have a 2nd degree of freedom, allowing it to reach a unique resonance that makes its short runners boost low-speed power. The single-plane manifold (right) has short, nearly equal-length runners with an open plenum, much like a tunnel ram but "laid" flat across the top of the engine. The manifold has excellent high-speed performance, but a loss of 2nd degree of freedom prevents full-manifold resonance. That reduces low-speed torque and degrades driveability and fuel economy.

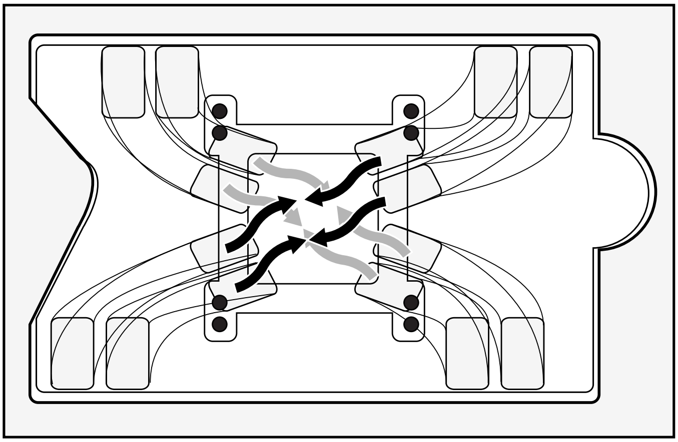
Many dual-plane manifolds are hybrids. This Edelbrock dual-plane manifold is designed for the 440 Chyrsler engine and has a partially open plenum. In this case, the opening adds mid-range and high-speed performance with little sacrifice in low-speed driveability. Not all hybrid designs are as successful as this one. In situations where you are not familiar with engine or manifold characteristics, it may be worthwhile to stick with "plane-vanilla" designs.

these designs, you can successfully use the "trend" method described earlier to estimate engine torque and power. However, there is no shortage of manifolds that can reduce power without giving anything back in driveability or fuel economy. In fact, some of the worst designs are remarkably bad. It is impossible to determine which of these combo designs is better than others using the Filling And Emptying simulation. Only a simulation that perform models intake passages, including the complex effects of multicylinder interference, can perform this analysis. Unless you can perform actual dyno testing on these manifolds to find what works and what doesn't, it may be worthwhile to stick with more "plain-vanilla" designs that produce predictable results.

Single-Plane Manifold—In a very real sense, a single-plane manifold as used on most V8 engines is simply a low-profile tunnel ram. The tunnel-ram manifold (discussed next) is a short-runner system combined with a large common plenum; a design that optimizes power on all-out racing engines where hood clearance is not an issue. The single-plane manifold combines short, nearly equal-length runners with an open plenum, but "lays" the entire configuration flat across the top of the engine. The results are quite predictable. A loss of 2nd degree of freedom prevents full-manifold resonance. That produces a loss of low-speed torque, and depending

A single-plane manifold is simply a low-profile tunnel ram. The design combines short, nearly equal-length runners with an open plenum, but "lays" the entire configuration flat across the top of the engine. The single-plane manifold combines improved flow capacity, higher charge density, and short runners to build substantial horsepower at higher engine speeds.





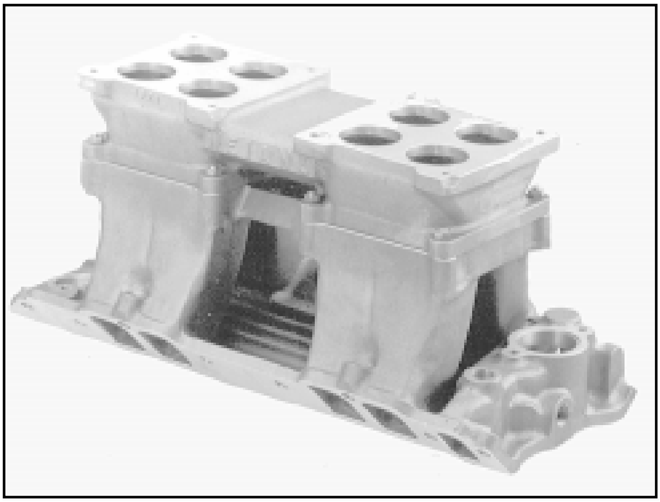
The typically compact, low-profile design of the single-plane manifold has drawbacks. The runners are connected to a common plenum. This arrangement tends to create unpredictable interference effects as pressure pulses moving through the runners meet in the plenum and stir up a complex soup, sometimes creating irregular fuel-distribution.

on the size of the plenum and runners, single-plane manifolds can also degrade driveability and fuel economy. Furthermore, a large-volume, undivided plenum contributes to low-speed problems by presenting every cylinder to all barrels of the carburetor, lowering venturi signal and low-speed fuel metering accuracy (again, this drawback is minimized with a fuel-injection system). On the other hand, the single-plane manifold (like the tunnel ram) combines improved flow capacity, potentially higher charge density, and short runner lengths to build substantially more horsepower at higher engine speeds.

As a high-performance, high-speed manifold, the single-plane design has many advantages, but it's compact, low-profile design has drawbacks too. The runners are connected to a common plenum like spokes to the hub of a wheel. This arrangement tends to create unpredictable interference effects as pressure pulses moving through the runners meet in the plenum and stir up a complex soup. Large plenum volumes help cancel some these effects, but open-plenum, single-plane manifolds may produce unexpected changes in fuel distribution and pressure-wave tuning with specific camshafts, headers, or cylinderheads (to some degree, these effects are present in all manifold designs). Predicting these will-o'-the-wisp anomalies requires rigorous modeling, well beyond the capabilities of the Filling-And-Emptying simulation. Currently, pinning down these problems requires dyno testing with exhaust temperature probes to measure fuel distribution accuracy.

Designers and engine testers have experimented with hybrid single-plane manifold designs that incorporate various dual-plane features. One common modification is to divide the plenum into a pseudo dual-plane configuration. While this does increase signal strength at the carburetor, uneven firing does not allow 2nd degree of freedom resonance. This modification can cause sporadic resonances to occur throughout the rpm range with unpredictable results. Spacers between the carburetor and plenum are also commonly used with single-plane manifolds often with positive results, particularly in racing applications. Spacers probably increase power for two reasons: 1) By increasing plenum volume they tend to reduce unwanted pressure-wave interactions, and 2) A larger plenum improves airflow by reducing the angle the air/fuel must negotiate as it transitions from "down" flow from the carburetor to "side" flow into the ports. While there is no way to use trend testing to evaluate the effects of a divided plenum, spacers can be partially simulated. The increase in plenum volume tends to mutate the single-plane manifold into a "mini" tunnel ram, so horsepower gains tend to mimic those obtained by switching to a tunnel ram design (i.e., small performance improvements, when found, usually occur at high rpm).

Since the single-plane manifold typically reduces low-speed torque and improves high-speed horsepower, it is often the best



This Weiand BB Chevy tunnel ram manifold is a single-plane induction system designed to produce optimum power on all-out racing engines. It has a large common plenum and short, straight, large-volume runners. The tunnel ram manifold menu selection has the potential to produce the highest peak horsepower of all the manifolds listed in the *Induction* menu.

compact manifold design for applications where engine speed is typically 4000rpm or higher. If the engine commonly runs through lower speeds, a dual-plane, individual runner, or tuned-port injection system will usually provide better performance, driveability, and fuel economy.

Tunnel-Ram Manifold—This intake manifold is a single-plane induction system designed to produce optimum power on all-out racing engines. The advantages of the tunnel ram come from its combination of a large common plenum and short, straight, large-volume runners. The large plenum has plenty of space to bolt on two carburetors, potentially flowing up to 3000+cfm to optimize charge density. The large plenum also minimizes pressure-wave interaction and fuel distribution issues. The short runners can be kept cooler than their lay-flat, single- and dual-plane counterparts, and they offer a straight path into the ports, optimizing ram-tuning effects.

Applications for the tunnel ram are quite limited because of its large size; vehicles using tunnel-ram manifolds often require a hole in the hood and/or a hood scoop for manifold and carburetor clearance. While a protruding induction system may be a "sexy" addition to a street rod, in single-carburetor configurations the tunnel ram offers very little potential power over a well-designed, single-plane manifold. Only at very high engine speeds, with multiple carburetors, will the advantages in the tunnel ram contribute substantially to power.

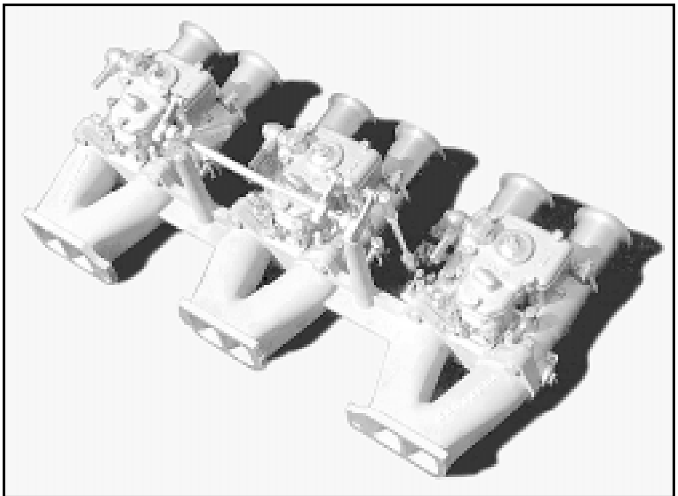
This tunnel-ram selection can also accurately model fuel-injection systems with large, individual stacks. Strictly speaking, while the simulation combines short runners and a large-volume plenum, this design mimics short injector stacks that open to the atmosphere quite well. For one-barrel-per-cylinder Weber carburetion or small-diameter, individual-injector systems, use the Individual Runner manifold described next. However, for large-diameter injectors, like Hillborn or Crower systems, the tunnel-ram manifold—along with the appropriate airflow selection (for all cylinders combined)—is a good induction model.

The tunnel ram manifold selection has the potential to produce the highest peak horsepower of all the manifolds listed in the Induction menu. The large cross-sectional areas, straight runners, and short tuned lengths make this manifold a "no compromise" racing design.

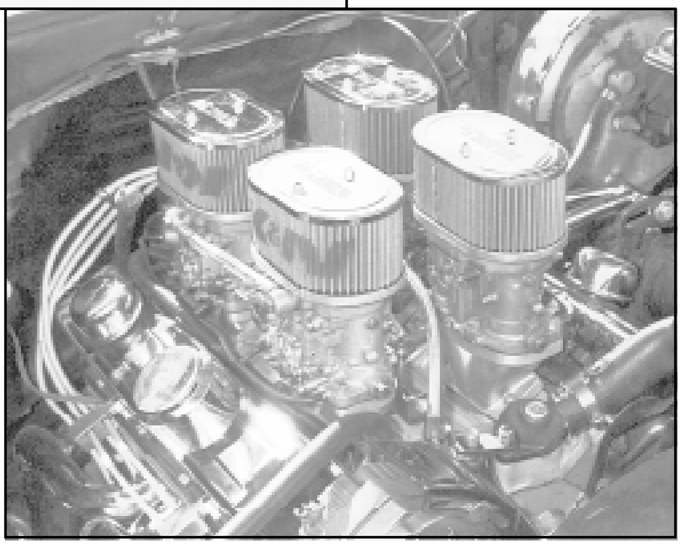
Individual Runner—A manifold that connects each cylinder to one "barrel" of single or multiple carburetors *with no interconnecting passages for shared flow* is considered an individual (or isolated) runner system (I.R. for short). A multiple Weber or Mikuni carburetor setup is a well-known example of this type of induction system.

On a V8 engine, four twin-barrel Webers make a very impressive sight, and at first glance they may appear to offer more airflow potential that any engine needs, particularly any street engine. While it may look like overkill, the one-barrel-per-cylinder arrangement often has substantial horsepower limitations due to airflow restriction! A typical Weber 48IDA carburetor flows about 330cfm per barrel. While the sum total of all eight barrels is over 2600cfm (a flow rating equivalent to two Holley Dominators), the important difference here is that each cylinder can draw from only one 330cfm barrel. In a single- or even a dual-plane manifold, each cylinder has access to more than one carburetor barrel, reducing restriction during peak flow and increasing high-speed horsepower. While an I.R. system offers substantial low-end performance benefits (more on that next), at 5000rpm and higher on typical smallblock installations, power can fall below the levels of an average single four-barrel, 780cfm induction system.

Again, on first impression, a multiple-carburetor I.R. induction may seem to offer way too much flow capacity, making it easy to believe that it's plagued with low-speed carburetion problems. Surprisingly, the same one-barrel-per-cylinder arrangement that produces a restriction at high engine speeds, transmits strong pressure waves to each carburetor barrel at low



A manifold that connects each cylinder to a single carburetor barrel *with no interconnecting passages for shared flow* is considered an individual (or isolated) runner system (I.R. for short). Multiple Weber or Mikuni carburetor systems are well-known examples of this type of induction system. This I.R. manifold was designed for early OHC Pontiacs.



A multiple-carburetor I.R. induction may seem to offer way too much flow capacity. Surprisingly, the one-barrel-per-cylinder transmits strong pressure waves to each carburetor barrel, producing excellent throttle response. Individual-runner manifolds are an outstanding induction choice for high-performance street engines.

speeds, producing ideal conditions for accurate fuel metering. Furthermore, the pressure waves moving in the runners are not dissipated within a plenum and don't interact with other cylinders. This ensures that the reflected waves strongly assist cylinder filling and reduce reversion. The combination of these effects makes individual-runner manifolds an outstanding induction choice for low-speed to medium/high-speed engine applications, such as high-performance street engines. Unfortunately, the high cost of these systems—and current smog regulations—prevents their wider acceptance.

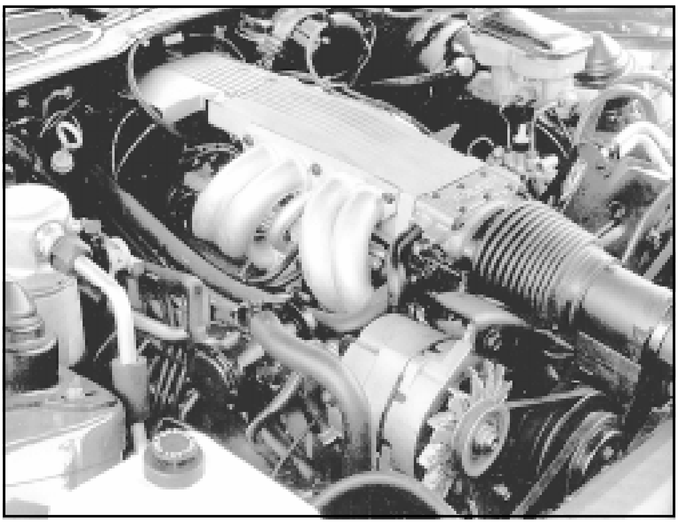
The simulation model for the Individual Runner choice in the Induction menu assumes that the runner sizes and the carburetor venturi diameters are of "medium" dimensions. Runner length, that is, the distance from the valve head to the top of the carburetors, is also assumed to be "mid-length," and so the simulation uses a mid-range rpm power bias. These assumptions work well with most I.R. applications, since this induction system is commonly used on street engines or in road racing applications that require good throttle response and a wide power band.

The I.R. menu selection can also model fuel-injection systems with small-diameter, medium-to-long length individual stacks. For large-diameter, short-length injectors, like drag-racing Hillborn or Crower systems, the tunnel-ram manifold selection provides a better induction model (see the above tunnel-ram description).

Tuned-Port Injection—This manifold was introduced by automakers in the mid 1980's and millions of them remain on the road today. It represents the first mass-produced induction system that clearly incorporated modern wave-dynamic principals. To optimize low-speed torque and fuel efficiency, the TPI manifold has very long runners (many configurations measure up to 24-inches from valve head to airbox). The runners on most TPI systems are also quite small in diameter—again, to optimize low-speed power—and, unfortunately, create considerable restriction at higher engine speeds. Characteristic power curves from this type of manifold fall slightly to significantly above a dual-plane up to about 5000rpm, then runner restriction and an out-of-tune condition substantially lowers peak power.

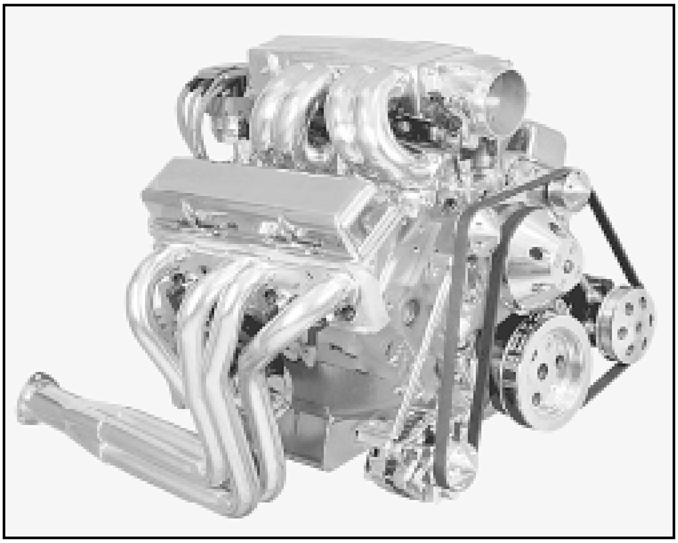
The TPI is a single-plane design that functions like a long-runner tunnel ram. Each runner is completely isolated until it reaches the central plenum. This design tends to maximize pressure-wave tuning and minimize wave interactions. Since fuel is injected near the valve, the TPI system delivers precise air/fuel ratios with no fuel distribution or puddling problems.

There is a wide range of aftermarket



The TPI manifold was introduced by automakers in the mid 1980's and millions of them remain on the road today. It represents the first mass-produced induction system that clearly incorporated modern wave-dynamic principals.

Some custom high-performance TPI and EFI (electronic fuel injection) packages are based on short-runner, high-flow tunnel ram bases. Even some long-runner systems, like this manifold from Induction Technology, allow much greater airflow than the original factory TPI. Model these manifolds by selecting dual-plane (for small-runner systems) or tunnel ram (for large-runner packages) to obtain more accurate power curves.

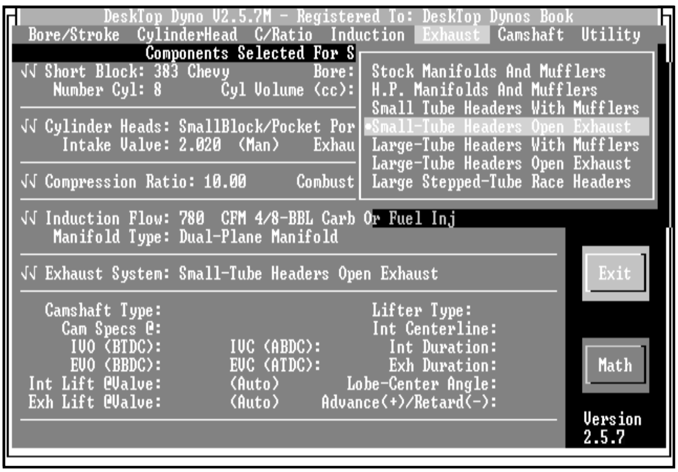


parts available for the TPI, including enlarged and/or Siamesed runners, improved manifold bases, high-flow throttle bodies, and sensor and electronic modifications. The Tuned-Port Injection selection in the Induction menu models a stock TPI. However, increasing the airflow (from the Induction Airflow menu) makes it possible to model some of the benefits of larger runners and high-flow throttle bodies.

There are now many “TPI-like” and EFI (electronic fuel injection) systems available for small- and big-block engines. Some of these custom packages are based on a short-runner tunnel ram model. Do not use the TPI manifold model to simulate these manifolds, instead, select a single-plane (for small-runner systems) or tunnel ram (for large-runner packages) to obtain more realistic power curves. Only choose a TPI manifold when the induction system uses a typical long-runner TPI configuration.

THE EXHAUST MENU

The **Exhaust** menu, the fifth component menu in the main menu bar, establishes an exhaust manifold or header configuration for the simulated test engine. The menu includes seven selections, four of which include mufflers. Since the program is designed to simulate the power levels for



The Exhaust menu selection establishes an exhaust manifold or header configuration for the simulated test engine. The menu includes seven choices, four of which include mufflers. Since the program simulates an engine <u>mounted on a dyno</u>, the exhaust system for muffled engines ends at the outlet of the muffler.

an engine mounted on a dyno testing fixture, the exhaust system for muffled engines ends at the outlet of the muffler and does not include any additional tubing used to route exhaust gasses to the rear of a vehicle.

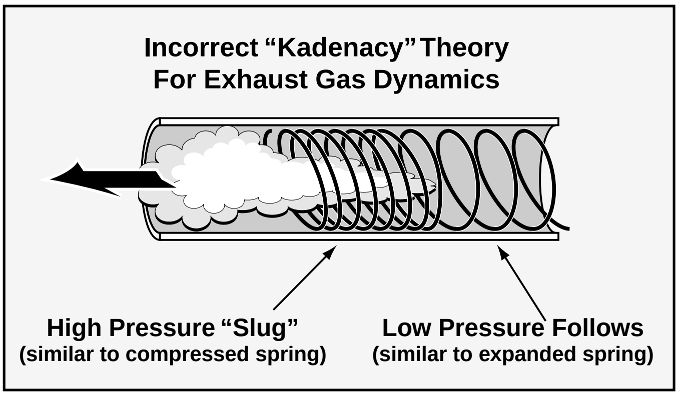
Each of these exhaust system selections apply a unique tuning model within the simulation. However, before we uncover the particulars of each choice, a short review of the wave dynamics acting within the exhaust system will help explain how "simple tubing" can boost power and engine efficiency. (Refer to the complete *DeskTop Dynos* book for a more rigorous look at the theory of exhaust-system tuning.)

Wave Dynamics In The Exhaust System

First of all, let's begin our discussion of exhaust wave-dynamic theory by describing its antithesis: the "Kadenacy" hypothesis. This theory claimed that a high pressure "slug" of gas blasted out of the port and down the header pipe when the exhaust valve opened. This moving mass was said to create a low pressure behind it, drawing additional gasses from the cylinder. This theory is analogous to compression waves traveling through a Slinky™ coil-spring toy; a tight group of coils representing high pressure waves moves along the spring followed by a more open group of coils that represent low-pressure waves. Despite the fact that this theory was conclusively proven to be incorrect in 1940, it is still believed by some engine "experts" to this day!

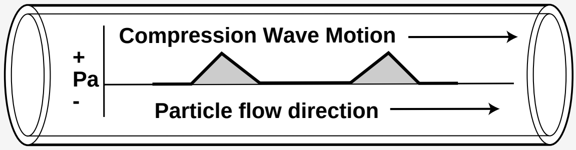
Earlier we described the high-pressure waves that move inside the induction system. These *finite-amplitude waves* contain so much energy that they no longer obey the simple laws of acoustic theory. Instead they follow an entirely different set of rules that describe interactions and wave transitions that do not occur with low-power sound waves. Induction system energies are just high enough to create finite-amplitude waves. However, exhaust system pressures are much higher and generate finite-amplitude waves even at lower engine speeds.

The same laws that describe the movement of finite-amplitude waves and gas particles in the induction system apply to exhaust flow. Without exploring the "depths" of finite-amplitude theory, the following principles should be considered essential knowledge to understanding wave action in IC engine passages, particularly the exhaust system: 1) Pressure waves and gas particles do not necessarily move at the same speeds. This phenomenon is visible as the waves on the surface of a lake wash through floating logs, pushing them nearer the shore. The waves move fairly quickly and the logs (an analogy for the gas particles) move more much slowly.

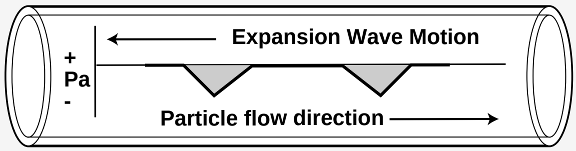


In an effort to explain gas flow in the exhaust system, it was believed that when the exhaust valve opened, a high pressure "slug" of gas blasted out of the port and down the header pipe. As this slug moved, it created a low-pressure "wave" behind it, similar to the way a compression wave travels through a Slinky™ coil-spring toy. While this easy-to-visualize theory seems to make sense, it was conclusively disproved nearly 60 years ago. Despite this, the Kadenacy effect is still widely believed by engine "experts" to this day!

In order to understand how pressure waves move within the exhaust system, these are the two most important things to remember: 1) When a positive pressure wave reaches the end of an open pipe, a negative suction wave is created that moves back up the pipe and vice versa, and 2) positive pressure waves move gas particles in the same direction as the waves, and negative pressure waves move gas particles in the opposite direction of the waves.



Compression waves before reaching end of open pipe drive particles in rightward direction



Returning expansion waves drive particles in same rightward direction

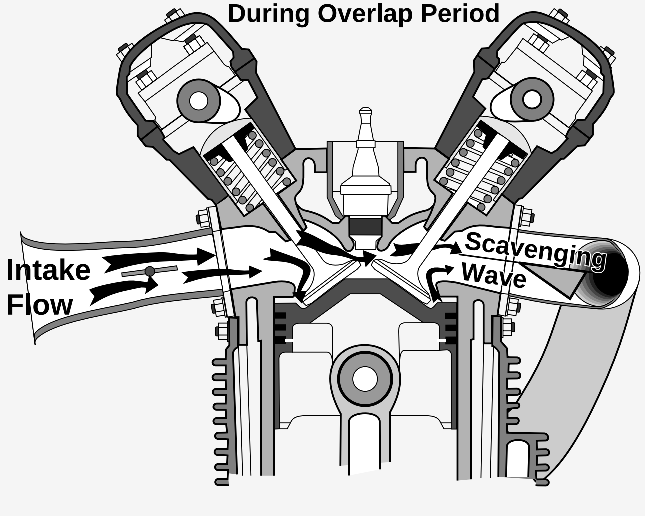
2) When a positive pressure wave reaches an area of transition, such as the end of an open pipe, a negative suction wave is created that moves back up the pipe. 3) Positive pressure waves move gas particles in the same direction as the waves; negative pressure waves move gas particles in the opposite direction of the waves.

Now let's apply this tuning theory to the exhaust system. When the exhaust valve opens, a high pressure blast (in other words, a finite-pressure wave) moves down the port and into the exhaust header pipe. This high intensity pressure wave drives gas particles in the same direction as wave motion and therefore assists the outflow of exhaust gasses. When the pressure wave reaches the end of the header pipe that's open to the atmosphere, a negative pressure wave of almost the same intensity is created and begins to travel back up the pipe toward the cylinder. Negative pressure waves move gas particles in the opposite direction as the wave, so this returning wave also assists exhaust gas outflow. When the negative pressure wave (or suction wave) reaches the cylinder, it delivers a substantial drop in pressure. If this wave arrives during the valve overlap period (when both the exhaust and intake valves are open) the pressure drop will help draw in fresh charge, a phenomenon called scavenging. The overall effects substantially boost power by 1) assisting exhaust gas outflow, 2) beginning air/fuel charge flow into the cylinder, and 3) helping to purge the cylinder of residual exhaust gasses.

Many factors contribute to optimizing these tuning effects. Two of the most important are header tubing size and length. Tubing size is really a measure of system volume and, on open headers at least, determines the restriction of internal passages. Large, free-flowing tubes produce lower pressures. Lower positive wave pressures generate lower amplitude suction waves, reducing scavenging and cylinder filling. On the other hand, smaller diameter tubing creates higher pressures that, while generating strong scavenging waves, increase restriction and pumping work. As is the case with every component category in the IC engine, the best power is produced by finding a balance between two or more counterbalancing factors. Here an optimum balance lies between the excellent scavenging effects of small tube headers vs. the reduction in restriction and pumping work produced by large tubes. The balance tilts one way or the other depending on cam timing, engine displacement, rpm, and several other factors!

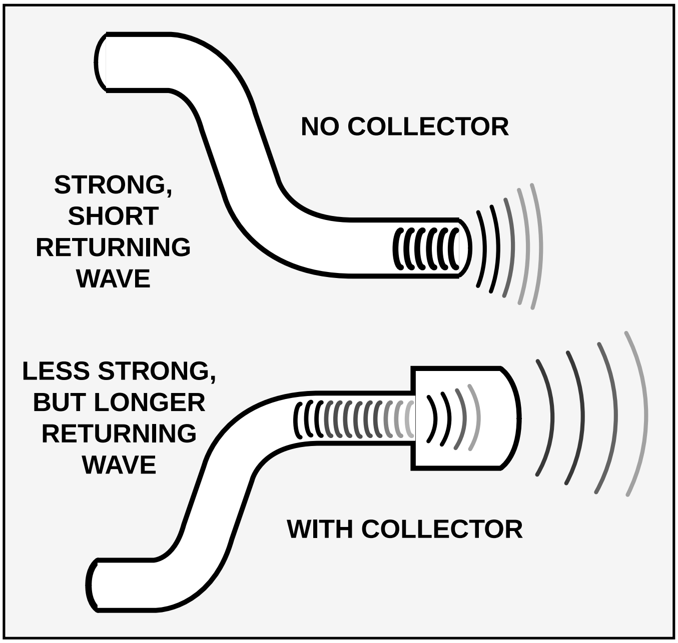
Tubing length affects when the negative suction wave arrives at the cylinder. Longer tubes delay the arrival of the scavenging wave, appropriate for lower-speed applications when more time elapses between the opening of the exhaust valve and the overlap period. Shorter tubes return a scavenging wave more quickly and "tune" at higher rpm. Once again, several counterbalancing factors must be included in the analysis. The first is cam timing. As engine speed increases, there is less time for exhaust gasses to "blowdown" from the cylinder after the exhaust valve opens. The higher residual cylinder pressures increase pumping work on the exhaust stroke. To counteract this, earlier EVO timing will assist blowdown, but it also tends to "waste" cylinder pressures that would otherwise drive the piston and generate power. At higher engine speeds, the net results show benefits from a shift toward earlier EVO timing. To adjust for this, exhaust tubing length must be increased to maintain the same scavenging wave arrival time. So again, another pair of counterbalancing factors are at work: Higher engine speeds require shorter tubing lengths, but higher speeds also require earlier EVO timing and that needs longer header tubes to maintain optimum scavenging. Peak engine power at any particular rpm is produced by a balance between these factors.

There are still more factors that affect optimum header lengths. If the primary header tubes terminate directly into the atmosphere, they generate a strong but narrow suction wave. The returning wave is so "peaky" that it only assists cylinder filling through a very narrow rpm range. To broaden the suction wave and extend the effective tuning range of the exhaust system, most headers group the primary tubes together in a larger "collector" before they open to the atmosphere. The collector produces a transition to a larger volume before the final transition to the atmosphere. This splits up or extends the width of the returning suction wave, broadening the effective rpm range during which the header system can deliver an effective scavenging wave. The diameter of the collector dictates which end of the suction wave has "emphasis." Larger collectors mimic a more direct opening to the atmo-



Negative Scavenging Wave Should Arrive During The Valve Overlap Period

Exhaust tubing length affects when the negative suction wave arrives at the cylinder. Longer tubes delay the arrival of the scavenging wave, appropriate for lower-speed applications. Shorter tubes return a scavenging wave more quickly and "tune" at higher rpm.



When primary header tubes terminate directly into the atmosphere, they generate a strong but narrow suction wave. The returning wave is "peaky" and only assists cylinder filling through a very narrow rpm range. Primary tubes terminating in a larger "collector" produce a wider suction wave, broadening the effective tuning range.

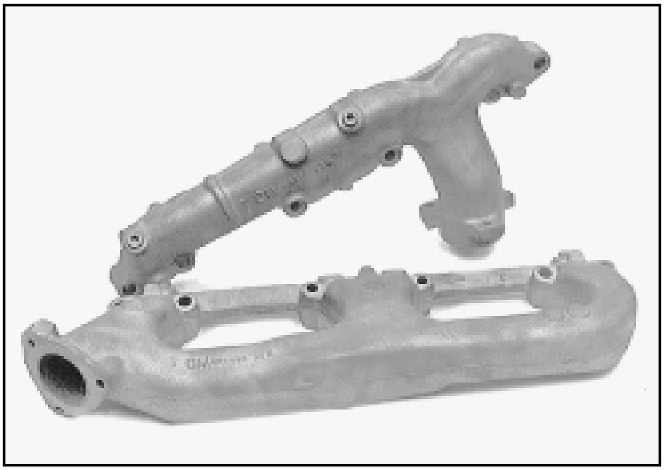
sphere, so they emphasize the initial portion of returning suction wave. Smaller collectors provide less of a transition from the individual primary tubes and tend to emphasize the trailing edge of the suction wave. The length of the collector changes the time between the leading and trailing edges of the suction wave and can also affect the optimum primary tube length.

Most header systems are designed so that the lengths of the primary tubes are nearly equal. This, too, leads to another balancing act between the higher number of bends needed to obtain equal-length primary tubes vs. the use of low-restriction straight tubes. While increased restriction almost always hurts performance, unequal primary lengths can broaden the power range and provide more usable power in racing situations. As a result, the most effective header designs use as few bends as possible to minimize restriction and relinquish equal primary lengths to a role of lesser importance.

Exhaust Menu Selections

Now that we have peeked into the wave dynamics at work inside the exhaust system, we can turn our attention to how these effects are simulated by the various choices in the Exhaust component menu. As you have discovered, the exhaust system—perhaps more than any other single part of the IC engine—is a virtual "playground" for finite-amplitude waves. You are also well aware that these interactions can be solved only by sophisticated, computationally-intensive methods that are not part of the Motion Filling And Emptying program. While flow restriction (back pressure) is accurately modeled using "pressure-drop" techniques, the effects of changes in tubing lengths and diameters that influence the flow of exhaust and induction gasses are closely tied to high-pressure wave dynamics. However, the program does use a powerful "mini-wave model" that *accurately simulates scavenging effects for three classes of headers with optimum tubing lengths and diameters*. So while the program does not resolve specific header dimensions, the model can predict engine power changes from various exhaust manifolds and headers of large and small tubing diameters (relative to the engine under test).

Stock Manifolds And Mufflers—The first choice in the Exhaust menu simulates the most restrictive exhaust system. It as-



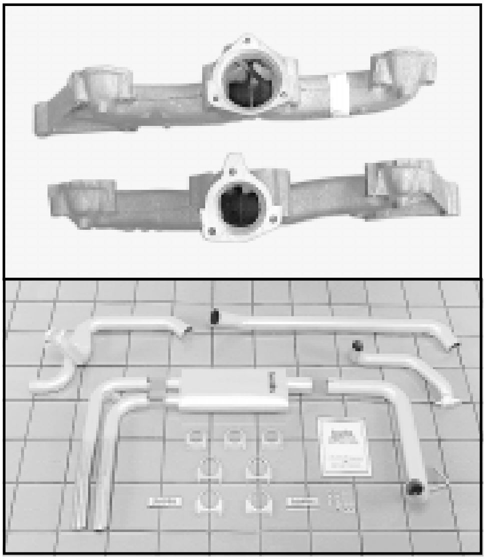
The *Stock Manifolds And Mufflers* selection assumes that the exhaust manifolds are a typical, production, cast-iron “log-type” design, where all ports connect at nearly right angles to a common “log” passage.

sumes that the exhaust manifolds are a typical, production, cast-iron, “log-type” design, where all ports connect at nearly right angles to a common “log” passage. These manifolds are designed more to provide clearance for various chassis and engine components than to optimize exhaust flow. Exhaust manifolds of this type have widespread application on low-performance production engines.

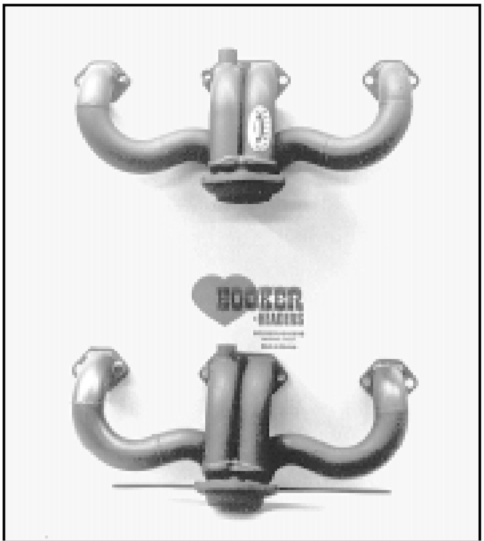
The *Stock Manifolds And Mufflers* selection assumes that the exhaust manifolds are connected to twin mufflers with short sections of pipe. Because the engine environment is a simulated dyno cell, the exhaust system terminates at the muffler outlets.

The exhaust manifolds and mufflers cancel all scavenging effects, and the system is a completely “non-tuned” design. Any suction waves that might be generated are fully damped or never reach the cylinders during valve overlap. The restriction created by this system mimics most factory muffler and/or catalytic converter with muffler combinations. Back pressure levels in the exhaust system nearly cancel the blowdown effects of early EVO timing and increase the pumping work losses during the exhaust cycle.

H.P. Manifolds And Mufflers—This choice offers a measurable improvement over the stock exhaust system modeled in the previous selection. The high-perfor-



The *H.P. Manifolds And Mufflers* choice simulates high-performance exhaust manifolds designed to improve exhaust gas flow and reduce system restriction. They are usually a “ram-horn” or other “sweeping” design with fewer sharp turns and larger internal passages, like the early Corvette manifolds pictured above. The connecting pipes to the mufflers are large diameter and the mufflers generate less back pressure. This PowerPack system from Banks Engineering for 1982 to 1991 Camaros/Firebirds is an excellent example of the low-restriction system modeled by the *H.P. Manifolds And Mufflers* choice.



Here are excellent examples of high-performance “manifolds” and free-flowing mufflers from Hooker Headers. The low-restriction manifolds fit 1992-1995 Corvettes with an LT1 engine. The Maximum-Flow mufflers are available in 2- to 3-inch inlet/outlet configurations for Ford and Chevy applications. Model these components using the *H.P. Manifolds And Mufflers* menu choice.

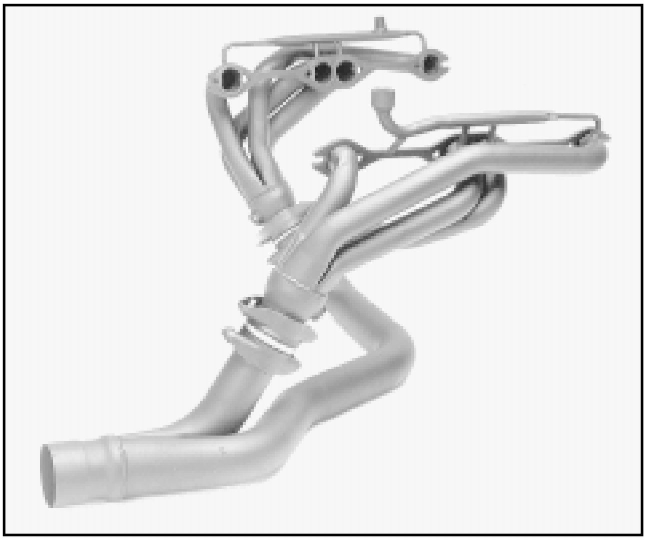
mance exhaust manifolds simulated here are designed to improve exhaust gas flow and reduce system restriction. They are usually a “ram-horn” or other “sweeping” design with fewer sharp turns and larger internal passages. The connecting pipes to the mufflers are large diameter and the mufflers generate less back pressure and, typically, produce more noise.

While this system is a “high-performance” design, it offers no tuning effects and all suction waves are fully damped or never reach the cylinders during valve overlap. All performance benefits from this selection are due to a decrease in passage restrictions and lower system back pressure. System pressure levels mimic factory high-performance mufflers and/or catalytic-converter with muffler combinations. This exhaust system may allow some benefits from early-EVO timing blowdown effects (depending on the engine component combination) and overall pumping work losses are slightly reduced by lower back pressures.

IMPORTANT NOTE FOR ALL HEADER CHOICES: Some engines, in particular, 4- or 2-cylinder applications, can develop a “full resonance” in the exhaust system—refer to the previous discussion of dual-plane manifolds for information about “full” induction system resonance. This phenomenon can derive scavenging benefits (although some studies have revealed that the benefits are relatively small) from suction waves created in the collector by adjacent cylinders. These “one-cylinder-scavenges-another” tuning techniques are not modeled in the simulation. Instead, the headers are assumed to deliver a scavenging wave only to the cylinder that generated the initial pressure wave.

Small Tube Headers With Mufflers—This is the first component selection that begins to harness the tuning potential of wave dynamics in the exhaust system. These simulated headers have primary tubes that individually connect each exhaust port to a common collector. The collector—or collectors, depending on the number of cylinders—terminates into a high-performance muffler(s). Suction waves are created in the collector, but are somewhat damped by the attached muffler. Since exact tubing lengths are not simulated, the program assumes that the primary tube will deliver the scavenging wave to the cylinder during the valve overlap period.

The primary tubes modeled by this menu selection are considered “small,” and should be interpreted to fall within a range of dimensions that are commonly associ-



While not a "true" header, this tubular exhaust system from Edelbrock for late model cars and trucks is CARB certified and offers some controlled wave dynamics for improved scavenging. The Filling-And-Emptying simulation will model these headers more accurately with mufflers than without.

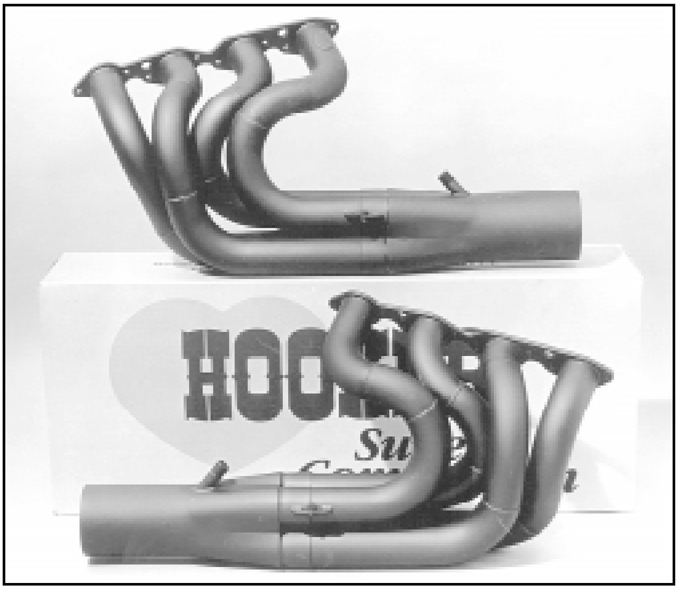
ated with applications requiring optimum power levels at or below peak-torque engine speeds. These headers typically show optimum benefits on smaller displacement engines (such as "smallblocks"), and may produce less power on large displacement engines. The following rules of thumb give approximations of tubing diameters used by the simulation: Headers with tubes that measure 95% to 105% of the exhaust-valve diameter are considered "small" for any particular engine; tubes that measure 120% to 140% of the exhaust-valve diameter are "large" tube headers.

Small-Tube Headers Open Exhaust—This menu selection simulates headers with "small" primary tubes individually connecting each exhaust port to a common collector. The collector—or collectors, depending on the number of cylinders—terminates into the atmosphere. Strong suction waves are created in the collector that provide a substantial boost to cylinder filling and exhaust gas outflow. Since exact tubing lengths are not simulated, the program assumes that the primary tube will deliver the scavenging wave to the cylinder during the valve overlap period.

The primary tubes modeled by this menu selection are considered "small," and should be interpreted to fall within a range of dimensions that are commonly associated with applications requiring optimum power levels at or slightly above peak-torque engine speeds. These headers typi-

Typical small-tube headers are usually designed with high-performance street applications in mind. The better pieces have 2-1/2-inch collectors and 1-1/2- or 1-5/8-inch primary tubes. They are made from heavy-gauge tubing that will withstand years of use, and do not necessarily have equal-length tubes.

Large-tube stepped headers have large-diameter primary tubes with several transitions to slightly larger tubing diameters. These "steps" can reduce pumping work and improve horsepower on large displacement and/or high-rpm applications. These Hooker Pro-Stock BB Chevy headers have 2-3/8-inch primary tubes that step to 2-1/2-inch by the time they reach the 4-1/2-inch collectors.



cally show benefits on smaller displacement engines but may produce less power on large-displacement big-block engines. The following rules of thumb should give a reasonable approximation of tubing diameters used in the simulation: Headers with tubes that measure 95% to 105% of the exhaust-valve diameter are considered "small" for any particular engine; tubes that measure 120% to 140% of the exhaust-valve diameter are "large" tube headers.

Large-Tube Headers With Mufflers—This menu selection simulates headers with "large" primary tubes individually connecting each exhaust port to a common collector. The collector—or collectors, depending on the number of cylinders—terminates into a high-performance muffler(s). Suction waves are created in the collector, but are somewhat damped by the attached muffler. Since exact tubing lengths are not simulated, the program assumes that the primary tube will deliver the scavenging wave to the cylinder during the valve overlap period.

The primary tubes modeled by this menu selection are considered "large," and should be interpreted to fall within a range of dimensions that are commonly associated with applications requiring optimum power at peak engine speeds. These headers typically show benefits on high-rpm racing smallblocks or large displacement big-block engines. These headers may produce less power on small-displacement engines operating in the lower rpm ranges. The following rules of thumb should give a reasonable approximation of tubing diameters used in the simulation: Headers with tubes that measure 95% to 105% of the exhaust-valve diameter are considered "small" for any particular engine; tubes that measure 120% to 140% of the exhaust-valve diameter are "large" tube headers.

Large-Tube Headers Open Exhaust—This menu selection simulates headers with "large" primary tubes individually connecting each exhaust port to a common collector. The collector—or collectors, depending on the number of cylinders—terminates into the atmosphere. Strong suction waves are created in the collector that provide a substantial boost to cylinder filling and exhaust gas outflow. Since exact tubing lengths are not simulated, the program assumes that the primary tube will deliver the scavenging wave to the cylinder during the valve overlap period.

The primary tubes modeled by this menu selection are considered "large," and should be interpreted to fall within a range of dimensions that are commonly associated with applications requiring optimum power at peak engine speeds. These headers typically show benefits on high-rpm racing smallblocks or large displacement big-block engines. These headers produce less power on small-displacement engines, particularly those operating in the lower rpm ranges. The following rules of thumb should give a reasonable approximation of tubing diameters used by the simulation: Headers with tubes that measure 95% to 105% of the exhaust-valve diameter are considered "small"; tubes that measure 120% to 140% of the exhaust-valve diameter are "large" tube headers.

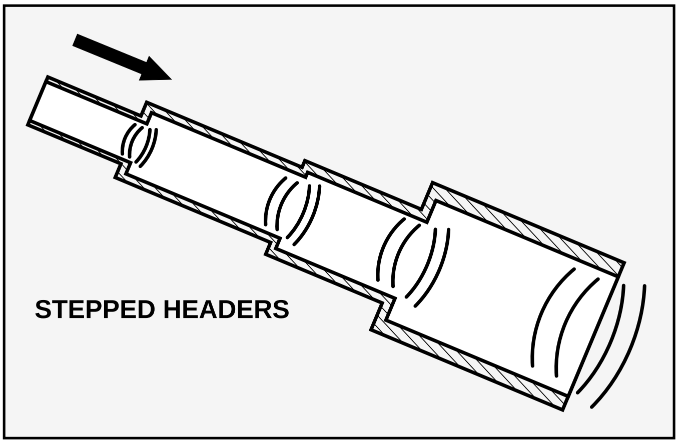
Large Stepped-Tube Race Headers—This menu selection simulates headers with "large" primary tubes individually connecting each exhaust port to a common collector. Each primary tube has several transitions to slightly larger tubing diameters as it progresses towards the collector. These "steps" can reduce pumping work and improve horsepower as described below. The collector—or collectors, depending on the number of cylinders—terminates into the atmosphere. Strong suction waves are created in the collector that provide a substantial boost to cylinder filling and exhaust gas outflow. Since exact tubing lengths are not simulated, the program assumes that the primary tube will deliver the scavenging wave to the cylinder during the valve overlap period.

The "stepped" design of the primary tubes can reduce pumping work on some engines. As the high-pressure compression wave leaves the port and encounters a step in the primary tube, it returns a short-duration rarefaction wave. This low-pressure "pulse" moves back up the header and assists the outflow of exhaust gasses. When the rarefaction wave reaches the open exhaust valve, it helps depressurize the cylinder and lower pumping work. This can generate a measurable increase in horsepower on large displacement and/or high-rpm engines.

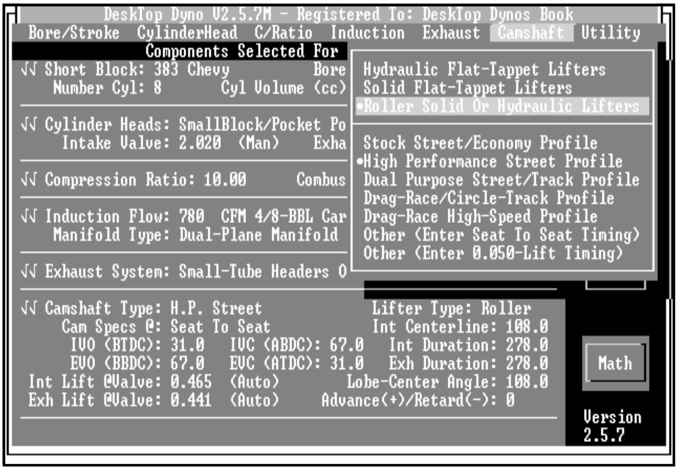
The primary tubes modeled by this menu selection are considered "large," and should be interpreted to fall within a range of dimensions that are commonly associated with applications requiring optimum power at peak engine speeds. The following rules of thumb should give a reasonable approximation of tubing diameters: Headers with tubes that measure 95% to 105% of the exhaust-valve diameter are considered "small" for any particular engine; tubes that measure 120% to 140% of the exhaust-valve diameter are "large" tube headers.

The Camshaft Menu

The final component menu allows the selection of the single most important part in the IC engine: the camshaft. For many



"Stepped" headers can reduce pumping work in some engines. As the high-pressure compression wave leaves the port and encounters a step, it returns a short duration expansion wave. These low-pressure "pulses" move back up the header and assists the outflow of exhaust gasses. This can generate a measurable increase in horsepower on engines that are suffering from substantial pumping-work losses, such as large-displacement, high-rpm, drag-racing engines.



The *Camshaft* menu is the final component menu. With it you select cam profiles for the simulated engine. The following pages in this chapter provide essential information about cam design and program assumptions that will help you use this powerful feature of Motion's Filling-And-Emptying simulation software.

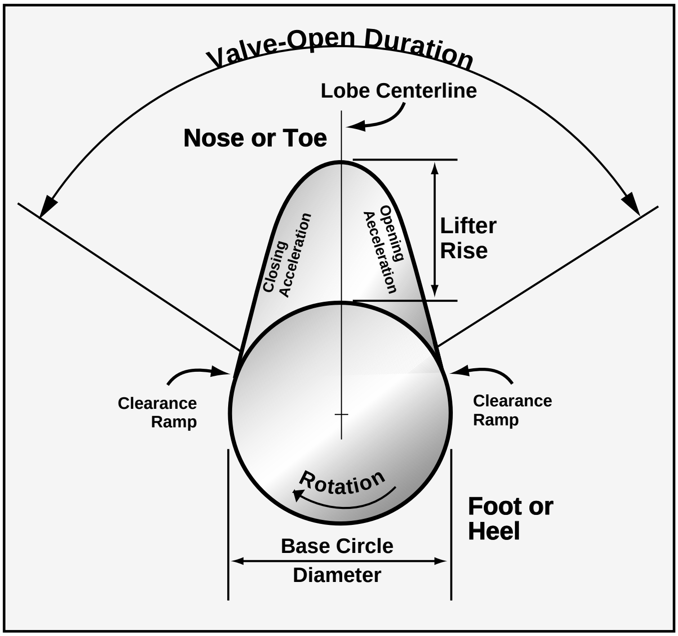
enthusiasts and even professional engine builders, the subtleties of cam timing defy explanation. The reason for this confusion is understandable. The camshaft is the "brains" of the IC engine, directing the beginning and ending of all four engine cycles. Even with a good understanding of all engine systems, the interrelatedness of the IC engine can make the results of cam timing changes read like a mystery story. In many cases there are only two ways to determine the outcome of a modification: 1) run a real dyno test or 2) run a simulation. Since the camshaft directly affects several functions at once, e.g., exhaust and intake scavenging, induction signal, flow efficiency, cylinder pressures, etc., using a computer-based engine simulation program is often the only way to <u>predict</u> the outcome.

Motion Software's Filling-And-Emptying simulation makes it possible to test the effects of cam timing in seconds. The ability of the program to take multiple elements into consideration and "add up the effects over time" is key to analyzing the effects of camshaft timing changes. Unfortunately, even with such a powerful tool at hand, the <u>subject</u> of cam timing cannot be made simple. The camshaft is a component that will thoroughly test your knowledge and comprehension of the IC engine. Review this section in light of what has been discussed earlier, and we hope you'll find a deeper understanding of camshaft operation and new insight into IC engine function.

Cam Basics

In the simplest terms, the camshaft is a straight steel or iron shaft with eccentric lobes. It is connected to the crankshaft with a chain or gear train and is usually rotated at one-half crank speed. Lifters (or cam followers)—and in the case of in-block cam locations, pushrods, and rockerarms—translate the rotary motion of the cam into an up-and-down motion that opens and closes the intake and exhaust valves. This entire assembly must function with high precision and high reliability. Street engines driven hundreds-of-thousands of miles operate their valvetrain components *billions of cycles*. If the overall camshaft and valvetrain design is good, a precision micrometer will detect only negligible wear.

The camshaft controls the valve opening and closing points by the shape and rotational location of the lobes. Most cams are ground to a precision well within one crankshaft degree, ensuring that the valves actuate exactly when intended. Timing variations of a few degrees can develop in the cam drive, especially in chain-drive systems, but racing gear drives reduce variations to within one or two crank degrees of indicated timing. Camshaft lobes also determine how far the valves will lift off of the valve seats by the height of the lobes (heal to toe height) and the multiplying ratio of the rockerarms (if used). The



Camshaft terminology can be confusing, so here's the low-down. To start off, the camshaft is a round shaft incorporating cam *lobes*. The *base circle diameter* is the smallest diameter of the cam lobe and is shaped perfectly round. *Clearance ramps* form the transition from the round base circle to the *acceleration ramps*. As the cam turns, the lifter smoothly accelerates up the *clearance ramp* and continues to rise as it approaches the *nose*, then begins to slow to a stop as it reaches maximum *lift* at the *lobe centerline*. Maximum *lifter rise* is determined by the height of the *toe* of the cam lobe over the base circle diameter. The lifter then accelerates in the closing direction and when the valve approaches its seat, the lifter is slowed down by closing clearance ramp. *Valve-open duration* is the number of crankshaft degrees that the valve or lifter is held above a specified height by the cam lobe (usually 0.006-, 0.020-, or 0.050-inch). A symmetric lobe has the same lift curve on both the opening and closing sides; an asymmetric lobe is shaped differently on each side of the lobe. A single-pattern cam has the same profile on both the intake and exhaust lobes; a dual-pattern cam has different profiles for the intake and exhaust lobes.

rates at which the valves are accelerated open and then returned to the seats are also "ground into" cam lobe profiles. Only a limited range of contours will maintain stable valve motion, particularly with high-lift, racing profiles. Unstable profiles or excessive engine speed will force the valvetrain into "valve float," leading to rapid component failure.

Visualizing And Calculating Valve Events

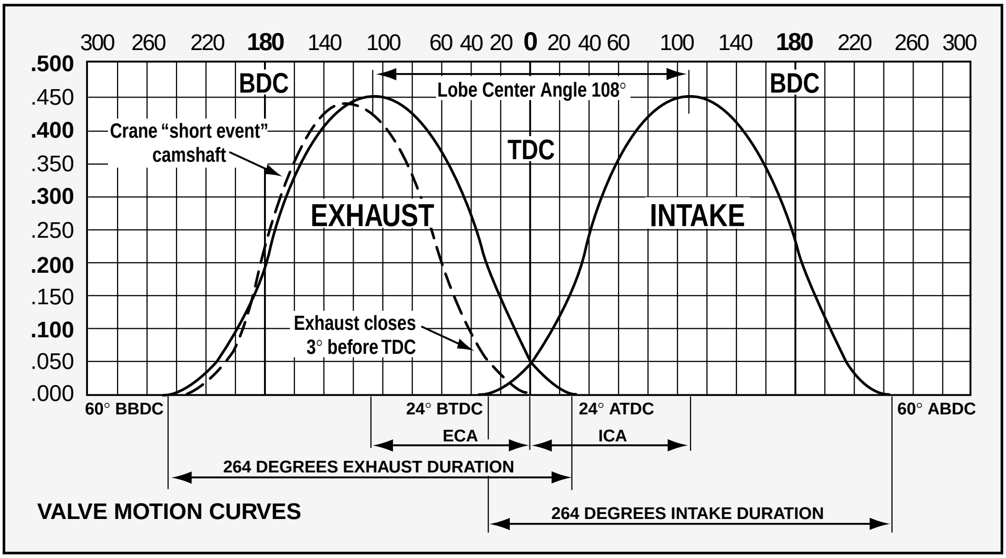
There are six basic cam timing events ground into the lobes of every camshaft. These timing points are:

1—Intake Valve Opening (IVO)
2—Intake Valve Closing (IVC)
3—Exhaust Valve Opening (EVO)
4—Exhaust Valve Closing (EVC)
5—Intake Valve Lift
6—Exhaust Valve Lift

These six points can be "adjusted" somewhat (we'll discuss which and how cam timing events can be altered in the next section), but for the most part they are fixed by the design of the cam. Other timing numbers are often discussed, but they are always derived from these basic six events. These derivative events are:

7—Intake Duration
8—Exhaust Duration
9—Lobe Center Angle (LCA)
10—Valve Overlap
11—Int. Center Angle or Centerline (ICA)
12—Exh. Center Angle or Centerline (ECA)

The first four basic timing points (1 thorough 4) pinpoint the "true" beginning and end of the four engine cycles. These valve opening and closing points indicate when



The best way to visualize camshaft timing is with this "twin-hump" event drawing. It depicts the valve-motion curves for the exhaust lobe on the left and the intake lobe on the right, locating the valve overlap period and TDC at the center. If you become sufficiently familiar with this drawing so that you can easily picture it in your mind, you will be able to quickly evaluate any cam timing specs and visualize how they relate to one another.

the function of the piston/cylinder mechanism changes from intake to compression, compression to power, power to exhaust, and exhaust back to intake. What could be more important from the standpoint of understanding engine function and performing engine simulations?

Compared to the basic timing events, many simulation experts believe that most of the second six timing values are not only unimportant, they actually "blow smoke" over the whole issue of cam timing analysis. Naturally, four of the <u>second</u> six events are the most publicized by the cam manufacturers and enthusiast magazines. This unfortunate situation evolved over many years of selling and marketing camshafts in the automotive aftermarket. Long before engine simulations were widely used and a good understanding of the relationship between the individual valve events and engine power ever existed, the descriptions of the *distance <u>between</u> valve events rather than the valve events themselves* became a standard measure of a camshaft.

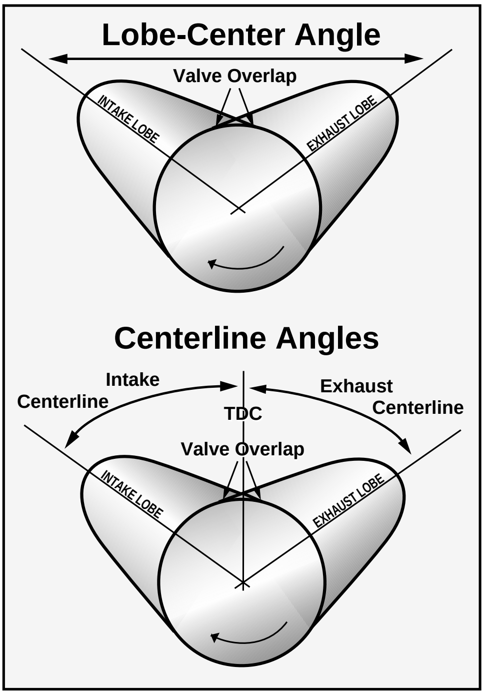
Since this assortment of cam specifications is used almost interchangeably, it is almost impossible to "talk camshaft" without a good understanding of all these terms. Probably the best way to organize and visualize this nomenclature is to picture the common "twin-hump" event drawing. This illustration depicts valve-motion curves (sometimes called valve displacement curves) for the exhaust lobe on the left and the intake lobe on the right, locating the valve overlap period and TDC (top dead center) at the center of the graph. If you become sufficiently familiar with this drawing so that you can easily picture it in your mind, you should be able to figure out any of the cam timing specs and how they relate to one another.

The graph plots crank degrees on the X-axis (left to right) and valve lift on the Y-axis (up and down). The width of the graph is slightly shortened. Since no valve motion occurs during about 200 degrees of crank rotation when the "true" compression and power strokes take place, the graph chops off 60-degrees on each end, running from 300-degrees before to 300-degress after TDC (total four-cycle sweep is actually 720 degrees).

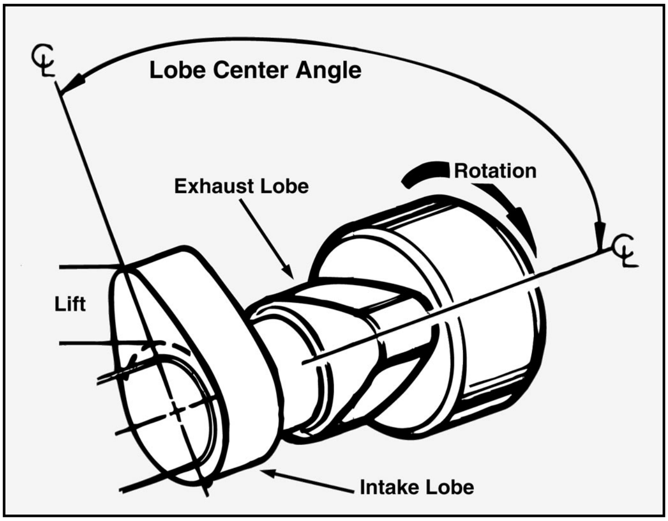
Picture events on the graph taking place from left to right, passing through TDC at the center. The left "hump" is the exhaust valve motion curve. Focus your attention at the 240-degree point before TDC. Note that the 180-degree vertical line is actually a BDC (bottom dead center) marker, so 240-degrees before TDC lies 60-degrees before BDC. This is the EVO (exhaust valve opening) timing point, the beginning of exhaust blowdown and rapid cylinder decompression. Follow the exhaust valve curve through its maximum lift—about 0.450-inch—that occurs at 108-degrees before TDC to its closing point at 24-degrees after TDC. Now notice the intake valve motion curve. At 24-degrees before TDC, when the exhaust valve is still open, the intake valve leaves its seat and begins to follow its motion curve. During the space between 24-degrees before TDC and 24-degrees after TDC, both valves are open, defining the overlap period when exhaust scavenging can take place. Now continue to follow the intake motion curve to its maximum lift—also about 0.450-inch—at 108-degrees after TDC. The intake closing point occurs at 240-degrees after TDC or 60-degrees after BDC (the 180-degree line). You have now traced out the six basic cam timing points. Let's see how these relate to the six derivative events.

First consider duration—the number of crank degrees that the valves are off their seats. This "length of time," expressed in crank degrees, is a measure of the true intake and exhaust cycles. The exhaust valve opens 60-degrees before BDC and closes 24-degrees after TDC, and since 180 degrees of crank rotation exists between BDC and TDC, the exhaust duration is: 60 + 180 + 24 = 264 degrees off-seat duration. In the case of the intake valve, valve opening occurs 24-degrees before TDC and closing 60-degrees after BDC, so the intake duration is: 24 + 180 + 60 = 264 degrees. Duration is easy to calculate from the opening and closing points; however, watch out for the one "tricky" element: *short events*. For cams that open the intake valve *after TDC* instead of before TDC or close the exhaust valve *before TDC* instead of after TDC (common

Lobe Center Angle **is the angle measured in <u>camshaft</u> degrees (multiply by two for equivalent crankshaft degrees) between the maximum-lift points on the intake and exhaust lobes for the same cylinder. The lobe center angle is "ground" into the cam when it is manufactured and cannot be changed (unless the cam is reground). As the lobe-center angle is decreased, the valve overlap period (when both intake and exhaust valves are open) is increased. The *lobe centerline angles* are the angles measured in <u>crankshaft</u> degrees between the points of maximum lift on the intake and exhaust lobes and Top Dead Center. These values are determined by the "indexing" of the cam to the crankshaft.**



The Lobe Center Angle describes *the angular distance between the centers of the intake and exhaust lobes as viewed on the cam itself*. This distance is a measurement of the cam—not relative to TDC—and is the only cam spec that cannot be altered regardless of how the cam is "degreed" with crankshaft. Because of this, the LCA is measured in <u>cam degrees</u>.



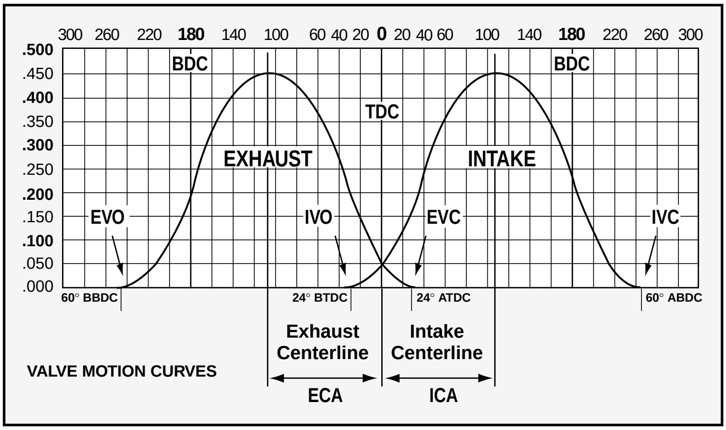
with 0.050-timing specs), make sure to <u>subtract</u> the timing point instead of adding it to calculate the duration. For example, a Crane Cams, Inc. profile HMV-272-2-NC opens the exhaust valve 51-degrees before BDC and closes it 3-degrees *before TDC*. To calculate the duration subtract the "short" closing point: 51 + 180 –3 = 228 degrees. While these "short" timing events are much more common when working with 0.050-inch timing specs, you may also come across seat-to-seat timing specs for emissions-restricted camshafts with "short" events, designed to minimize or eliminate valve overlap. In a nutshell, the duration is simply the number of crank degrees swept out by the lift curve.

Next, let's look at the three centerline angles: Intake Centerline, Exhaust Centerline, and Lobe Center Angle (sometimes called Lobe Centerline). The terms "Centerline" and "Center Angle" are methods of describing the distance to or from the exact center of a lobe. The Intake and Exhaust Centerlines describe the distance from the center of the Intake and Exhaust lobes to TDC. Both of these timing specs are measured in crank degrees. That means that the number of degrees the crank rotates from the point at which the piston rests at TDC until the lifter contacts the exact center of the intake lobe is the Intake Centerline. When the lobes are symmetric, that is they have the same shape and valve motion rates on the opening and closing sides, the Intake Centerline and Exhaust Centerlines will occur at the point of maximum valve lift. Asymmetric profiles are quite common, although, the amount of asymmetry typically is very small, so the centerlines should still fall within two or three of degrees of maximum lift point.

All but one of the basic and derivative cam-timing specs are measured in crank degrees. This makes sense since the timing specs fundamentally describe valve positions (and durations) as they relate to piston positions (and piston movement). This applies to every timing spec but Lobe Center Angle (LCA). The LCA is meant to describe *the angular distance between the centers of the intake and exhaust lobes as viewed on the cam itself*. This distance is a cam-specific measurement, not relative to TDC, and is the only cam spec that cannot be altered regardless of how the cam is "degreed" with crankshaft. It is said to be "ground into" the cam. Because this spec relates one lobe position to the other, independent of the engine or crankshaft, it is measured in <u>cam degrees</u>. The number of <u>crank degrees</u> between the center of the lobes is twice the LCA.

All of the camshaft specifications described thus far tell something about how the cam is manufactured and how it should be installed in the engine. To help keep them straight in your mind, let's reorganize

This group of valve events are all measured from TDC or BDC piston positions and are dependent on how the cam is installed, or indexed, in the engine.



them in two new groups. The first group contains all the timing specs that are measured from TDC or BDC piston positions and are dependent on how the cam is installed, or indexed, in the engine:

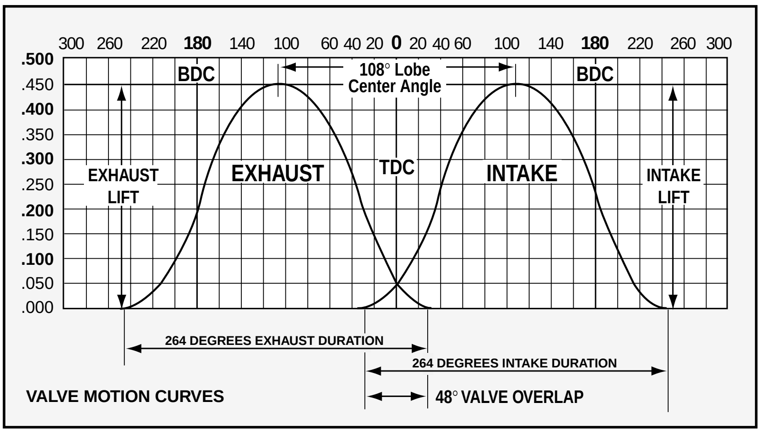
1—Intake Valve Opening (IVO)
2—Intake Valve Closing (IVC)
3—Exhaust Valve Opening (EVO)
4—Exhaust Valve Closing (EVC)
5—Int. Center Angle or Centerline (ICA)
6—Exh. Center Angle or Centerline (ECA)

Each of these timing specs indicate a opening, closing, or lobe centerline point measured from a TDC or BDC piston position. If the cam is installed in an advanced or retarded position relative to TDC, the value of these timing specs will change. In fact, advancing and retarding the cam is one way to change cam timing that we'll discuss in detail later in this guide.

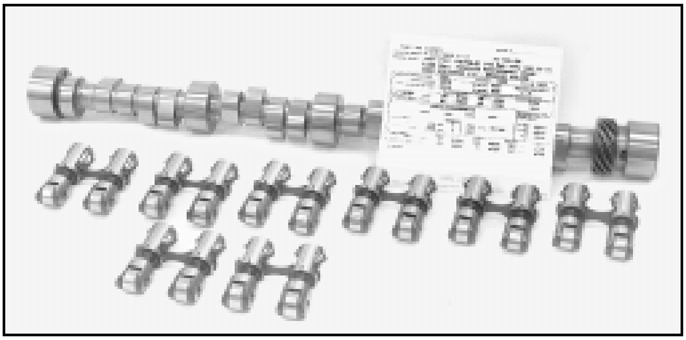
The following group of cam timing events and specs at dependant on TDC or crankshaft position:

7—Intake Duration
8—Exhaust Duration
9—Lobe Center Angle (LCA)
10—Valve Overlap
11—Intake Valve Lift
12—Exhaust Valve Lift

Each of these six terms <u>compare one cam spec to another</u>; they are not measured relative to any fixed crank position. Refer again to the valve-motion plot to confirm this. Remember that the duration of each lobe is the distance in crank degrees between the valve opening and closing points. Sliding these curves left or right (analogous to advancing or retarding the cam) does not change the distance <u>between</u> the opening and closing points and does not change duration. Duration is, therefore, another "ground in" cam spec. The same



These cam timing events <u>compare one cam spec to another</u> and are not dependant on TDC or crank position.



Before engine simulations were widely used, cam manufacturers established a methodology for identifying and classifying camshafts. Unfortunately, these "catalog" specs place the emphasis on the span between the valve events rather than on the events themselves.

limitation applies to valve overlap. The lobes are spaced by a fixed LCA, and they can't move with respect to each other, so the length of the overlap period—the distance between EVC and IVO at a specific lifter rise—never changes for a particular camshaft. Finally, valve lift is another comparison of cam specs. In this case, it's the distance between the lobe heal height and toe height multiplied by the rocker ratio. Valve lift is measured in inches (or millimeters) and is never related to crank angle or TDC.

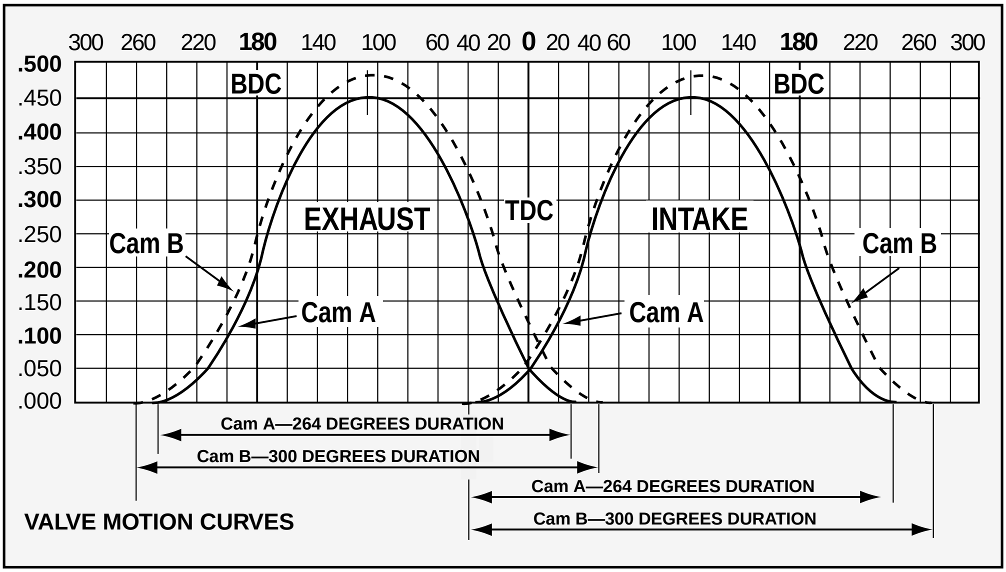
Take some time to review the relationships between all of the cam specs and the valve motion drawings. Trace the action of the valves from left to right through the exhaust cycle and, as TDC approaches, through overlap, then continue to the right through the intake cycle. These drawings give an excellent mental "picture" of the relationships between LCA and the individual ICA and ECA values. In fact, assuming a symmetric profile, it is possible to calculate all of the center angles, durations, and the overlap from the four basic valve events (EVO, EVC, IVO, IVC). It is also possible to calculate the four valve events from the duration, the LCA, and either the ICA or the ECA. We'll discuss the details of converting timing specifications using information from cam manufacturer's catalogs later in this guide.

How Valve-Event Timing Affects Power

As we mentioned earlier, years of marketing efforts by cam manufacturers have established an accepted methodology for identifying and classifying camshafts. The specifications most commonly listed by manufacturers are:

1—Intake Duration
2—Intake Valve Lift
3—Exhaust Duration
4—Exhaust Valve Lift
5—Lobe Center Angle (LCA)
6—Intake Center Angle or Centerline (ICA)

Long before engine simulations were widely used and designers gained an understanding of how the changes in valve-event timing affect power, "manufacturer's catalog" specs became a standard measure of cam profiles. Unfortunately, these terms place the emphasis on the span between the valve events rather than on the events themselves. For example, it is common to compare two cams by comparing their intake and/or exhaust valve-open duration. While duration does point to the intended use for the cam, it doesn't indicate the valve events, making it difficult to predict engine performance. If **Cam A** has 264 degrees of exhaust duration and **Cam B** has 300 degrees, the longer duration spec doesn't give a clue about how the additional valve-open timing will be allocated to the opening and closing events. Are the entire forty degrees added to **Cam B**'s valve opening point; are they added to the closing point; or are they split between the two in some proportion? Without knowing the exact valve events, one can only guess at the outcome. A more critical situation exists for engine simulation programs: Without knowing the exact valve events, a simulation isn't even possible! Not using exact valve timing events during a simulation is like building and testing an engine



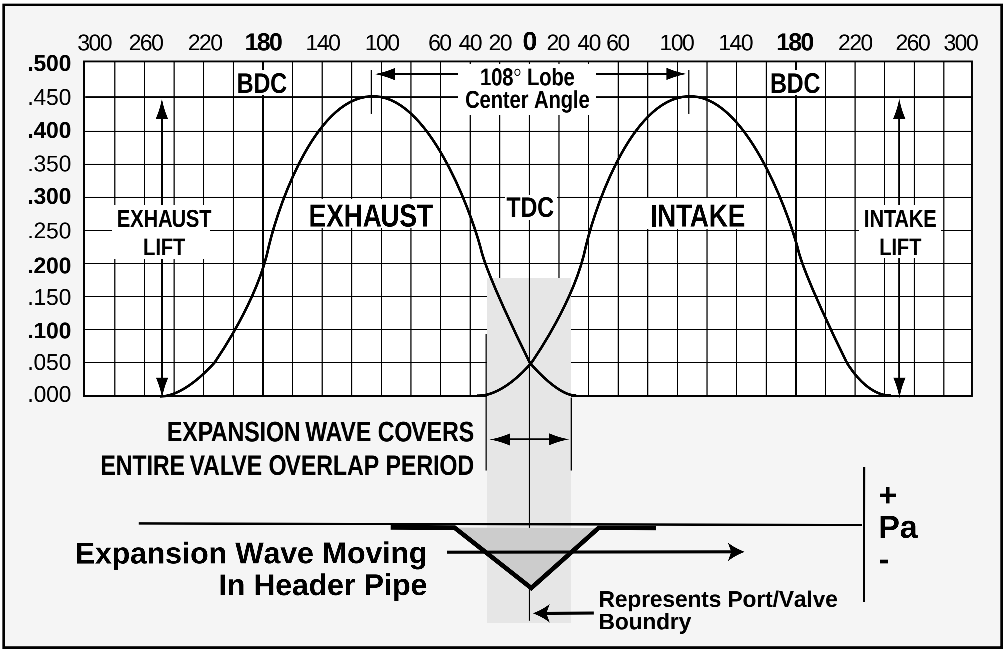
A manufacturer's catalog lists *Cam A* as having 264 degrees of duration while *Cam B* has 300 degrees. But just the duration spec doesn't give a clue about how the additional valve-open timing will be allocated to the opening and closing events. These valve motion curves indicate just one of an infinite number of possibilities. Without knowing the exact valve events, running an engine simulation isn't possible.

without defining when the valves open and close; the whole concept doesn't make any sense.

The emphasis on event timing in engine simulation programs has driven many leading-edge designers to discount what they now term as ambiguous or less-useful cam timing specifications, in particular, advance and retard figures, centerlines, and durations. For some, this may be a difficult paradigm shift, but the rewards are substantial: you may find a new understanding of the IC engine at the end of your efforts. The next few paragraphs delve into the effects of individual valve events learned from both real-world and simulated testing. We'll also relate these basic timing events to other popular cam specifications, since it will be many years—if ever—before some of the less-relevant specs disappear from cam manufacturer's catalogs.

The four basic valve-event timing points (EVO, EVC, IVO, IVC) can be grouped into three categories based on their influence on engine performance: 1) EVC and IVO are the least important individually, but together comprise the overlap period that has a significant effect on power, and 2) the EVO is the next most important timing point since is determines the beginning of the exhaust cycle and cylinder blowdown, and 3) IVC is the most critical since it fixes the balance between cylinder filling and intake reversion, each having a potent effect on engine output.

EVC/IVO, The Valve Overlap Period—The valve overlap period occurs as the piston passes through TDC after the main portion of the exhaust stroke. The intake valve opens before TDC (usually) and signals the beginning of period of time during which both intake and exhaust valves are off their seats. As we found in our discussion of exhaust systems, a high pressure wave produced when the exhaust valve opens (EVO) travels to the end of the header and returns a strong negative pressure wave that delivers a pressure drop at the exhaust valve. If this pressure drop arrives during the overlap period, it will help purge the cylinder of exhaust gasses and,



The goal of the engine designer is to "cover" the overlap period with the arriving scavenging wave during as wide an rpm range as possible. When this is done, the exhaust system and the engine are said to be "in tune" during that range of engine speeds. *If any part of overlap does not coincide with the presence of a low-pressure scavenging wave, reversion or reduced cylinder filling will drive the engine partially out of tune.*

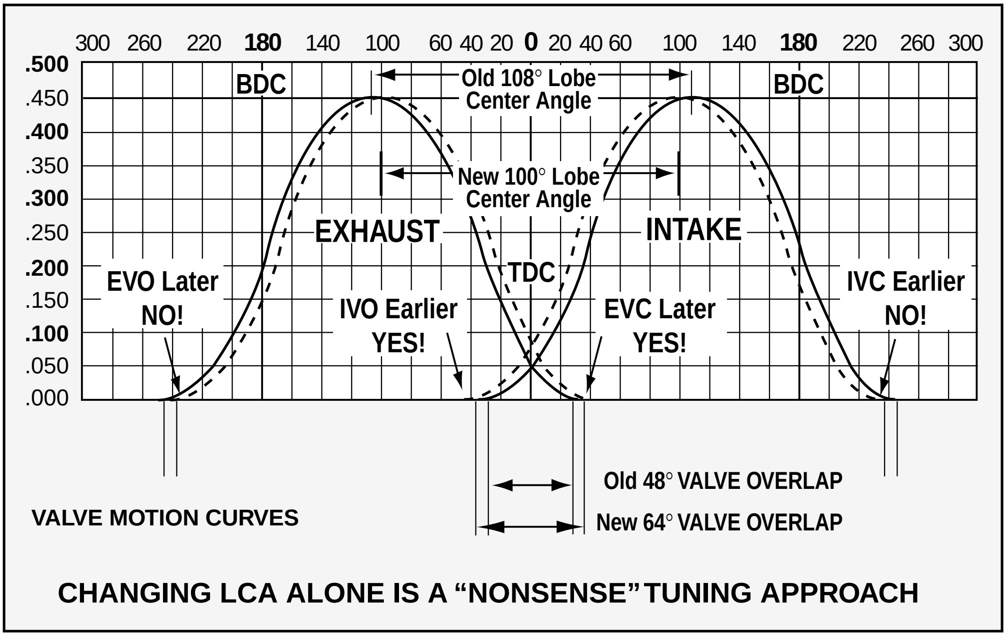
despite the upward movement of the piston, begin the inflow of fresh charge from the induction system. This phenomenon is called scavenging, and an engine that delivers its scavenging wave during the overlap period is said to be "in tune." If the pressure wave is early or late, or not even created by exhaust system, the piston rising in the bore during the first part of overlap will force exhaust gasses into the induction system, producing a phenomenon called "reversion." When this occurs, cam timing and the exhaust system are "out of tune."

Reversion is a power killer. When exhaust gasses are driven into the induction system, they force air/fuel mixtures back upstream. Severe reversion can drive the air/fuel charge out of the air inlet, creating a "standoff" of vapors above the carburetor. When the charge is drawn back into the engine, it is re-atomized with fuel creating "double-rich" mixtures. Additional, less severe, symptoms of reversion include a drop in manifold vacuum, rough idle and/or high idle speeds, and a substantial increase in emissions from over-rich mixtures.

An engine that develops reversion and runs poorly at lower engine speeds may run fine at higher speeds. In fact, most race engines exhibit these symptoms; the common side effects of a high-speed race tune. The scavenging wave that missed the overlap period at low speeds may return on time at higher engine speeds, optimizing tuning, cylinder filling, and power output. The nature of the exhaust system is to create in-and-out-of-tune conditions as the engine moves through its rpm range. The goal of the engine designer is to broaden the arrival of the scavenging wave as much as possible (remember our discussion of header-pipe collectors) "covering" the overlap period through as much of the max-power rpm range as possible. *If* *any part of overlap does not coincide with the presence of a low-pressure scavenging wave, reversion or reduced cylinder filling will drive the engine partially out of tune.*

Valve overlap periods vary from zero to about 40-degrees on street and basic performance engines. All-out racing engines use overlap periods as long as 120 degrees. Engines with long overlap tend to have very "peaky" power bands, since a returning scavenging wave "covers" the valve overlap period through only a relatively narrow rpm range. However, wide overlap periods are needed to effectively scavenge large-displacement engines at high engine speeds. Shorter overlap periods, on the other hand, stay in-tune through a wider rpm range, but tend to reduce peak horsepower because they limit scavenging. The number of overlap degrees is a good indicator of the intended use of the cam: short overlap for broad torque and good low-speed power, long overlap for high power at high engine speeds.

Lobe Center Angle and Overlap—Since valve overlap is directly related to the Lobe Center Angle (LCA)—refer again to the valve motion diagram—is has become commonplace to discuss changes in valve overlap as being synonymous with changes in LCA. In fact, some believe that changing LCA (requires regrinding cam) is the correct method of adjusting overlap. This is not true. This common misconception comes from an inadequate understanding of the function and effects of the individual valve-event timing points. For example, suppose that you wish to increase the valve overlap period for a particular camshaft. Decreasing the LCA will, in fact,



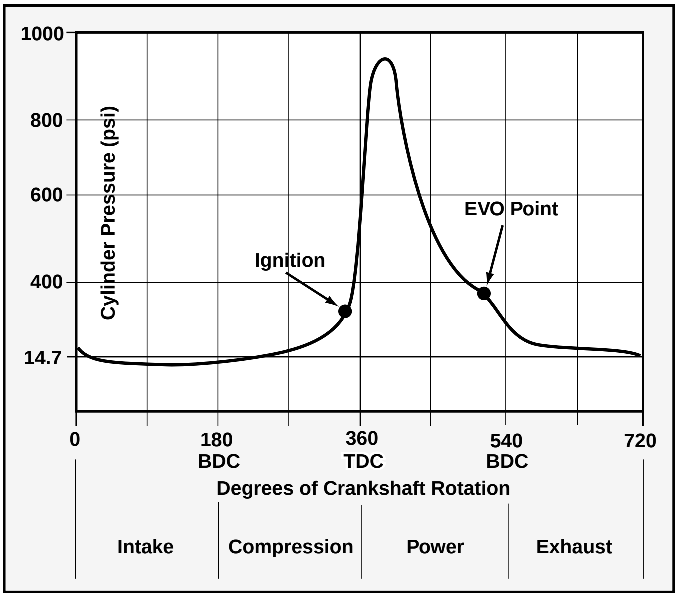
Decreasing the LCA will directly increase valve overlap by moving IVO earlier and EVC later. However, if the goal is to boost high-speed horsepower (the reason for additional overlap), narrowing the LCA also moves the IVC earlier. This <u>reduces</u> "ram effects" in the induction system at higher engine speeds, clearly the wrong approach for performance. In addition, a narrower LCA opens the exhaust valve (EVO) later and that <u>delays cylinder blowdown</u>, an effect that tends to boost low-speed—not high-speed—power. The correct method of adjusting overlap, and every other cam timing, should be to apply the appropriate changes to the individual valve opening and closing events.

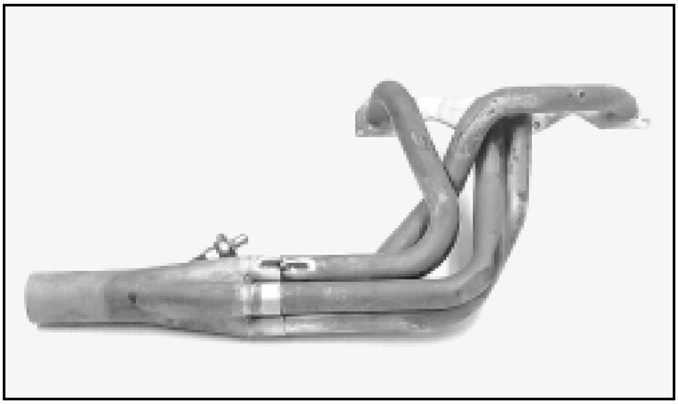
directly increase the overlap, but let's look at the consequences. If the goal is to increase overlap, the need must be to boost high-speed horsepower. With a smaller LCA, the EVC occurs later and IVO is earlier, both of these changes directly increase overlap and tend to boost high speed power. So far, so good. However, changing the LCA has the effect of "rotating" the lobes on the camshaft and moving the IVC earlier. Since later IVC timing tends to take advantage of the "ram effects" in the induction system at higher engine speeds, closing the intake valve earlier is clearly the wrong approach. In addition, with a narrower LCA, the EVO occurs later and that delays cylinder blowdown, another effect that tends to boost low-speed power. Again, the wrong approach. This "schizophrenic" or nonsensical tuning approach applies the correct EVC and IVO timing, but counteracts these effects with earlier IVC and later EVO timing; the net results are often a modest shift in tuning toward higher engine speeds. The correct method of adjusting overlap, and every other cam timing alteration, should be to apply the appropriate changes to the individual valve opening and closing events. In this case, a longer overlap should be produced by applying an earlier IVO and EVO and a later IVC and EVC. Yes, this also increases duration, but applying individual valve-event changes that complement—rather that counteract—each other, produce a much more effective camshaft for the intended purpose.

EVO, The Exhaust Valve Opening Point—EVO timing is the next most important of the basic valve events. Changes in this timing point can have a substantial impact on engine efficiency and performance. The opening of the exhaust valve creates a high-pressure wave that travels through the exhaust system that, with properly designed headers, returns as a strong scavenging wave during the overlap period. EVO timing must be coordinated with header tubing length, engine speed, and valve overlap to obtain full benefit from scavenging. The EVO point also signals the beginning of the blowdown phase of the exhaust cycle by starting the rapid decompression of the cylinder at the end of the power stroke.

The optimum EVO timing point is another "moving target" for the engine designer. It should be no surprise that a perfectly timed EVO simply doesn't exist. However, if perfection were possible, an ideal EVO would delay the opening of the exhaust valve until the piston reached BDC, allowing the engine to harness every last

EVO signals the beginning of the blowdown phase by starting the rapid decompression of the cylinder near the end of the power stroke. The opening of the exhaust valve also creates a high-pressure wave that travels through the exhaust system that, with properly designed headers, returns as a strong scavenging wave during the overlap period.





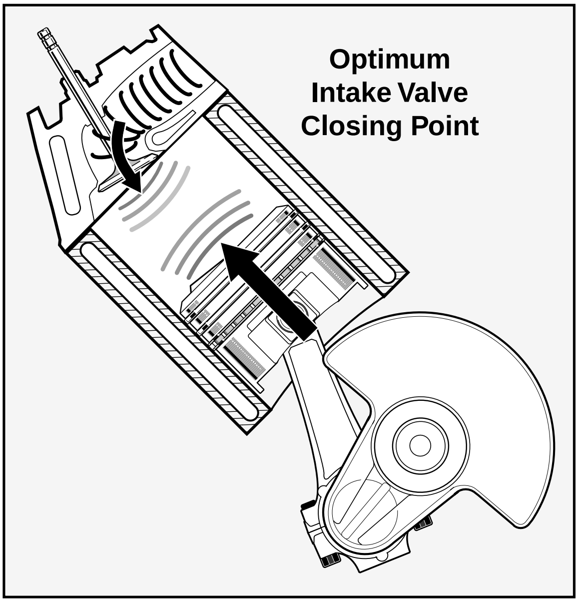
Header lengths need to be adjusted to harness EVO timing and scavenging effects to full benefit. However, header tubing <u>diameter</u> also plays an important part in determining exhaust system restriction and can change blowdown and pumping-loss characteristics. Despite the fact that EVO is an extremely important timing event, a combination of many tuning factors make it very difficult to determine the outcome of specific changes to EVO timing. Motion's Filling-And-Emptying simulation can "home-in" on the optimum combination from a large number of interdependent conditions.

bit of energy generated by combustion. Then, precisely at BDC, the exhaust valve would "pop" open and, miraculously, all of the residual gas pressure would vanish (even better, a slight vacuum would develop in the cylinder). Then as the piston moves from BDC to TDC, no horsepower would be wasted on "pumping" spent gasses from the engine. Unfortunately, the real world of engine gas dynamics is far from perfection. Substantial power is consumed by driving the piston up the bore on the exhaust stroke, especially at high engine speeds with large-displacement engines that generate prodigious amounts of exhaust gas. To offset these losses, early EVO timing starts the blowdown of high-pressure gasses before the exhaust stroke even begins. But the reduction in pumping work doesn't come without a drawback; earlier EVO timing "wastes" some of the power-producing gas pressure from combustion. The balance between these two factors optimizes power, but the balance changes as engine speed changes (and, of course, it also changes as displacement, flow restriction, and the timing of other valve events change).

To complicate matters even further, once an optimum EVO timing has been found, we need to consider how this timing affects the arrival of the scavenging wave. Remember, that the chain of events that leads to the arrival of the scavenging wave begins at the opening of the exhaust valve. When the pumping-loss/blowdown balance has been found, header lengths (and overlap timing) may need to be adjusted to harness scavenging effects to full benefit. Then when you consider that header tubing <u>diameter</u> plays a part in determining exhaust system restriction and can change blowdown and pumping-loss characteristics, the jumble of tunable elements grows even larger. Finally, if the induction system is improved, higher post-combustion pressures can worsen the pumping problem and that may require further changes to EVO timing, starting the whole tuning process over again.

Despite the fact that EVO is an extremely important factor in engine output, this web of interrelated effects make it very difficult (probably impossible) for engine experts to determine the outcome of specific changes to EVO timing. Here is another example of how engine simulations are useful. A series of simulations can "home-in" on the optimum combination from a large number of interdependent conditions.

IVC, The Intake Valve Closing Point—IVC is the most critical of the basic valve timing events. The intake valve closing point establishes a balance between cylinder filling and intake reversion, each having a potent effect on engine performance.



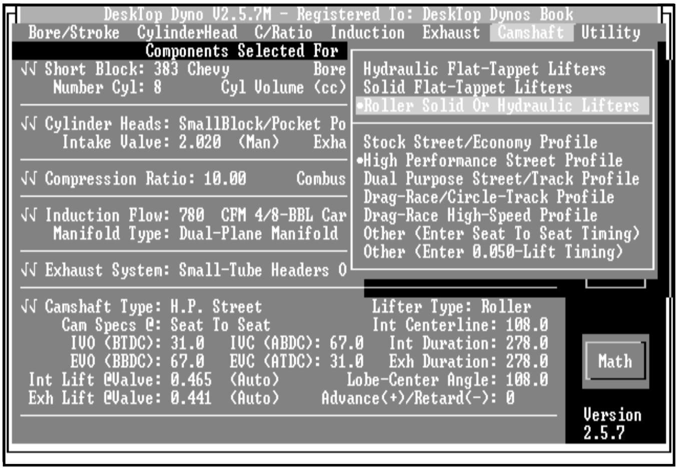
IVC is the most critical of the basic valve timing events. The intake valve closing point establishes a balance between cylinder filling and intake reversion, each having a potent effect on engine performance. When the pressure produced in the cylinder as the piston begins to move up the bore on the compression stroke exceeds the pressure of the incoming charge, the induced charge starts to "revert" or flow back into the induction system. This is the ideal point for IVC, because the cylinder has received the greatest volume of air and fuel and will generate the highest pressures on the power stroke.

After the intake stroke is completed and the piston reaches BDC, the column of air/fuel mixture moving through the induction system has built up considerable momentum (the "ram tuning" effect). This internal energy forces additional air and fuel to flow into the cylinder even as the piston begins to move up the bore on the early part of the compression stroke. At some point, however, the pressure in the cylinder begins to exceed the pressure of the incoming charge, and the induced charge starts to "revert" or flow back into the induction system. This is the ideal point for IVC, because the cylinder has received the greatest volume of air and fuel and will generate the highest pressures on the power stroke.

Unfortunately, optimum IVC occurs only at one engine speed and is dictated by cylinderhead flow, induction ram-tuning effects, intake valve opening timing, and of course by engine rpm. The longer the intake valve is held open the more "peaky" engine performance becomes. Late IVC can create induction flow reversion with an accompanying drop in manifold vacuum, rough idle and/or poor idle quality, and an increase in emissions from over-rich mixtures (rich mixtures are created when the charge is forced back up the manifold and passes through the carburetor—or by the fuel injector—a second time). When the induction system on a racing engine is properly designed, the pressure wave created when the intake valve opens is returned to the cylinder as a strong suction wave just about the time cylinder pressures begin to overcome the ram tuning effects. This momentary drop in pressure allows a bit more cylinder filling and a slightly later IVC. This critical tuning can add the winning edge to Pro Stock engines or other max-power applications, however, induction system design, IVO, and IVC timing must all be synchronized to produce these effects. Since induction components are typically hand built "one offs" on engines of this type, custom cams are ground for individual engines to optimize power in these competitive classes.

On the other end of the spectrum, if the goal is to build an engine that performs well throughout a wide rpm range and offers good idle characteristics, late IVC is definitely the wrong approach. The intake valve must close early enough to prevent reversion at lower engine speeds, an essential step in producing low-speed torque. However, early IVC limits cylinder filling at higher engine speeds and reduces peak power. IVC timing in the high 50-degree

The upper category of the camshaft menu provides a list of three common lifter types. The lower part of the menu offers an application-specific group of individual cam grinds, and the final two “Other” choices allow the direct entry of cam timing events to simulate virtually any camshaft.

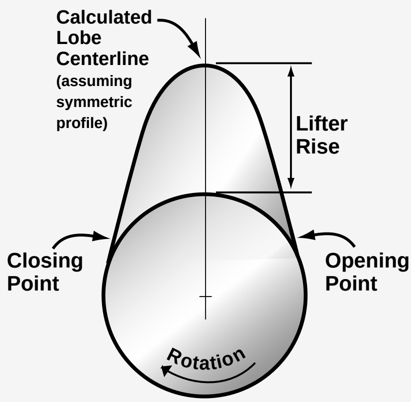


range is typical for a mild street cam, the high 60-degree to the mid 70-degree range is common on high-performance and mild racing grinds, and IVC from the 80-degree to low 100-degree range is all-out racing timing.

Camshaft Menu—Lifter Choices

The camshaft menu consists of two groups. The upper category provides a list of three common lifter types used to model camshaft acceleration rates. The lower part of the menu offers an application-specific group of individual cam grinds, and the final two “Other” choices allow the direct entry of cam timing events to simulate virtually any camshaft. Making a selection from each of these two groups “programs” the simulation to develop a valve motion curve for the selected camshaft.

The Filling-And-Emptying program uses a sophisticated model to accomplish this simulation, but the user must keep in mind that valve motion curves for both the intake and exhaust valves are being simulated from only six data points, three for the intake valve and three for the exhaust valve. The starting point for each simulation is the opening and closing timing and the lobe lift. From these three points, and the lifters selection, the program creates a motion curve that pinpoints valve lift at each degree of crank position. The results are remarkably accurate, however, the simulation cannot model subtle differences between cam grinds that use the same event timing and valve lift specs. Furthermore, the model develops a symmetric valve motion curve, although some cam grinds are asymmetric (meaning that the “opening” side of the lobe differs in shape from the “closing” side). Asymmetric modeling



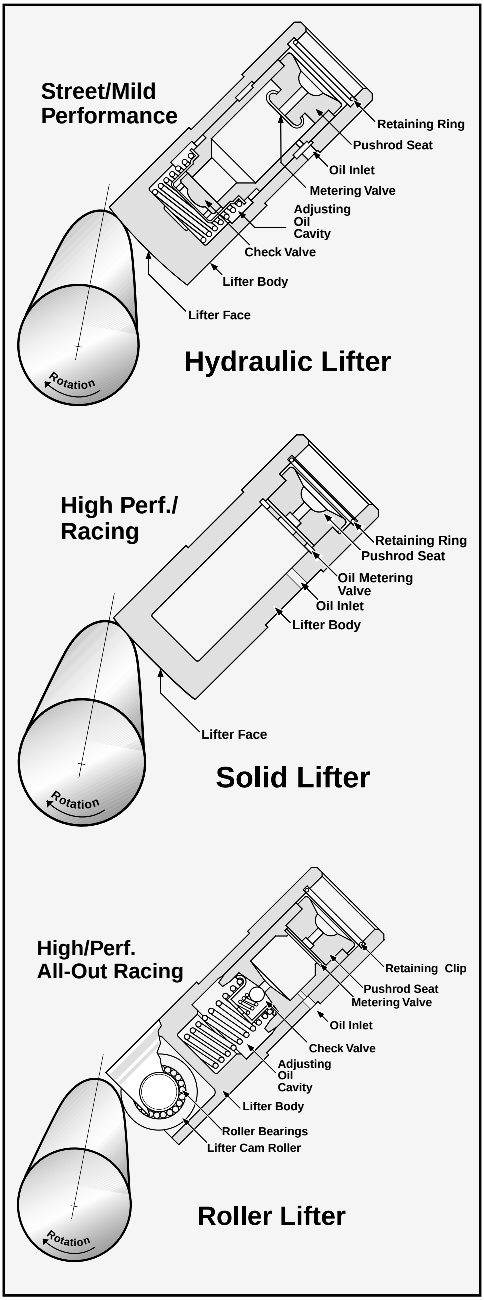
Lobe Profiles Are Calculated From Opening Point, Closing Point, And Lifter Rise

The Filling-And-Emptying program models symmetric valve motion curves from six data points, three for each lobe: 1) the opening point, 2) the closing point, and 3) the lobe lift. Although some cam grinds are asymmetric, performance differences between a symmetric model and actual asymmetric valve motion is quite small.

is impossible with only three data input points, luckily, performance differences between symmetric models and actual asymmetric valve motions are often quite small.

The first part of the Camshaft menu offers three choices: 1) *Hydraulic Flat-Tappet Lifters*, 2) *Solid Flat-Tappet Lifters*, and 3) *Roller Solid Or Hydraulic Lifters*. Each of these choices instructs the simulation to apply a unique "ramp-rate" model to the valve motion curve. The first two choices are flat-tappet lifters. This lifter uses a flat surface to contact or "rub" on the cam lobes. Flat-tappets are simple and quite reliable in stock and many high-performance applications. However, their design limits the rate at which the valves can be opened and closed. Slower valve acceleration reduces the exposed curtain area and the flow capability of the cylinderheads at every point during the lift curve, except for the fraction of a second when the valve passes through maximum lift. (Interestingly, flat tappet lifters can be made to out accelerate roller lifters during the first several hundredths of an inch of the lift curve, but roller lifters easily surpass flat tappet lift rates by the time the valves reach 0.100- to 0.200-inch of lift.) The third choice in the lifter group is a roller lifter (either solid or hydraulic, more on these differences next). This design incorporates a cylindrical element that rolls over the cam lobe. While there are slight gains from reduced friction, the greatest benefit from a roller cam/ lifter design lies "hidden" in the mechanical relationship between the roller and cam lobe. Roller cams can be ground with profiles that generate very high acceleration rates, opening and closing the valves much more quickly than flat-tappet cams. Faster valve acceleration increases the average curtain area exposed throughout the lift curve. This can substantially improve cylinderhead flow and horsepower.

The three choices in the lifter menu, as previously indicated, establish a "ramp-rate" model for the simulated valve-motion curve. The lowest acceleration is assigned to the first menu choice: *Hydraulic Flat-Tappet Lifters*. Hydraulic lifters incorporate a self-adjusting design that maintains zero lash



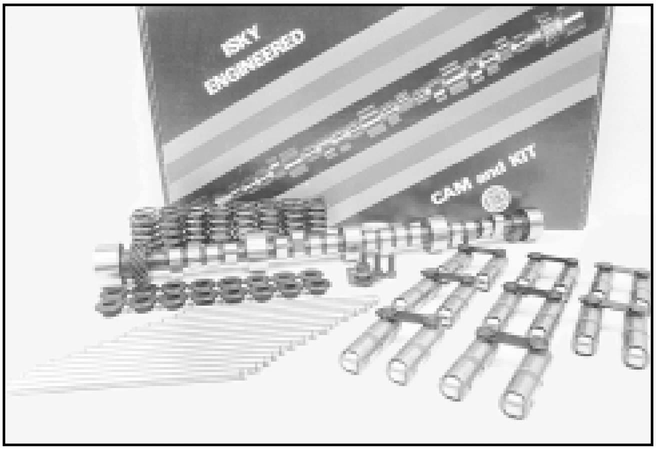
The three lifter choices establish a "ramp-rate" model for the simulated valve-motion curve. The lowest acceleration is assigned to *Hydraulic Flat-Tappet Lifters*. The next highest acceleration is applied to *Solid Flat-Tappet Lifters*. The highest acceleration is reserved for the last menu choice: *Roller Solid Or Hydraulic Lifters*.

in the valvetrain. They are well-known for providing quiet, trouble-free operation in mild- to high-performance street engines. Hydraulic, flat-tappet cam profiles usually generate low acceleration rates to optimize valvetrain reliability and extend engine life. These are the characteristics of the model used by the simulation program when *Hydraulic Flat-Tappet Lifters* is chosen. The next highest acceleration rates are assigned to *Solid Flat-Tappet Lifters*. These lifters incorporate no lash adjusting mechanism and require an operating clearance (or lash) in the valvetrain, usually 0.020- to 0.030-inch. Clearance is typically adjusted at the rockerarm or with spacers in the case of overhead cams with cam followers. Solid lifter cams are often ground with faster acceleration rate ramps, generate more valvetrain noise and wear, and are designed for performance-oriented applications. These more aggressive characteristics are used by the simulation to derive a valve-motion curve when *Solid Flat-Tappet Lifters* is chosen from the menu. Finally, as described above, the highest acceleration rates are applied to the last menu choice: *Roller Solid Or Hydraulic Lifters*. This choice applies to very aggressive ramp acceleration rates and derives valve motion curves appropriate for most racing, roller-lifter camshafts.

The simulation uses increasing valvetrain acceleration to model hydraulic, solid, and finally roller-lifter camshafts. This is a good assumption, since cams typically use lifters that are suited for the intended application, and cam profiles for specific applications typically apply predictable valve acceleration rates. However, this is not always the case. For example, some camshafts currently available for mild street engines use roller lifters, not to achieve high valve acceleration rates, but to optimize reliability. In these cases, choosing roller lifters from the Camshaft menu will produce optimistic power curves from the simulation. So, to improve program accuracy, ask yourself if the camshaft you are modeling fits the following application-specific description before you make a lifter selection:

Menu Choice	**Intended Application**
Hydraulic Flat-Tappet	Street/HP
Solid Flat-Tappet	HP/Racing
Roller	Very HP/Racing

If the cam you're modeling is a roller-lifter grind but a very mild-street profile, select Hydraulic or Solid Flat-Tappets from the menu since this choice will produce a lift



To improve program accuracy, ask yourself if the camshaft you are modeling fits the application-specific description listed in the above text. If your cam uses roller lifters but is a mild street profile, select *Hydraulic* or *Solid Flat-Tappets* from the menu since these choices will produce a lift curve that matches a mild camshaft. On the other hand, if the cam is a high-performance grind, select *Solid Lifters* or *Roller Lifters* since these will model the faster acceleration rates of an aggressive performance grind. If you are modeling a large-diameter, solid-lifter racing cam, like some "mushroom" lifter grinds, the Solid Lifter choice may underestimate the acceleration rate of these competition camshafts. In this case you may find more accurate predictions from the *Roller Lifter* selection.

curve that best matches a mild street camshaft. On the other hand, if the cam is a high-performance grind, select Solid Lifters since this will model the faster acceleration rates of aggressive performance grinds. If you are modeling a large-diameter, solid-lifter racing cam, like some "mushroom" lifter grinds, the Solid Lifter choice may underestimate the acceleration rate of these competition camshafts. In this case you may find more accurate predictions from the Roller Lifter selection.

Camshaft Menu—Application-Specific Camshafts

The second group within the Camshaft menu contains five camshaft "grinds" that are listed by application: 1) *Stock Street/ Economy Profile*, 2) *High Performance Street Profile*, 3) *Dual Purpose Street/Track Profile*, 4) *Drag-Race/Circle-Track Profile*, and 5) *Drag-Race High-Speed Profile*. Any of the three lifter types can be applied to these cam profiles, adjusting the acceleration rates from mild to very aggressive. When any of these cam profiles are selected, the seat-to-seat IVO, IVC, EVO, and EVC are loaded into the Component Selection Box along with the camshaft description (valve lift specs are calculated and displayed as described next). Simulation versions 2.5 and later also display the Intake Centerline, Intake Lobe Center Angle, Intake Duration, and Exhaust Duration for all simulated camshafts.

IMPORTANT NOTE: *The* ***intake and exhaust valve lifts*** *for all application-specific cams are automatically calculated by the simulation and displayed on screen followed by the term "(Auto)". Valve lifts are based on the valve-head diameters chosen by the user and displayed in the Cylinderhead category of the Component Selection Box. Note: If* ***valve diameters*** *are also being automatically calculated—by selecting "Auto Calculate Valve Size" from the CylinderHead menu—a cylinder-bore diameter and a cylinderhead selection must be made before the program can calculate the valve diameters and, consequently, the valve lifts. The simulation adjusts the intake and exhaust valve lifts to maintain appropriate lift-to-diameter ratios for a wide variety of applications, from single-cylinder small bore engines to large displacement racing engines with large valve diameters. The "auto calculation" feature for valve lift will be suspended and, instead, permanent values used for any camshaft by choosing one of the two "Other" selections at the bottom of the Camshaft menu AFTER choosing the desired camshaft (more on this in the next section).*

Stock Street/Economy Profile—This first profile is designed to simulate a typical factory-stock cam. All cam timing events displayed in the Component Selection Box are seat-to-seat measurements.

The EVO timing utilizes combustion pressure late into the power stroke and early IVC minimizes intake flow reversion. Late IVO and early EVC produce only 22 degrees of overlap, enough to harness some scavenging effects but restricted

This first menu profile simulates a typical factory-stock cam. It is generally used with hydraulic lifters.

```
  Camshaft Type: Stock Street/Economy     Lifter Type: Hydraulic
     Cam Specs @: Seat To Seat           Int Centerline: 115.0
      IVO (BTDC): 12.0    IVC (ABDC): 62.0  Int Duration: 254.0
      EVO (BBDC): 66.0    EVC (ATDC): 10.0  Exh Duration: 256.0
Int Lift @Valve: 0.XXX  (Auto)       Lobe-Center Angle: 116.5
Exh Lift @Valve: 0.XXX  (Auto)     Advance(+)/Retard(-): 0
```

This profile simulates a high-performance solid-lifter cam similar to *ISKY* #201025.

```
  Camshaft Type: High-Performance Street  Lifter Type: Hydraulic
     Cam Specs @: Seat To Seat             Int Centerline: 108.0
      IVO (BTDC): 31.0    IVC (ABDC): 67.0   Int Duration: 278.0
      EVO (BBDC): 67.0    EVC (ATDC): 31.0   Exh Duration: 278.0
Int Lift @Valve: 0.XXX  (Auto)        Lobe-Center Angle: 108.0
Exh Lift @Valve: 0.XXX  (Auto)     Advance(+)/Retard(-): 0
```

```
  Camshaft Type: Dual-Purpose Street       Lifter Type: Solid
    Cam Specs @: Seat To Seat          Int Centerline: 109.0
     IVO (BTDC): 32.0   IVC (ABDC): 70.0   Int Duration: 282.0
     EVO (BBDC): 70.0   EVC (ATDC): 32.0   Exh Duration: 282.0
Int Lift @Valve: 0.XXX  (Auto)        Lobe-Center Angle: 109.0
Exh Lift @Valve: 0.XXX  (Auto)     Advance(+)/Retard(-): 0
```

The third menu choice models a dual-purpose solid-lifter cam similar to *ISKY* #201281.

```
  Camshaft Type: Drag-Race Circle-Track    Lifter Type: Solid
    Cam Specs @: Seat To Seat          Int Centerline: 106.0
     IVO (BTDC): 42.0   IVC (ABDC): 74.0   Int Duration: 296.0
     EVO (BBDC): 77.0   EVC (ATDC): 45.0   Exh Duration: 302.0
Int Lift @Valve: 0.XXX  (Auto)        Lobe-Center Angle: 106.0
Exh Lift @Valve: 0.XXX  (Auto)     Advance(+)/Retard(-): 0
```

This profile simulates a competition camshaft similar to *ISKY* #201555.

```
  Camshaft Type: Drag-Race High-Speed      Lifter Type: Roller
    Cam Specs @: Seat To Seat          Int Centerline: 108.0
     IVO (BTDC): 52.0   IVC (ABDC): 88.0   Int Duration: 320.0
     EVO (BBDC): 88.0   EVC (ATDC): 52.0   Exh Duration: 320.0
Int Lift @Valve: 0.XXX  (Auto)        Lobe-Center Angle: 108.0
Exh Lift @Valve: 0.XXX  (Auto)     Advance(+)/Retard(-): 0
```

This profile models an all-out cam. The profile is similar to *ISKY Roller* #201600.

enough to prevent exhaust gas reversion into the induction system. The characteristics of this cam are smooth idle, good power from 1000 to 4500rpm, and good fuel economy. This cam works well in high-torque demand applications. The *Stock Street/Economy Profile* cam is typically used with hydraulic lifters. As described earlier, the intake and exhaust valve lifts for all application-specific profiles are automatically calculated by the simulation and are based on the valve diameters.

High Performance Street Profile—This profile is designed to simulate a high-performance factory camshaft. All cam timing events displayed in the Component Selection Box are seat-to-seat measurements.

This camshaft uses relatively-late EVO to fully utilize combustion pressure and early IVC minimizes intake flow reversion. IVO and EVC produce 62 degrees of overlap, a profile that is clearly intended to harness exhaust scavenging effects. The modestly-aggressive overlaps allow some exhaust gas reversion into the induction system at lower engine speeds, affecting idle quality and low-speed torque. The characteristics of this cam are fair idle, good power from 1500 to 6000rpm, and good fuel economy. This cam develops considerable power at higher engine speeds and is especially effective in lightweight vehicles. This *High Performance Street Profile* choice can be used with either hydraulic or solid lifters, and the simulation will accurately model this cam with either lifter selection (choose hydraulic lifters for more street-oriented applications and solid lifters for more high-performance oriented applications). This cam is nearly identical to the *ISKY Hi-Rev Flat-Tappet* cam part 201025 for the smallblock Chevy. As described earlier, the intake and exhaust valve lifts for all application-specific profiles are automatically calculated by the simulation and are based on the valve diameters.

Dual Purpose Street/Track Profile—This profile is designed to simulate a high-performance aftermarket camshaft. All cam timing events displayed in the Component Selection Box are seat-to-seat measurements.

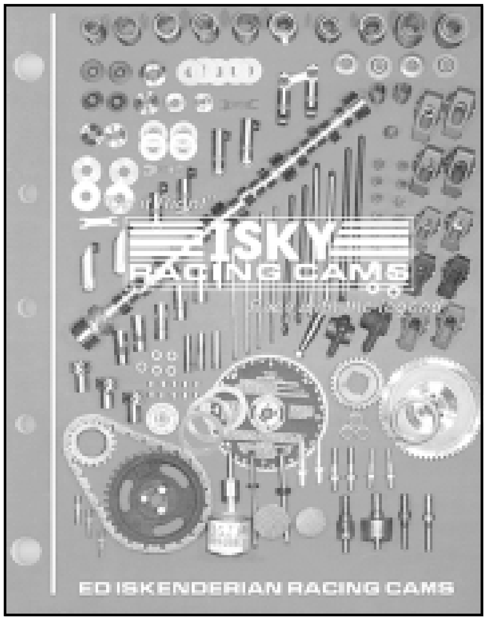
EVO timing on this camshaft is beginning to move away from specs that would be expected for simply utilizing combustion pressure with more of an emphasis toward early blowdown and minimizing exhaust pumping losses. The later IVC attempts to strike a balance between harnessing the ram effects of the induction system while minimizing intake flow reversion. IVO and EVC produce 64 degrees of overlap, a profile that is clearly designed to harness exhaust scavenging effects. The modestly aggressive overlap can allow some exhaust gas reversion into the induction system at lower engine speeds, affecting idle quality and low-speed torque. The characteristics of this cam are lopey idle, good power from 2500 to 6500rpm, and modest fuel economy. This cam develops considerable power at higher engine speeds and is especially effective in lightweight vehicles. This *Dual Purpose Street/Track Profile* choice can be used with either hydraulic or solid lifters, and the simulation will accurately model this cam design with either lifter selection (choose hydraulic lifters for more street-oriented applications and solid lifters for more competition-oriented applications). The profile of this cam is close to the *ISKY Hydraulic Series* cam part 201281 for the smallblock Chevy. As described earlier, the intake and exhaust valve lifts for all application-specific profiles are automatically calculated by the simulation and are based on the valve diameters.

Drag-Race/Circle-Track Profile—This profile is designed to simulate a competition aftermarket camshaft. All cam timing events displayed in the Component Selection Box are seat-to-seat measurements.

EVO timing on this racing camshaft places less emphasis on utilizing combustion pressure and more emphasis on beginning early blowdown to minimize exhaust pumping losses. The later IVC attempts to strike a balance between harnessing the ram effects of the induction system while minimizing intake flow reversion. IVO and EVC produce 90 degrees of overlap, a profile that is clearly intended to optimize exhaust scavenging effects. This aggressive overlap is designed for open headers and allows exhaust gas reversion into the induction system at lower engine speeds, affecting idle quality and torque below 3500rpm. The characteristics of this cam are very lopey idle, good power from 3600 to 7600rpm, with no consideration for fuel economy. This cam develops considerable power at higher engine speeds and is especially effective in lightweight vehicles. This *Drag-Race/Circle-Track Profile* choice can be used with either solid or roller lifters, and the simulation will accurately model this cam design with either lifter selection. The profile of this cam is similar to the *ISKY Oval Track Flat Tappet Series* cam part 201555 for the smallblock Chevy. As described earlier, the intake and exhaust valve lifts for all application-specific profiles are automatically calculated by the simulation and are based on the valve diameters.

Drag-Race High-Speed Profile—This profile is designed to simulate an all-out competition aftermarket camshaft. All cam timing events displayed in the Component Selection Box are seat-to-seat measurements.

All timing events on this camshaft are designed to optimize power on large dis-



Several of the "generic" grinds that are included in the menu selection of the Filling-And-Emptying simulation were selected from the ISKY CAMS catalog. ISKY's catalog is "simulation-friendly," listing seat-to-seat valve event timing for nearly every cam in their line.

placement engines at very high engine speeds with large-tube, open headers, and high compression ratios. This camshaft may not be effective in small displacement engines. EVO timing on this racing profile places the utilization of combustion pressure on the "back burner" and focuses emphasis on beginning early blowdown to minimize pumping losses during the exhaust stroke. This technique will help power at very high engine speeds, especially on large-displacement engines that do not easily discharge the high volume of exhaust gasses they produce. The late IVC attempts to harness the full ram effects of the induction system while relying on intake pressure wave tuning to minimize intake flow reversion. IVO and EVC produce 104 degrees of overlap, a profile that is clearly intended to utilize exhaust scavenging effects. This very aggressive overlap seriously affects idle quality and torque below 4000rpm. The characteristics of this cam are extremely lopey idle, good power from 4500 to 8500+rpm, with no consideration made for fuel consumption. This *Drag-Race High-Speed Profile* is designed to be used with roller lifters. The profile of this cam is similar to the *ISKY Roller Series* cam part 201600 for the smallblock Chevy. As described earlier, the intake and exhaust valve lifts for all application-specific profiles are automatically calculated by the simulation and are based on the valve diameters.

Each of the above application-specific cams can be modified in any way by directly entering valve-event or other cam-timing specs by choosing one of the two "Other" selections at the bottom of the Camshaft menu (more on this next).

Entering 0.050-Inch and Seat-To-Seat Timing

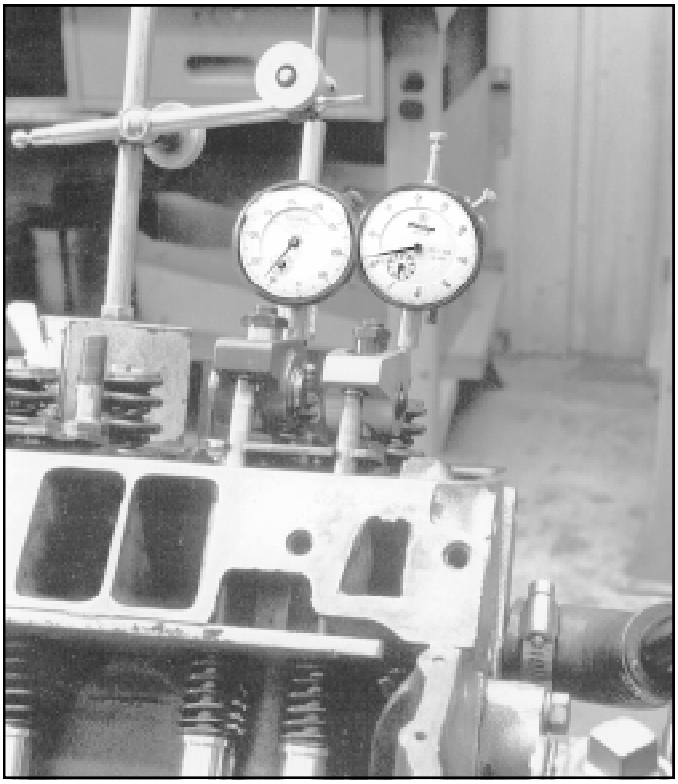
The last two selections in the Camshaft menu are "Other" choices that allow the direct entry of cam timing events to simulate virtually any camshaft. The first choice forces all specifications displayed and entered in the Camshaft category of the on-screen Component Selection Box to be treated as *Seat-To-Seat timing specs*. The second choice instructs the simulation to assume that all cam timing values are *0.050-inch timing specs*. Whenever an "Other" selection is made that requires the program to switch from one timing method to another, any currently displayed timing values are NOT changed; however, a warning message clearly indicates the timing method has changed. In addition, the new selected timing method is displayed next to "Cam Specs @:" in the Camshaft category. Furthermore, after an "Other" selection, the intake and exhaust valve-lift fields are switched to "(Man)," making entered or displayed valve-lift data permanent (turns off any auto-calculation techniques that may have been used to previously calculate valve lift).

Only the four basic timing events are affected by changes in seat-to-seat vs. 0.050-inch measurement methods. As you'll notice on the valve-motion curve drawing, changes in timing methods can only affect IVO, IVC, EVO, EVC, and the calculated intake and exhaust duration. The remaining timing events, including Intake Centerline (ICA), Exhaust Centerline (ECA), Lobe Center Angle (LCA), and Intake and Exhaust Valve Lift are <u>not altered</u> by either measurement method because none of these specs are derived relative to any of the basic four valve events.

The seat-to-seat timing method measures the valve timing—relative to piston position—when the valve or lifter has only just begun to rise or has *almost* completely returned to the base circle on the closing ramp. Unfortunately, there are no universal seat-to-seat measuring standards. These are some of the more common:

- **0.004-inch valve rise** for both intake and exhaust
- **0.006-inch valve rise** for both intake and exhaust
- **0.007-inch open/0.010-close valve rise** for both valves
- **0.010-inch valve rise** for both intake and exhaust
- **0.020-inch LIFTER rise** for both intake and exhaust

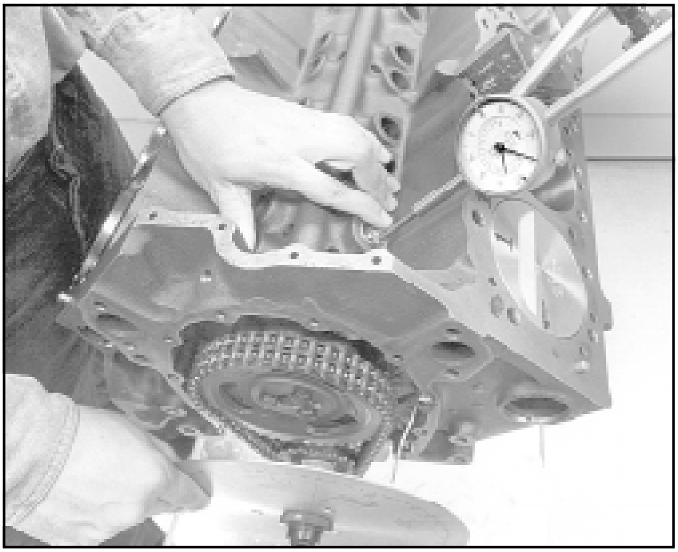
The timing specs measured using these methods are meant to approximate the actual valve opening and closing points that



Seat-to-seat timing measures the valve timing—relative to piston position—when the valve or (more rarely the lifter) has just begun to rise. Here dial indicators are positioned on the valvespring retainers and are measuring valve rise, which is the most common technique used with seat-to-seat timing (0.020-inch LIFTER rise is a notable exception). Timing specs measured using these methods are meant to approximate the actual valve opening and closing points that occur within the running engine. Because of this, seat-to-seat valve events are often called the *advertised* or *running* timing and will always produce the most accurate simulations.

occur within the running engine. Because of this, seat-to-seat valve events are often called the *advertised* or *running* timing. As we mentioned previously in this book, an engine simulation program needs just this type of information to calculate the beginning and end of mass flow in the ports and cylinders, a crucial step in the process of determining cylinder pressures and power output. *Because of this, seat-to-seat timing specifications produce the most accurate simulation results.* The 0.050-inch timing figures, while accepted by Motion's simulation, must be internally converted to seat-to-seat figures, unfortunately a less-than-perfect process, before they can be used in the simulation.

In the early days of the "cam wars," primarily during the 50's and 60's, the seat-to-seat timing method became popular with cam manufacturers as a way to "advertise" the duration of their popular grinds. Remember the bigger-is-better axiom? It probably reached its peak during this period. At that time, the way to be declared a winner in the marketplace was to offer the "biggest" and "baddest" camshaft, and that meant a cam with the longest duration. Manufacturers got so caught up in this foolishness that they used "trick" grinding methods to extend the clearance ramps and artificially increase seat-to-seat duration without appreciably affecting the valve-open duration (it was already too big). By this time, enthusiasts were confused by the myriad of seat-to-seat timing specs, and many were installing camshafts incorrectly. Even worse, the very nature of seat-to-seat timing makes it difficult to "nail down" the precise (rotational) position of the cam during engine assembly. To solve this problem, cam manufacturers united (picture water and oil!) to introduce a universal cam spec primarily aimed at making cam installation easier and more accurate. The new technique, called 0.050-inch timing, was based on the movement of the cam follower (lifter) rather than the valve. Since the lifter is moving quite quickly at 0.050-inch it was easy to accurately index the



The 0.050-inch lifter rise cam timing method has become one of the few standards in the performance marketplace. It measures the valve timing—relative to piston position—when the lifter has risen 0.050-inch off of the base circle of the cam. In the setup pictured here, the dial indicator is positioned on an intake lifter and is reading 0.050-inch; the 0.050-inch valve timing point can now be read directly off of the degree wheel attached to the crankshaft. Timing specs measured using this method <u>are not meant</u> to approximate the actual valve opening and closing points, instead their purpose is to permit accurate cam installation. All 0.050-inch timing specs entered into the Filling-And-Emptying program are <u>internally converted</u> to seat-to-seat timing. Because there is no way to precisely perform this conversion, always try to obtain and use seat-to-seat event timing to optimize simulation accuracy.

cam to the crank position. Today, regardless of who manufactures the cam, you will always find 0.050-inch lifter rise timing points published on the cam card, simplifying cam installation.

The 0.050-inch lifter rise timing method has become one of the few standards in the performance marketplace. In fact, some cam manufacturers have come to embrace this method so completely that they won't publish the old "advertised" or seat-to-seat timing events. This may go a long way toward establishing a standard, but it's a real step backwards from the standpoint of testing cams in engine simulation programs. As we have said, an engine simulation program MUST know when the valves lift off the seats and when they return to their seats in order to calculate mass flow into and out of the engine.

Let's examine how much VALVE lift typically occurs at 0.050-inch of LIFTER rise:

Valve Lift @ 0.050-inch Lifter Rise =
= (0.050 x Rocker Ratio) - Valve Lash

For example (for 1.5 rockers and 30 thousandths lash):

Valve Lift @ 0.050 =
= 0.050 x 1.5 - 0.030
= 0.045-inch

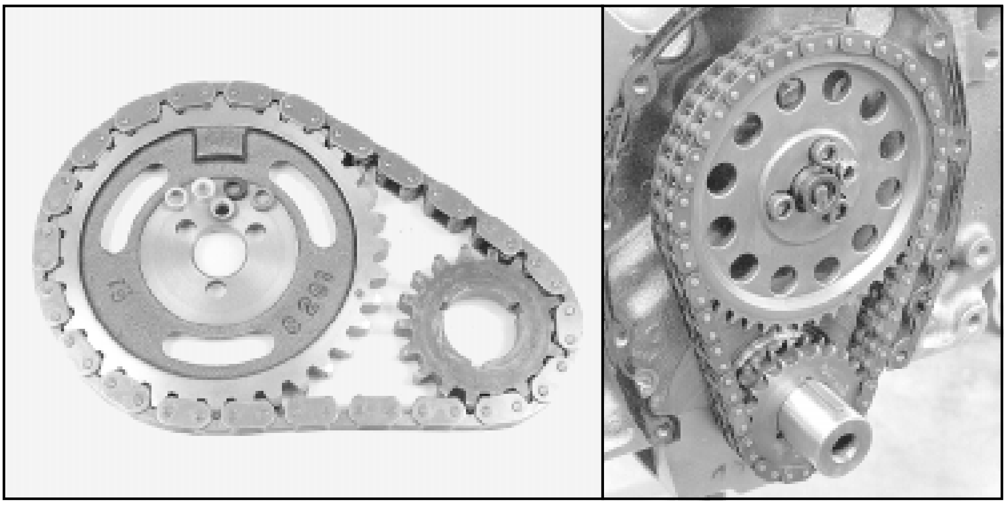
This example shows that the net valve lift is nearly equal to lifter rise. At this point, your knowledge of IC valve events should tell you that substantial flow occurs during this seemingly insignificant period. Consider the EVO point for example. By 0.050-inch, the exhaust valve is well on its way to depressurizing the cylinder and having dramatic effects on the power balance between induced torque and pumping losses on the exhaust stroke. Furthermore, the EVO point blasts a pressure wave through the exhaust system that returns as a scavenging wave during the overlap period. Similar critical functions occur at the other valve timing points. Valve motion during the first 0.050-inch of lift cannot be disregarded as insignificant when performing engine simulations. But what do you do when you just can't find seat-to-seat timing specs?

The first answer to that question is provided in the second "Other" choice of the Camshaft pull-down menu (additional tips for finding "missing" valve event specs is provided later in this guide). This selection declares that all specifications displayed and entered in the program are assumed to have been obtained using the *0.050-inch Lifter-Rise timing method*. (Whenever an "Other" selection is made that requires the program to switch from one timing method to another, a warning messages is displayed indicating a change has been made in cam timing methods.) All 0.050-inch timing specs entered into the program are <u>internally converted</u> to seat-to-seat timing points before the simulation is performed—remember, engine simulation is simply not possible without seat-to-seat valve timing specs. Unfortunately, there is no precise way to make this conversion, so an estimation is performed of where the seat-to-seat points might lie based on the known 0.050-inch timing points. Sometimes the program is able to guess very closely, and the power curves will match dyno results with that camshaft. At other times, the shape of the lobe is considerably different than the program's best guess, and accuracy will suffer.

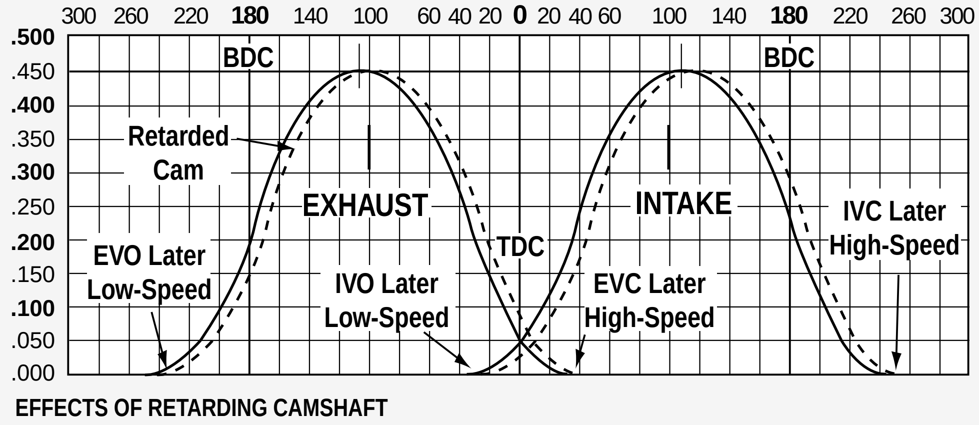
We realize that this situation can be frustrating for some users, particularly when you consider that a few tech support people at cam manufacturers become belligerent when asked for seat-to-seat valve events or tell customers that the information is "proprietary." Until these behind-the-times individuals "wake up" and realize that enthusiasts have tools like engine simulations and want to "test" cams before they buy, our best suggestion is to take your business elsewhere. But before you give up completely on your favorite cam grinder, take a look at the upcoming section on calculating valve events. It may show you how to calculate the needed timing from the jumble of specs printed in their cam catalog.

Camshaft Advance and Retard

In our earlier discussions on overlap, we found that this important cam timing event can be adjusted in two ways: 1) the wrong way by changing the Lobe Center Angle, or 2) the right way by changing the appropriate valve events to complement overlap timing. Changing LCA alone effectively moves two valve events to improve



Installing offset cam bushing in the cam gear is a common method of advancing or retarding cam timing. While this method can improve power, it hurts almost as much as it helps. Camshafts that show significant power gains from advanced or retarded timing have the wrong event timing for the engine.



ADVANCING/RETARDING A CAM IS ANOTHER "NONSENSE" TUNING APPROACH

The "schizophrenic" tuning approach we discussed earlier involving LCA also occurs when the cam is advanced or retarded. If advancing or retarding the cam is actually "poor practice," why is it so popular? The answer is simple: It is just about the only "tuning" change available to the engine builder without regrinding or replacing the cam. Advancing the cam slightly improves low-speed power, while retarding the cam gives a small boost in high-speed power. If advancing or retarding allows the engine to perform better, the cam profile was not optimum in the first place.

high-speed performance and the other two events decrease high-speed power. This "schizophrenic" tuning approach also occurs when the cam is advanced or retarded.

Selecting an "Other" choice from the Camshaft menu moves the cursor to the Component Selection Box and allows the direct entry of cam timing specifications. After you have entered all four valve events and both valve lift specs, the cursor moves to the "Advance(+)/Retard(-)" field. Changing this spec from zero (the default) to a positive value advances the cam (in crank degrees) while negative values retard the cam. The *Advance/Retard* function "shifts" all the intake and exhaust lobes the same advanced or retarded amount relative to the crankshaft. Why would you want to do this? The answer is simple: It is just about the only "tuning" change available to the engine builder without regrinding or replacing the cam. While it's possible to "tune" the cam using offset keys, special bushings, or multi-indexed sprockets, let's investigate what happens when all the valve events are advanced or retarded from the cam manufacturer's recommended timing.

It is generally accepted that advancing the cam improves low-speed power while retarding the cam improves high-speed power. When the cam is advanced, IVC and EVC occur earlier and that tends to improve low-speed performance; however, EVO and IVO also occur earlier, and these changes tend to improve power at higher engine speeds. The net result of these conflicting changes is a slight boost in low-speed power. The same goes for retarding the cam. Two events (later IVC and EVC) boost high-speed power and two (later EVO and IVO) boost low-speed performance. The net result is a slight boost in high-speed power.

Advancing or retarding a camshaft has the overall affect of reducing valve-timing efficiency in exchange for slight gains in low- or high-speed power. Consequently, most cam grinders recommend avoiding this tuning technique. If advancing or retarding allows the engine to perform better in a specific rpm range, the cam profile was probably not optimum in the first place. More power can be found at <u>both ends</u> of the rpm range by installing the right cam rather than advancing or retarding the wrong cam. However, if you already own a specific camshaft, slightly advanced or retarded timing may "fine tune" engine output to better suit your needs.

Calculating Valve Events And Using The Cam Math Calculator

Valve-motion curve drawings clearly indicate that cam timing events are related. No specific event is entirely isolated from the others. With sufficient information about the derivative cam timing specs, it is possible to calculate the basic four valve events needed to run a simulation. What is the minimum amount of information needed to figure out valve-event timing? There is no one answer to this question, but the answer that applies to the specs found in many cam manufacturers' catalogs is: *Intake Duration, Exhaust Duration, Lobe Center Angle (LCA), and the Intake Centerline (ICA)*. With these four cam specs, it is possible to calculate the IVO, IVC, EVO, and EVC. The calculated valve opening and closing events will be based on either seat-to-seat or 0.050-inch timing methods, depending on how the duration figures were measured. Remember, whenever you have a choice, always use seat-to-seat timing figures; the simulation results will have the highest accuracy. Also remember, the LCA and ICA do not change with timing methods, so the same LCA and ICA values can be used with either seat-to-seat or 0.050-inch duration figures.

Let's use the following example and work through the process of calculating the valve events hidden within this cam timing. A particular cam catalog lists a smallblock Ford cam as:

Cam Type:	**Hydraulic**
Net Valve Lift (Int):	**0.448-inch**
Net Valve Lift (Exh):	**0.472-inch**
Duration @ 0.050 (Int):	**204-degrees**
Duration @ 0.050 (Exh):	**214-degrees**
Duration @ 0.006 (Int):	**280-degrees**
Duration @ 0.006 (Exh):	**290-degrees**
Overlap @ 0.006:	**61-degrees**
Intake Centerline:	**107-degrees**
Lobe Center Angle:	**112-degrees**

This is a lot of information, but none of the basic valve events are listed. Refer to the valve-motion drawings. See if you can discover the relationships described by the following equations (they assume all cam lobes are symmetric; while asymmetric profiles are quite common, the amount of asymmetry typically is very small, so the following calculations should be accurate to within two or three degrees regardless of the cam profile).

The first step is to calculate how many degrees before TDC the intake valve opens (when IVO occurs). This can be done with the following formula (we'll use seat-to-seat duration figures):

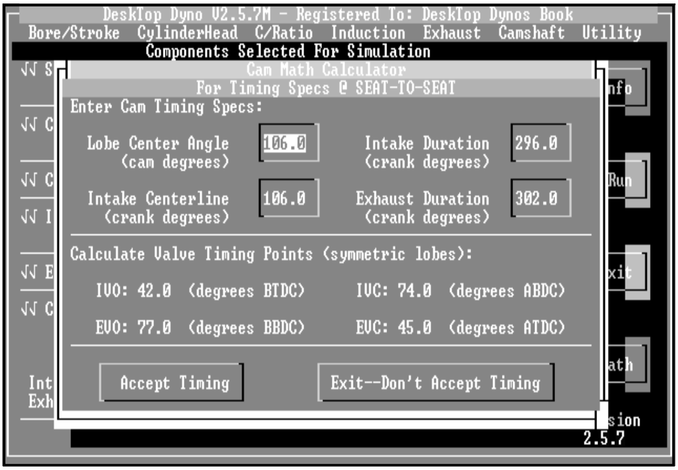
Intake Valve Opening (IVO) =
= (Intake Duration / 2) - ICA
= 280/2 - 107
= 140 - 107
= 33 degrees BTDC

Knowing the IVO, it is possible to calculate the intake valve closing point (IVC) by simply subtracting the IVO from the intake duration and then subtracting an additional 180-degrees to account for the full intake stroke:

Intake Valve Closing (IVC) =
= Intake Duration - IVO - 180
= 280 - 33 - 180
= 67 degrees ABDC

We're halfway there. The next step relates the known intake lobe timing to the exhaust lobe and then calculates the exhaust timing events. We must first locate the center of the exhaust lobe from the known intake centerline using the lobe center angle. The lobe center angle (LCA) is the distance (in cam degrees) between the exact center of both lobes. Since we know that the intake centerline (ICA) is the distance from TDC to the center of the intake lobe, it is possible to calculate where the center of the exhaust lobe lies relative to TDC:

Exhaust Centerline (ECA) =
= (2 x Lobe Center Angle) - ICA
= (2 x 112) - 107



The series of calculations performed on these pages can be done instantly by the *Cam Math Calculator* included in Motion's Filling And Emptying simulation version 2.5 (under development as this book went to press; send in your registration card to find out more about upgrades and new products). By clicking on the MATH button in the lower right of the screen, the *Cam Math Calculator* will "pop up" a window pre-loaded with the current cam timing. Any single event can be altered and the remaining events will be recalculated. This handy addition to the program makes short work of entering cam data from some manufacturer's catalogs.

= 224 - 107
= 117 degrees

Now that the exhaust centerline is known, we can repeat the same process to find the exhaust valve-event timing:

Exhaust Valve Closing (EVC) =
= (Exhaust Duration / 2) - ECL
= (290/2) - 117
= 145- 117
= 28 degrees ATDC

Finally, knowing the EVC, we can calculate the Exhaust Valve Opening (EVO) point by subtracting the EVC from the exhaust duration and then subtracting another 180 degrees to account for the full exhaust stroke:

Exhaust Valve Opening (EVO) =
= Exhaust Duration - EVC - 180
= 290 - 28 - 180
= 82 deg. BBDC

So the valve events for this cam are:

IVO = 33 degrees BTDC
IVC = 67 degrees ABDC
EVO = 82 degrees BBDC
EVC = 28 degrees ATDC

If any of these valve events had turned out to be negative, which is possible for some stock-type cams measured using 0.050-inch timing, enter the calculated timing figures into the simulation with the minus sign. For example, if EVC was determined to be -10 degrees, this would mean that the valve closes 10 degrees BEFORE TDC rather than after it. To tell the simulation that this is the case, enter a -10 for EVC.

This series of calculations is performed automatically by the *Cam Math Calculator* included in Motion's Filling And Emptying simulation version 2.5 (under development as this book went to press). By clicking on the MATH button in the lower right of the screen, the *Cam Math Calculator* will open a window and pre-load it with the cam timing currently displayed on screen. If any single event is changed, the remaining events are instantly recalculated and may be saved to the main screen or discarded. This handy addition to the program makes short work of not only entering cam data from manufacturer's catalogs, but also you can test the results of changes to Lobe Center Angle, Intake Centerline, Intake Duration and Exhaust Duration. Combined with the ability to change IVO, IVC, EVO, EVC, and overall advance and retard from the main screen, the new version 2.5 with the *Cam Math Calculator* allows changing virtually EVERY cam timing event and measuring its result.

Appendix-A Common Questions

COMMONLY ASKED QUESTIONS

The following information may be helpful in answering questions and solving problems that you encounter installing and using the DeskTop Dyno. If you don't find an answer to your problem here, send in the **Mail/Fax Tech Support Form** provided on page 97 and in the Installation And QuickStart guide provided with your software. We will review your problem and reply to you as soon as possible.

Question: Received an "Error Reading Drive A (or B)" message when attempting to run or install the DeskTop Dyno. What does this mean?

Answer: This means DOS cannot read the disk in your floppy drive. The disk may not be fully seated in your drive or the drive door (or lock arm) may not be fully latched. If you can properly read other disks in your drive, but the DeskTop Dyno distribution disk produces error messages, try requesting a directory of a known-good disk by entering **DIR A:** or **CHKDSK A:** and then perform those same operations with the DeskTop Dyno disk. If these operations produce an error message only when using the DeskTop Dyno disk, the disk is almost certainly defective. Return the disk to Motion Software, Inc., for a free replacement (address at bottom of Tech Support Form).

Question: Encountered a "DeskTop Dyno Has Not Been Properly Installed..." error message when trying to run the DeskTop Dyno. Why?

***Answer*:** This means that files required by the DeskTop Dyno were not found on your system. Reinstall the DeskTop Dyno using the SETUP program from the original distribution disk.

Question: The results of the simulation are listed in a simple chart on screen. Why are no power curves displayed?

***Answer*:** Any monitor and display card will work with the DeskTop Dyno, however, systems with EGA or high-resolution graphics capability will allow the display of horsepower and torque curve graphics. If you do not have an EGA or better graphics system (you are using a CGA or another low-resolution display), an on-screen "chart" listing for horsepower and torque will be substituted for power curves.

Question: What are the atmospheric and environmental conditions assumed by the program to predict horsepower?

Answer: Motion's Filling-And-Emptying software closely simulates the conditions that exist during an actual engine dyno test. The goal is to reliably predict the torque and horsepower that a dynamometer will measure throughout the rpm range while the engine and dyno are running through a programmed test. However, engine power can vary considerably from one dyno test to another if environmental and other critical conditions are not carefully controlled. In fact, many of the discrepancies between dyno tests are due to variabilities in what should have been "fixed" conditions. Among the many interviews conducted during the research and development of the software and for this book, dyno operators and engine owners readily acknowledged the possibilities of errors in horsepower measurements; however, very few these individuals believe that they had been lead astray by erroneous dyno readings. It seems that when test numbers "pop up" on a computer screen after a ground-pounding dyno test, the horsepower and torque values were accepted as "gospel." In other words, "dynos do lie, but they wouldn't do that to me"!

Unless the dyno operator and test personnel are extremely careful to monitor and control the surrounding conditions, including calibration of the instrumentation, dyno measurements are nearly worthless. Controlling these same variables in an engine simulation program is infinitely easier, but nevertheless just as essential. Initial conditions of temperature, pressure, energy, and methodology must be established and carefully followed. Here are some of the assumptions within the simulation software that establish a modeling baseline:

Fuel:

1) Gasoline rated at 19,000btu/lb as a standard fuel

2) The fuel is assumed to have sufficient octane to prevent detonation.

3) The air/fuel ratio is always maintained at 13:1 for optimum power.

Environment:

1) Air for induction is 68-degrees (F), dry (0% humidity), and of 29.86-in/Hg atmospheric pressure.

2) The engine, oil, and coolant have been warmed to operating temperature.

Methodology:

1) The engine is put through a series of "step" tests. That is the load is adjusted to "hold back" engine speed as the throttle is opened wide. Then the load is adjusted to allow the engine speed to rise to the first test point, 2000rpm in the case of the simulation. The engine is held at this speed for a few seconds and a power reading is taken. Then engine speed is allowed to increase to the next step, 2500rpm, and a second power reading is taken. This process is continued until the maximum testing speed is reached.

2) Since the testing procedure takes the engine up in 500rpm steps, and it is held steady during the measurement, the measured power does not reflect any losses from accelerating the rotating assembly (crank, rods, etc.) as would be found in a "real-world" application, such as drag racing.

Question: When I choose a carburetor that is too large for an engine (for example 1200cfm on a 283 Chevy), why does the power increase without the typically seen "bog" at low speeds?

***Answer*:** The DeskTop Dyno, along with virtually any current computer simulation program, cannot model over-carburetion and show the usual reduction in low-end performance that this causes. In reality, carburetors that are too large for an engine develop fuel atomization and air/fuel ratio instabilities. Unfortunately, this phenomenon is extremely complicated to model. The DeskTop Dyno assumes an optimum air/fuel ratio regardless of the selected CFM rating. While you can get positive results from larger-and-larger induction flows (by the way, this is not far from reality when optimum air/fuel ratios can be maintained, as is the case in electronic fuel-injection systems), you can't go wrong if you use common sense when selecting induction/carburetor flow capacities.

Question: I built a relatively stock engine but installed a drag-race camshaft. The engine only produced 9 hp @ 2000 rpm. Is this correct?

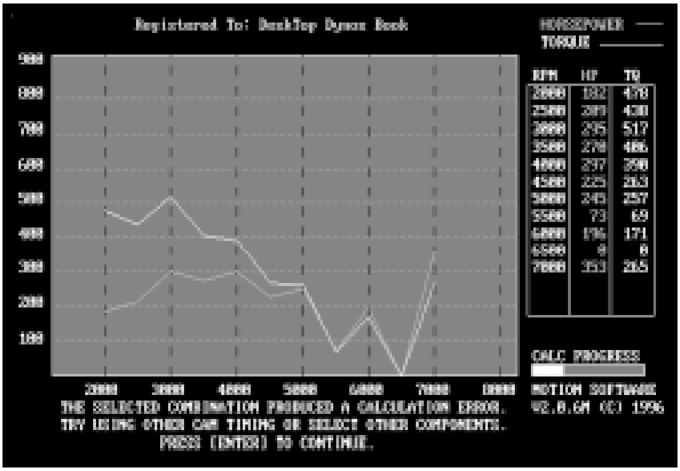
***Answer*:** Yes. Very low power outputs at low engine speeds occur when radical camshafts are used without complementary components, such as high-flow cylinderheads, high compression ratios, and exhaust system components that match the performance potential of the cam. In fact, some low performance engines with radical camshafts will show zero horsepower at low speeds. This means that if the engine was assembled and installed on a dynamometer, it would not produce enough power to offset any measurable load.

Question: The power predicted with the DeskTop Dyno seems too high for the Oldsmobile 455 engine that I am testing. Why?

***Answer*:** You selected cylinderheads for your Olds engine from the **Big Block** choices in the CylinderHead menu. While this may seem like the logical choice, some big-block engines are equipped with heads that flow significantly less air than other large (big-block) engines. You will obtain more accurate power predictions by choosing one of the **Smallblock** head selections when building one of these large-displacement engines. In addition to Oldsmobile, versions of the 429-460 Ford engines also came with restrictive cylinder-heads. When you make cylinderhead choices, keep these tips in mind: **Restrictive** applies to small-port heads (e.g., low-performance engines, such as early 260cid Ford engines or even flatheads) where the ports are small and restrictive for the installed valve sizes; consider **Smallblock** to describe heads with ports that are sized adequately or with performance in mind; **Bigblock** applies to heads where the ports are large or the valves are canted for improved flow (the first three big-block choices model oval-port heads, the last two choices simulate rectangular-port heads); and **4-Valve Heads** apply to many import and some racing configurations.

Question: The DeskTop Dyno produced an error message “THE SELECTED COMBINATION PRODUCED A CALCULATION ERROR...” What went wrong?

***Answer*:** The combination of components you have selected produced a calculation error in the simulation process (this is usually caused by a mathematical instability as the program repeatedly performed calculations to “home in” on a variable—called iteration). This is often caused by using restrictive induction flow on large-displacement engines, or by using radical cam timing on oth-



When severely unbalanced components are selected from the menus or entered manually, the iterative process of simulation can fall into mathematical instabilities, typically causing the curves to form a ragged "sawtooth" plot. Simply selecting a more balanced combination of parts and component specs will eliminate instabilities.

erwise mild engines. Try reducing the EVO timing specs, increasing the induction flow, selecting a cam with less duration, or reducing the compression ratio. A balanced group of components should not produce this error.

Question: The DeskTop Dyno takes 15 minutes to complete a simulation and draw horsepower and torque curves. Is there a problem with my computer or the software?

***Answer*:** Your computer does not have a math coprocessor (speeds up the simulation from 50 to 300 times). The DeskTop Dyno uses a powerful full-cycle simulation that performs millions of calculations for each point on the power curves, and this takes some time. But what you lose in speed with the DeskTop Dyno, you gain in accuracy over less sophisticated programs.

Question: How can I stop the simulation calculation if I realize that I've made a mistake selecting a component so that I don't have to wait for the program to draw the complete power and torque curves?

Answer: You can halt a simulation by pressing the **ESCape** key. Pressing **Enter** will resume the calculation; however, a second press of the **ESCape** key will abort the simulation run and return to the Main Program Screen.

Question: The DeskTop Dyno calculated the total Combustion Volume at 92ccs. But I know my cylinderheads have only 75ccs. What's wrong with the DeskTop Dyno?

***Answer*:** Nothing. The confusion comes from assuming that the calculated Total Combustion Volume is the same as your measured combustion-chamber volume. The Total Combustion Volume is the <u>entire volume</u> that remains when the piston reaches top dead center. This includes the combustion chamber, the remaining space above the piston top and below the deck surface, and the valve pockets; but it excludes any portion of the piston that protrudes into the combustion chamber. Compression ratio is calculated by this formula:

$$CR = \frac{\text{Cyl/Volume} + \text{Comb/Vol}}{\text{Comb/Volume}}$$

Both cylinder volume and combustion volume are calculated and displayed in the Component Selection Boxes.

Question: The horsepower produced when I enter the seat-to-seat timing on my cam card does not match the horsepower when I enter the 0.050-inch timing figures for the same camshaft. Why are there differences?

***Answer*:** The DeskTop Dyno uses the timing specs found on your cam card, and in cam manufacturer's catalogs, to develop a valve-motion curve (and from this, develops the instantaneous airflow for each port). Neither the seat-to-seat nor 0.050-inch timing figures precisely describe actual valve motion; you would need to measure valve position at each degree of crank rotation to come close to developing an exact valve-motion diagram! Lacking this, the DeskTop Dyno "creates" its own valve-motion diagram for use in later calculations of power and torque. A lot can happen in induction airflow between the time the valve is on the seat and when is reaches 0.050-inch of lifter rise. It is impossible (without exact measurement) to precisely "guess" the shape of the cam or the motion of the valve. This is the reason for the calculated differences. When in doubt, use seat-to-seat timing figures. They provide the DeskTop Dyno more information about valve motion, and are more likely to produce accurate simulated power levels.

Question: When I build an engine, the menus close after each component selection. Why don't the menus open, one-after-the-other, until the entire engine is assembled?

***Answer*:** This would make engine assembly *from scratch* easier. But it gets in the way when you want to change individual components to evaluate the changes in torque and horsepower. And since the true power of the DeskTop Dyno lies in its ability to make back-to-back tests—and it's these back-to-back tests that are essential steps in finding the best combination—we designed the menus to optimize changing individual components and quickly running back-to-back tests.

Question: How does the DeskTop Dyno allow for hydraulic, solid, and roller lifters?

***Answer*:** The DeskTop Dyno calculates a valve-motion diagram that is used in the subsequent calculations to predict horsepower and torque. When the choice is made to move from hydraulic to solid, and then from solid to roller lifters, the DeskTop Dyno increases the valve acceleration rates to coincide with the lobe shapes that are commonly found on these cam grinds. It is impossible (without exact measurement of valve position at each degree of crank motion) to "guess" the precise shape of the cam or the motion of the valve, but the DeskTop Dyno has proven to be remarkably accurate in simulating real-world valve motion and horsepower.

Question: Can I change rockerarm ratios with the DeskTop Dyno?

***Answer*:** Yes. Simply use this formula to alter valve lift (the DeskTop Dyno will calculate the new valve motion throughout the lift curve):

$$\textbf{New Lift = Old Lift x} \frac{\textbf{New Ratio}}{\textbf{Old Ratio}}$$

When you have calculated the new maximum valve lifts for the intake and exhaust valves, enter these numbers directly into the Component Selections Box by choosing one of the OTHER choices in the Camshaft menu.

Question: Why can't I just enter my port flow? Wouldn't that result in more accurate power predictions?

***Answer*:** If you could input the flow of both the intake and exhaust ports at valve lifts from zero to maximum lift for each 0.010-inch of valve motion, yes. <u>Published flow figures</u> for ports are virtually meaningless when it comes to predicting power potential. This is because the peak port flow only occurs when the valve reaches some maximum lift figure specified by the head manufacturer (and maximum lift in your engine may not be the same as the maximum lift that was used to rate port flow). Furthermore, the valve is only at maximum lift for a small fraction of its motion. To predict power you need to know the flow at each point of valve lift. This is calculated by the DeskTop Dyno from: 1) the valve head diameter, 2) knowing how the port shape (restrictive, smallblock, big-block, etc.) will affect flow across the valve, and 3) the valve motion diagram calculated from the valve-event timing.

Question: I found the published factory seat-to-seat valve timing for Pontiac engine that I am building, but I can't enter the valve events into the DeskTop Dyno. The IVC occurs at 110 degrees (ABDC), and I can only enter up to 100 in the program.

***Answer*:** There are so many ways that cam specs can be described for cataloging purposes that it's confusing to anyone trying to enter timing specs into an engine simulation program. Your Pontiac is a classic example of this lack of standards. The Pontiac cam listed in the factory manual is a hydraulic grind with seat-to-seat timing measured at 0.001-inch valve rise. Because the cam is designed for long life and quite operation, it has shallow opening ramps. This is the reason for the large number of crank degrees between the opening and closing points. In fact, during the first 35 degrees of crank rotation, the lifter rises less than 0.010-inch. If this wasn't the case and the valve opened and closed at the specified timing points listed in the factory manual, the cam would have over 350-degrees duration, and it's unlikely the engine would even start! The DeskTop Dyno can use 0.004- or 0.006-inch valve rise, 0.007-open/0.010-close valve rise, or even 0.020-inch lifter rise for seat-to-seat timing. But the 0.001-inch figures published in your factory manual are useless for engine simulation purposes.

Question: My cam manufacturer's catalog does not list seat-to-seat valve-event timing. But it does list seat-to-seat intake and exhaust duration, lobe-center angle, and intake centerline. Can I calculate the valve-event timing from these figures?

Answer: Yes. To calculate the intake and exhaust opening and closing points, you must have all of the following information:

1) **Intake Duration**
2) **Exhaust Duration**
3) **Lobe-Center Angle** (sometimes called lobe separation angle).
4) And the **Intake Centerline Angle**.

To perform the required calculations refer to the step-by-step procedure described on pages 86 and 87. Note: Version 2.5.7M of the DeskTop Dyno (under development when this guide was published) incorporates a *Cam Math Calculator* that performs this calculation instantly.

Question: I have been attempting to test camshafts from a listing in a catalog. I can find the duration and lobe center angle. The cam manufacturer won't give me the seat-to-seat timing (they act like it's a trade secret). Can I use the available data to test their cams?

Answer: No. As stated in the previous answer, you also need the intake-center angle to relate cam lobe positions to TDC and, therefore, crank position. Freely providing seat-to-seat timing or any of the other cam specs discussed in this booklet poses no threat to any cam grinder. It takes a lot more than valve-event timing to manufacture a quality cam; full profiles of the lobes are needed to ensure mechanically and dynamically stable operation. Cam companies that refuse to provide potential customers with simple valve-event information for evaluation in programs like the

The *Cam Math Calculator* is included in Motion's Filling And Emptying simulation version 2.5. By clicking on the MATH button the *Cam Math Calculator* will "pop up" a window pre-loaded with the current cam timing. Any single event can be altered and the remaining events will be recalculated.

DeskTop Dyno are simply living in the “dark ages.” Our suggestion is to contact another cam manufacturer (look into ISKY cams; see page 80).

Question: Everyone talks about how longer rods make more power. Why isn’t rod length one of the choices in the pull-down menus?

***Answer*:** Tests we have performed with the DeskTop Dyno show that rod length has virtually no affect on power. We realize that many actual dyno tests have shown power increases, but our simulation tests tell us that the power, when found, probably has little to do with piston dwell at TDC (and the associated thermodynamic effects) or changes in rod angularity on the crank pin. The measured power differences are most likely due to a reduction of friction on the cylinderwall from changes in side-loading on the piston. This can vary with bore finish, ring stability, piston shape, the frictional properties of the lubricant, etc. These variabilities are highly *unpredictable*. Some development, after all, can only be done in the real world on a engine dynamometer.

Question: Can I print out the horsepower and torque curves? I've tried using PrintScreen and all I get are the number, no curves.

Answer: You can print a dyno test sheet by selecting “P” from the **Simulation Completed Choices** prompt after any simulation run. The DeskTop Dyno printout includes a listing (in chart form) of the exact horsepower and torque figures and a complete summary of the components selected for that test. Because of the limitations in DOS (the DeskTop Dyno operates under DOS even while running in Windows) graphics printing of the power curves is not directly supported. However DOS and Windows users can still obtain a printout of the Simulator Screen including the horsepower and torque curves. Here are some tips and hints that will help you obtain printed graphic output.

Printing under DOS or Windows 3.x

You can print the Simulator Screen, including power curves, on almost any IBM-compatible, graphics-capable printer using the DOS utility called GRAPHICS.COM. If you are using DOS version 5.0 or later, you will find this nifty program in your DOS subdirectory. Type GRAPHICS at the DOS prompt before you start the DeskTop Dyno to enhance the Print-Screen function to include graphics printing. Then, with the curves displayed on screen that you would like to print, simultaneously press the **Shift** and **PrintScreen** keys to send the current graphics screen to your printer.

GRAPHICS has several options that “fine tune” the command for various printers and output styles. Here are a few of the more useful methods of using GRAPHICS (refer to your DOS manual for a complete list of options):

Type at the DOS prompt:

GRAPHICS LASERJET

or

GRAPHICS LASERJETII

to tell the GRAPHICS command that the output will be printed on a HP or compatible laser printer.

Type at the DOS prompt:

GRAPHICS LASERJET /R

to print the image as it appears on screen rather than reversed. Reversed (black to white) printing is the default.

Printing under Windows 95

Windows 95 incorporates DOS version 7.0 and is not supplied (nor is compatible) with the GRAPHICS command. Instead, when a simulation is completed, press the **ALT** and **ENTER** keys together to switch the Simulator Screen from full-screen to a window. Then select and copy the screen, including the power curve graph, to the clipboard (see the Windows help system for instructions on how to accomplish this). Open the Windows *Paint* program and paste the clipboard into *Paint*. Then, simply print the image to your Windows printer.

Question: I have an NEC *Ready* computer system. When I run a simulation, the curves are displayed properly, but the numbers and other words surrounding the power curves is garbled and completely unreadable. How can I fix this?

***Answer*:** The NEC *Ready* series of computers was released for sale in the U.S. during the 1995 Christmas sale season. This system has a “bug” in firmware (unchangeable, internal code stored in ROM chips) that causes it to improperly display EGA graphics characters. It maps the lower 128 ASCII to their upper 128 counterparts. Motion Software has developed a fix for this problem (a VGA version of the power curves screen) that is available at no charge. Return your DeskTop Dyno disk, along with a note mentioning your NEC computer problem, to: Motion Software, Inc., NEC replacement, 535 West Lambert, Bldg. E, Brea, CA, 92821.

Question: I have tried many different engine combinations using the same engine displacements and have noticed that several of the power curves begin at nearly the same horsepower and torque values at 2000rpm. Why are they so similar at this engine speed?

***Answer*:** Since the DeskTop Dyno uses a simulation technique that iterates toward an answer—performs a series of calculations that approach a more and more accurate result—the first power point must be developed based on educated “guesses” about mass flow and other variables. The next point, at 2500rpm, is calculated from the starting point, plus the data obtained from the completed simulation, so accuracy is higher. By 3000rpm, the power points are based on simulation calculations with virtually no remaining influence from the initial estimations.

Mail/Fax Tech Support Form For DeskTop Dyno

Please use this form (or a copy) to obtain technical support for the DeskTop Dyno from Motion Software, Inc. Fill out all applicable information about your system configuration, program version, and describe your problem as completely as possible. We will attempt to duplicate the problem and respond to your question. Mail or fax this form to the address below.

Your Phone (   ) ______-____________   Your Fax (   ) ______-_____________

Your Name ____

Address ____ **Apt. or Building** ___________

City ____________________________________ **State** ______ **ZipCode** _____________

Brand of computer ____________________________ **CPU** __________ **Speed** _______

Floppy ❑—5-1/4 ❑—3-1/2 Size of hard drive ______________ **Amt of RAM** ________

Describe your monitor and graphics card ______________________________________

____

Version of DOS (enter the command VER at the DOS prompt) _____________________

Version of DeskTop Dyno (refer to distribution disk label) _________________________

List the TSR (Terminate and Stay Resident) programs loaded by your Autoexec.bat or Config.sys files that are present when you are running the DeskTop Dyno.

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Please describe the problem, the point in the program at which the problem first occurred and, if necessary, the menu choices that caused the problem to occur.

____

____

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____

____

Can you duplicate the problem? ____

Mail this form to: Motion Software, Inc., 535 West Lambert, Bldg. E, Brea, CA 92821

Or fax to: 714-255-7956 (24 hours)

Appendix-B Glossary

0.050-Inch Cam Timing Method—See **Cam Timing, @ 0.050-inch**.

ABDC or **After Bottom Dead Center**—Any position of the piston in the cylinder bore after its lowest point in the stroke (BDC). ABDC is measured in degrees of crankshaft rotation after BDC. For example, the point at which the intake valve closes (IVC) may be indicated as 60-degrees ABDC. In other words, the intake valve would close 60 degrees after the beginning of the compression stroke (the compression stroke begins at BDC).

Air-Fuel Ratio—The proportion of air to fuel—by weight—that is produced by the carburetor or injector.

ATDC or **After Top Dead Center**—Any position of the piston in the cylinder bore after its highest point in the stroke (TDC). ATDC is measured in degrees of crankshaft rotation after TDC. For example, the point at which the exhaust valve closes (EVC) may be indicated as 30-degrees ATDC. In other words, the exhaust valve would close 30 degrees after the beginning of the intake stroke (the intake stroke begins at TDC).

Atmospheric Pressure—The pressure created by the weight of the gases in the atmosphere. Measured at sea level this pressure is about 14.69psi.

Back Pressure—A pressure developed when a moving liquid or gaseous mass passes through a restriction. "Backpressure" often refers to the pressure generated within the exhaust system from internal restrictions from tubing and tubing bends, mufflers, catalytic converters, tailpipes, or even turbochargers.

BBDC or **Before Bottom Dead Center**—Any position of the piston in the cylinder bore before its lowest point in the stroke (BDC). BBDC is measured in degrees of crankshaft rotation before BDC. For example, the point at which the exhaust valve opens (EVO) may be indicated as 60-degrees BBDC. In other words, the exhaust valve would open 60 degrees before the exhaust stroke begins (the exhaust stroke begins at BDC).

Big-Block—A generic term that usually refers to a V8 engine with a displacement that is large enough to require a physically "bigger" engine block. Typical big-block engines displace over 400 cubic inches.

Blowdown or **Cylinder Blowdown**—Blowdown occurs during the period between exhaust valve opening and BDC. It is the period (measured in crank degrees) during which residual exhaust gases are expelled from the engine before the exhaust stroke begins. Residual gasses not discharged during blowdown must be physically "pumped" out of the cylinder during the exhaust stroke, lowering power output from consumed "pumping work."

Bore or **Cylinder Bore**—The internal surface of a cylindrical volume used to retain and seal a moving piston and ring assembly. "Bore" is commonly used to refer to the cylinder bore diameter, unusually measured in inches or millimeters. Bore surfaces are machined or ground precisely to afford an optimum ring seal and minimum friction with the moving piston and rings.

Brake Horsepower (bhp)—Brake horsepower (sometimes referred to as *shaft* horsepower) is always measured at the flywheel or crankshaft by a "brake" or absorbing unit. *Gross brake horsepower* describes the power output of an engine in stripped-down, "race-ready" trim. *Net brake horsepower* measures the power at the flywheel when the engine is tested with all standard accessories attached and functioning. Also see Horsepower, Indicated Horsepower, Friction Horsepower, and Torque.

Brake Mean Effective Pressure (bmep)—A theoretical average pressure that would have to be present in each cylinder during the power stroke to reproduce the force on the crankshaft measured by the absorber (brake) on a dynamometer. The bmep present during the power stroke would produce the same power generated by the varying pressures in the cylinder throughout the entire four-cycle process.

BTDC or **Before Top Dead Center**—Any position of the piston in the cylinder bore before its highest point in the stroke (TDC). BTDC is measured in degrees of crankshaft rotation before TDC. For example, the point at which the intake valve opens (IVO) may be indicated as 30-degrees BTDC. In other words, the intake valve would open 30 degrees before the intake stroke begins (the intake stroke begins at TDC).

Cam Timing, @ 0.050-Lift—This method of determining camshaft valve timing is based on 0.050 inches of <u>tappet rise</u> to pinpoint timing events. The 0.050 inch method was developed to help engine builders accurately install camshafts. Lifter rise is quite rapid at 0.050-inch lift, allowing the cam to be precisely indexed to the crankshaft. Camshaft timing events are always measured in crankshaft degrees, relative to TDC or BDC.

Cam Timing, @ Seat-To-Seat—This method of determining camshaft timing uses a specific <u>valve lift</u> (determined by the cam manufacturer) to define the beginning or ending of valve events. There is no universally accepted valve lift used to define seat-to-seat cam timing, however, the Society of Automotive Engineers (S.A.E) has accepted 0.006-inch valve lift as its standard definition. Camshaft timing events are always measured in crankshaft degrees, relative to TDC.

Camshaft Advance/Retard— This refers to the amount of advance or retard that the cam is installed from the manufacturers recommended setting. Focusing on intake timing, advancing the cam closes the intake valve earlier. This setting typically increases low-end performance. The retarded cam closes the intake valve later which tends to help top end performance.

Camshaft Follower or **Lifter**—Usually a metal cylinder (closed at one end) that rubs against the cam lobe and converts the rotary motion of the cam to an up/down motion required to open and close valves, operate fuel pumps, etc. Cam followers (lifters) can incorporate rollers, a design that can improve reliability and performance in many applications. Roller lifters are used extensively in racing where valve lift and valve-lift rates are very high, since they can withstand higher dynamic loads. In overhead cam engines, the cam follower is usually incorporated into the rocker arm that directly actuates the valve; in this design push rods are eliminated.

Camshaft Grind—The shape of the cam lobe. Determines when the intake and exhaust valves open and close and how high they lift off of the seats. The shape also determines *how fast* the valves open and close, i.e., how much acceleration the valves and springs experience. High acceleration rate cams require large-diameter solid, mushroom, or roller lifters.

Camshaft Lift—The maximum height of the cam lobe above the base-circle diameter. A higher lobe opens the valves further, often improving engine performance. Lobe lift must be multiplied by the rocker ratio (for engines using rocker arms) to obtain total valve lift. Lifting the valve more than 1/3 the head diameter generally yields little additional performance. Faster valve opening rates add stress and increase valvetrain wear but can further improve performance. High lift rates usually require specially designed, high-strength components.

Camshaft Lobe—The eccentrically shaped portion of a camshaft on which the camshaft follower or lifter rides. The shape of intake and exhaust cam lobes are important engine design criterion. They directly affect engine efficiency, power output, the rate (how fast) the valves open and close, and control valvetrain life and maximum valvetrain/engine rpm.

Camshaft Timing—The rotational position of the camshaft, relative to the crankshaft, i.e., the point at which the cam lobes open and close the valves relative to piston position. Two common methods are used to indicate the location of valve events: the Seat-To-Seat and 0.050-inch timing methods. For simulation purposes, Seat-To-Seat timing values yield more accurate horsepower and torque predictions. Camshaft timing can be adjusted by using offset keys or offset bushings (or by redesigning the cam profile). Valve-to-piston clearance will vary as cam timing is altered; always ensure that adequate clearance exists after varying cam timing from manufacturer's specifications. See *Cam Timing @ Seat-To-Seat* and *Cam Timing @*

Appendix-B Glossary

0.050-Inch.

Carburetor—A device that combines fuel with air entering the engine; capable of precision control over the air volume and the ratio of the fuel-to-air mixture.

Centerline—An imaginary line running through the center of a part along its axis, e.g., the centerline of a crankshaft running from front-to-back directly through the center of the main-bearing journals.

Closed Headers or Closed Exhaust System—Refers to an exhaust system that includes mufflers; not open to the atmosphere.

Combustion Chamber or **Combustion Chamber Volume**—The volume contained within the cavity or space enclosed by the cylinderhead, including the "top" surfaces of the intake and exhaust valves and the spark plug. Not the same volume as the *combustion space volume*.

Combustion Space or **Combustion Space Volume**—The volume contained within the cylinderhead, plus (or minus) the piston dome (or dish) volume, plus any volume displaced by the compressed head gasket, plus (or minus) any additional volume created by the piston not fully rising to the top of the bore (or extending beyond the top of the bore) of the cylinder at TDC. This volume is used to calculate compression ratio.

Compression Pressure—The pressure created in the cylinder when the piston moves toward top dead center (TDC) after the intake valve closes, trapping the induced charge (normally a fuel/air mixture) within the cylinder. Compression pressure can be measured by installing a pressure gauge in the cylinder in place of the spark plug and "cranking" the engine with the starter motor. To improve measurement accuracy, the throttle is usually held wide open and the remaining spark plugs are removed to minimize cranking loads and optimize pressures in the cylinder under test.

Compression Ratio—The ratio of the total volume enclosed in a cylinder when the piston is located at BDC compared to the volume enclosed when the piston is at TDC (volume at TDC is called the *combustion space volume*). The formula to calculate compression ratio is: (Swept Cylinder Volume + Combustion Space Volume)/Combustion Space Volume = Compression Ratio.

Compression Stroke—One of the four 180-degree full "sweeps" of the piston moving in the cylinder of a four-stroke, internal-combustion engine (originally devised by Nikolaus Otto in 1876). During the compression stroke, the piston moves from BDC to TDC and compresses the air/fuel mixture. Note: The 180-degree duration of the compression stroke is commonly longer than the duration between the intake valve-closing point and top dead center (or ignition), sometimes referred to as the true "Compression Cycle." The compression stroke is followed by the power stroke.

Cubic Inch Displacement or **CID**—The swept volume of all the pistons in the cylinders in an engine expressed in cubic inches. The cylinder displacement is calculated with this formula: (Bore x Bore x Pi x Stroke x No.Cyl.)/4. When the bore and stroke are measured in inches, the engine displacement calculated in cubic inches.

Cylinder and **Cylinder Bore**—The cylinder serves three important functions in an internal-combustion (IC) engine: 1) retains the piston and rings, and for this job must be precisely round and have a uniform diameter (for performance applications 0.0005-inch tolerance is considered the maximum allowable); 2) must have a surface finish that ensures both optimum ring seal (smooth and true) and yet provides adequate lubrication retention to ensure long life for both the piston and rings; and 3) the cylinder bore acts as a major structural element of the cylinder block, retaining the cylinderheads and the bottom end components. The cylinder bore design, finish, and its preparation techniques are extremely important aspects of performance engine design.

Cylinder Block—The casting that comprises the main structure of an IC engine. The cylinder block is the connecting unit for the cylinderheads, crankshaft, and external assemblies, plus it houses the pistons, camshaft and all other internal engine components. The stability, strength, and precision of the block casting and machining are extremely important in obtaining optimum power and engine life. Cylinder blocks are usually made from a high grade of cast iron.

Cylinderhead—A component (usually made of cast iron or cast aluminum) that forms the combustion chambers, intake and exhaust ports—including water cooling passages—and provides support for valvetrain components, spark plugs, intake and exhaust manifolds, etc. The cylinderhead attaches to the engine block with several large bolts that squeeze a head gasket between the block deck and head surfaces; and when attached, the head becomes a load-carrying member, adding strength and rigidity to the cylinder block assembly. Modern cylinderhead designs fall into three major categories: 1) overhead-valve with wedge, canted-valve, or hemispherical combustion chambers; 2) single-overhead cam with wedge or hemispherical cambers; 3) double-overhead cam with hemispherical chambers.

Degree—1) An angular measurement. A complete circle is divided into 360 degrees; equal to one crankshaft rotation; 180 degrees is one-half rotation. 2) A temperature measurement. The temperatures of boiling and freezing water are: in the Fahrenheit system 212 and 32 degrees; in the Celsius system 100 and 0 (zero) degrees.

Density—A measurement of the amount of matter within a known space or volume. Air density is the measurement of the amount of air per unit volume at a fixed temperature, barometric pressure, altitude, etc.

Detonation—The secondary ignition of the air/fuel mixture in the combustion space causing extreme pressures. Detonation is caused by low gasoline octane ratings, high combustion temperatures, improper combustion chamber shape, too-lean mixtures, etc. Detonation produces dangerously high loads on the engine, and if allowed to continue, will lead to engine failure. Detonation, unlike preignition, requires two simultaneous combustion fronts (fuel burning in two or more places in the combustion chamber at once); whereas preignition occurs when the fuel-air mix ignites (with single burning front) before the spark plug fires. Both preignition and detonation produce an audible "knock" or "ping," but detonation does not produce the rapid "wild pinging" noise that is typically associated with preignition. The extreme pressures of detonation can lead to preignition, but even worse the high temperatures of preignition can cause detonation.

Duration or **Valve Duration**—The number of crankshaft degrees (or much more rarely, camshaft degrees) of rotation that the valve lifter or cam follower is lifted above a specified height; either <u>seat-to-seat valve duration</u> measured at 0.006-, 0.010-inch or other valve rises (even 0.020-inch lifter rise), or duration measured at 0.050-inch lifter rise called <u>0.050-inch duration</u>. Intake duration is a measure of all the intake lobes and exhaust duration indicates the exhaust timing for all exhaust lobes. Longer cam durations hold the valves open longer, often allowing increased cylinder filling or scavenging at higher engine speeds.

Dynamometer—A device used to measure the power output of rotating machinery. In it's simplest terms, a dynamometer is a power-absorbing brake, incorporating an accurate method of measuring how much torque (and horsepower) is being absorbed. Braking is accomplished through friction (usually a hydraulic absorber) or by an electric dynamo (converts energy to electricity). Modern computer-controlled dynamometers for high-performance automotive use have sophisticated speed controls that allow the operator to select the rpm point or range of speeds through which the torque is to be measured. Then the operator opens the throttle and the dynamometer applies the precise amount of load to maintain the chosen rpm points; horsepower is read out directly on a gauge and/or computer screen.

Dynomation—A engine simulation program developed by V.P. Engineering, Inc. ($600; 515-986-9197) that uses full wave-action analysis, currently using the Method Of Characteristics to provide solutions to the complex equations of wave dynamics.

Empirical or **Empirical Testing**—Meaning "after the fact" or "experimental," empirical testing involves actual "real-world" experiments to determine the outcome of component changes.

Exhaust Center Angle/Centerline or **ECA—** The distance in crank degrees from the point of maximum exhaust valve lift (on symmetric cam profiles) to TDC during the valve overlap period.

Exhaust Manifold—An assembly (usually an iron casting) that connects the exhaust ports to the remainder of the exhaust system. The exhaust manifold may include a heat-riser valve or port that heats the intake manifold to improve fuel vaporization.

Exhaust Ports—Cavities within the cylinderhead that form the initial flow paths for the spent gases of combustion. One end of the exhaust port forms the exhaust valve seat and the other end forms a connecting flange to the exhaust manifold or header.

Exhaust Stroke—One of the four 180-degree full "sweeps" of the piston moving in the cylinder of a four-stroke, internal-combustion engine (originally devised by Nikolaus Otto in 1876). During the exhaust stroke, the piston moves from BDC to TDC and forces exhaust gases from the cylinder into the exhaust system. Note: The 180-degree duration of the exhaust stroke is commonly shorter than the period during which the exhaust valve is open, sometimes referred to as the true "Exhaust Cycle." The exhaust stroke is followed by the intake stroke.

Exhaust Valve Closing or **EVC**—The point at which the exhaust valve returns to its seat, or closes. This valve timing point usually occurs early in the intake stroke. Although EVC does not have substantial effects on engine performance, it contributes to valve overlap (the termination point of overlap) that can have a significant effect on engine output.

Exhaust Valve Duration—See Duration

Exhaust Valve Lift—See Valve Lift

Exhaust Valve Opening or **EVO**—The point at which the exhaust valve lifts off of its seat, or opens. This valve timing point usually occurs late in the power stroke. EVO usually precedes BDC on the power stroke to assist exhaust-gas *blowdown*. This EVO timing point can be considered the second most important cam timing event.

Exhaust Valve—The valve located within the cylinderhead that control the flow of spent gases from the cylinder. The exhaust valves are precisely actuated (opened and closed) by the camshaft, usually through lifters, pushrods, and rockerarms. Exhaust valves must withstand extremely high temperatures (1500 degrees-F or higher) and are made from special steels, e.g., SAE J775 that has excellent strength at high temperatures and good resistance to corrosion and wear.

Filling & Emptying Multidimensional Simulation—This engine simulation technique includes multiple models (e.g., thermodynamic, kinetic, etc.), and by dividing the intake and exhaust passages into a finite series of sections it describes mass flow into and out of each section at each degree of crank rotation. The Filling And Emptying method can accurately predict average pressures within sections of the intake and exhaust system and dynamically determine VE and engine power. However, the basic Filling And Emptying model can not account for variations in pressure *within* individual sections due to gas dynamic effects. See Gas-Dynamic Multidimensional Simulation.

Finite-Amplitude Waves—Pressure waves of higher energy levels higher than acoustic waves. Finite-amplitude waves exhibit complex motions and interactions when traveling through engine passages. These actions make their mathematical analysis very complex.

Flat-Tappet Lifter—A camshaft follower having a flat surface at the point of contact with the cam lobe. Flat-tappet lifters actually have a shallow convex curvature at their "face" to allow the lifter to rotate during operation, extending the working life.

Flow Bench and **Flow-Bench Testing**—A flow bench is a testing fixture that develops a precise pressure differential to either "suck" our "blow" air through a cylinderhead or other engine component. A flow bench determines the flow capacities (restrictions) of cylinderhead ports and valves and assists in the analysis of alterations to port contours.

Four-Cycle Engine—Originally devised by Nikolaus Otto in 1876, the four-cycle engine consists of a piston moving in a closed cylinder with two valves (one for inlet and one for outlet) timed to produce four separate strokes, or functional cycles: Intake, Compression, Power, and Exhaust. Sometimes called the "suck, squeeze, bang, and blow" process, this technique—combined with a properly atomized air/fuel mixture and a precisely timed spark ignition—produced an engine with high efficiency and power potential. The software discussed in this book is designed to simulate the functional processes of a four-cycle engine.

Friction Horsepower (Fhp)—The power absorbed by the mechanical components of the engine during normal operation. Most frictional losses are due to piston ring pressure against the cylinder walls. Frictional power losses are not easily measured, however, they can be accurately *calculated* knowing the brake horsepower (from dyno testing) and the indicated horsepower (from pressure measurements). Also see Indicated Horsepower and Brake Horsepower.

Friction Mean Effective Pressure (Fmep)—A theoretical average pressure that would have to be present in each cylinder during the power stroke to overcome the power consumed by friction within the engine. Fmep is usually calculated by first determining the Indicated Mean Effective Pressure (Imep)—the maximum horsepower that can be produced from the recorded cylinder pressures. The Brake Mean Effective Pressure (Bmep) is then measured by performing a traditional "dyno" test. The Fmep is calculated by finding the difference between the Imep and the Bmep: Fmep = Imep - Bmep. Fmep can also be directly measured with a motoring (electric) dyno.

Friction—A force that opposes motion. Frictional forces convert mechanical motion into heat.

Full Three-Dimensional (CFD) Simulation—This highly advanced engine simulation technique incorporates multiple models (e.g., thermodynamic, kinetic, etc.), including full three-dimensional modeling that subdivides an area, such as the combustion chamber or port junction, into a series of volumes (or cells) through which the model solves the differential equations of thermodynamics and fluid flow (using Computational Fluid Dynamics—CFD). The interaction of these cells can reveal very subtle design features within the induction and exhaust systems. It can thoroughly evaluate their effect on horsepower, fuel efficiency, and emissions throughout the rpm range. This simulation is not available as a commercial program and it currently remains a "laboratory-only" tool.

Gas-Dynamic Multidimensional Simulation—This engine simulation technique includes multiple models (e.g., thermodynamic, kinetic, etc.), plus powerful finite-wave analysis techniques that account for variations in pressures within individual sections of the ports due to gas dynamic effects. This detailed, highly math-intensive technique can predict engine power with remarkably high accuracy. The Dynomation program from V.P. Engineering ($600; 515-986-9197)—discussed in the complete DeskTop Dynos book—is an example of a program using this simulation method.

Helmholtz Resonator—A device that increases the amplitude of pressure wave through a resonance phenomenon (an effect similar to the deep "whir" produced when air is blown over the neck of a jug). In some cases, the induction system in an IC engine can be modeled by employing Helmholtz resonance equations.

Horsepower—Torque measures how much work (an engine) *can* do, power is the rate-based measurement of *how fast* the work is being done. Starting with the static force applied at the end of a torque arm (torque), then multiplying this force by the swept distance through which the same force would rotate one full revolution finds the power per revolution: Power Per Revolution = Force or

Appendix-B Glossary

Weight x Swept Distance. James Watt (1736-1819) established the current value for one horsepower: 33,000 pound-feet per minute or 550 pound-feet per second. So horsepower is currently calculated as: Horsepower = Power Per Revolution/33,000, which is the same as Horsepower = (Torque x 2 x Pi x RPM)/33,000, or simply: Horsepower = (Torque x RPM)/5,252. The horsepower being calculated by these equations is just one of several ways to rate engine power output. Various additional methods for calculating or measuring engine horsepower are commonly used (to derive friction horsepower, indicated horsepower, etc.), and each technique provides additional information about the engine under test.

Hydraulic Lifter—See Lifters, Hydraulic

IC Engine—See Internal Combustion engine.

Inches of Mercury and **Inches of Water**—A standard method of pressure measurement, where pressures are compared to atmospheric or ambient pressure. Inches of displacement are recorded for a water or mercury column measured in a "U" shaped tube with one end open to the air and the other end connected to the test pressure. Commonly called a manometer, this pressure comparison device is quite sensitive and accurate. When mercury is used in the manometer tube, one psi differential from atmospheric pressure will displace 2.04-inches of mercury. However, when water is the liquid in the "U-tube," a substantial increase in pressure sensitivity is obtained: one psi will displace 27.72 inches of water. A water manometer is used to measure small vacuum and pressure signals.

Indicated Horsepower (Ihp)—is the maximum power that a particular engine can *theoretically* produce. It is calculated from an analysis of the gas pressures measured by installing pressure transducers in the cylinders. Also see Brake Horsepower and Friction Horsepower.

Indicated Mean Effective Pressure (Imep)—A theoretical average pressure that would have to be present in each cylinder during the power stroke to generate the maximum horsepower possible from the pressures recorded within the cylinder of an engine during an actual dyno test. The Imep pressure assumes that the recorded pressures within the test engine will be entirely converted into motive force (with no losses due to friction).

Induction Airflow—The airflow rating (a measurement of restriction) of a carburetor or fuel injection system. Four-barrel carburetors are rated by the measured airflow when the device is subjected to a pressure drop equal to 1.5-inches of Mercury. Two-barrel carburetors are tested at 3.0-inches of Mercury.

Induction System—Consists of the carburetor or injection system and the intake manifold. The intake manifold can be of many designs such as dual plane, single plane, tunnel ram, etc.

Intake Centerline Angle—The distance in crank degrees from the point of maximum intake valve lift (on symmetric cam profiles) to TDC during the valve overlap period.

Intake Stroke—One of the four 180-degree full "sweeps" of the piston moving in the cylinder of a four-stroke, internal-combustion engine (originally devised by Nikolaus Otto in 1876). During the intake stroke, the piston moves from *TDC* to *BDC* and inducts (draws in by lowering the pressure in the cylinder) air/fuel mixture through the induction system. Note: The 180-degree duration of the intake stroke is commonly shorter than the period during which the intake valve is open, sometimes referred to as the true "Intake Cycle." The intake stroke is followed by the compression stroke.

Intake Valve Closing or **IVC**—Considered the most important cam timing event. The point at which the intake valve returns to its seat, or closes. This valve timing point usually occurs early in the compression stroke. Early IVC helps low-end power by retaining air/fuel mixture in the cylinder and reducing charge reversion at lower engine speeds. Late IVC increases high-speed performance (at the expense of low speed power) by allow additional charge to fill the cylinder from the ram-tuning effects of the induction system at higher engine speeds.

Intake Valve Duration—See Duration

Intake Valve Lift—See Valve Lift

Intake Valve Opening or **IVO**—The point at which the intake valve lifts off of its seat, or opens. This valve timing point usually occurs late in the exhaust stroke. Although IVO does not have a substantial effect on engine performance, it contributes to valve overlap (the beginning point of overlap) that can have a significant effect on engine output.

Internal Combustion Engine—An engine that produces power from the combustion and expansion of a fuel-and-air mixture within a closed cylinder. Internal-combustion engines are based on two methods of operation: two cycle and four cycle. In each method, a mixture of fuel and air enters the engine through the induction system. A piston compresses the mixture within a closed cylinder. A precisely timed spark ignites the charge after it is compressed. The explosive burning produces very high temperatures and pressures that push the piston down and rotate the crankshaft, generating a motive force. Also see combustion space, compression ratio, compression stroke, power stroke, exhaust stroke, and intake stroke.

Lifters, Hydraulic Flat-Tappet—A camshaft follower having a flat surface at the point of contact with the cam lobe. Flat-tappet lifters actually have a shallow convex curvature at their "face" to allow the lifter to rotate during operation, extending the working life. A hydraulic lifter incorporates a mechanism that automatically adjusts for small changes in component dimensions, and usually maintains zero lash in the valvetrain. Hydraulic lifters also offer a slight "cushioning" effect and reduce valvetrain noise.

Lifters, Roller Solid Or Hydraulic—A camshaft follower having a round, rolling element used at the contact point with the cam lobe. A hydraulic lifter incorporates a mechanism that automatically adjusts for small changes in component dimensions, and usually maintains zero lash in the valvetrain. Hydraulic lifters also offer a slight "cushioning" effect and reduce valvetrain noise. Solid lifters lack this hydraulic adjusting mechanism and require a running clearance in the valvetrain, usually adjusted by a screw or nut on the rockerarm.

Lifters, Solid Flat-Tappet— A camshaft follower having a flat surface at the point of contact with the cam lobe. Flat-tappet lifters actually have a shallow convex curvature at their "face" to allow the lifter to rotate during operation, extending the working life. Solid lifters lack an automatic hydraulic adjusting mechanism and require a running clearance in the valvetrain, usually adjusted by a screw or nut on the rockerarm. Solid lifter cams usually generate more valvetrain noise than hydraulic-tappet cam.

Lobe-Center Angle or **LCA**—The angle in cam degrees from maximum intake lift to maximum exhaust lift. Typical LCAs range from 100 to 116 camshaft degrees (or 200 to 232 crank degrees).

Multi Dimensional—As it refers to engine simulation programs, multi dimensional indicates that the simulation is based on multiple models, such as thermodynamic and kinetic, plus the multidimensional geometric description of inlet and outlet passages and a dynamic model of induction and exhaust flow.

Negative Pressure—A pressure below atmospheric pressure; below 14.7psi absolute. Very low pressures are usually measured by a manometer. See Inches of Water.

Normally Aspirated—When the air-fuel mix is inducted into the engine solely by the lower pressure produced in the cylinder during the intake stroke; aspiration not aided by a supercharger.

Otto-Cycle Engine—See Four-Cycle Engine

Overlap or **Valve Overlap**—The period, measured in crank degrees, when both the exhaust valve and the intake valve are open. Valve overlap allows the negative pressure scavenge wave to return from the exhaust and begin the inflow of air/fuel mixture into the cylinder even before the intake stroke begins. The effectiveness of the overlap period is dependent on engine speed and exhaust "tuning."

Pocket Porting—Relatively minor porting work performed below the valve seat and in the "bowl" area under the valve head. These changes, while straightforward, can produce a significant improvement in airflow and performance. Proper contours must be maintained, particularly below the valve seat, to produce the desired results.

Porting or **All-Out Porting**—Aggressive porting work performed to the passages within the cylinderhead with intention of optimizing high-speed airflow. Often characterized by large cross-sectional port areas, these ports generate sufficient flow velocities only at higher engine speeds; low speeds produce weak ram-tuning effects and exhaust scavenging waves. This porting technique is a poor choice for low-speed power and street applications.

Pounds Per Square Inch—See PSI.

Power Stroke—One of the four 180-degree full "sweeps" of the piston moving in the cylinder of a four-stroke, internal-combustion engine (originally devised by Nikolaus Otto in 1876). During the power stroke, the piston moves from TDC to BDC as the burning air/fuel mixture forces the piston down the cylinder. Note: The 180-degree duration of the power stroke is commonly longer than the duration between top dead center and the exhaust-valve opening point, sometimes referred to as the true "Power Cycle." The power stroke is followed by the exhaust stroke.

Pressure Crank-Angle Diagram—Also called a *"Pie-Theta* "diagram, the crank-angle diagram is a plot of the indicated cylinder pressures vs. the angular position of the crankshaft during the entire four-cycle process. This diagram provides an easily understood view of the varying pressures in the cylinder. Also see Pressure-Volume Diagram.

Pressure—A force applied to a specific amount of surface area. A common unit of pressure is psi, i.e., pounds per square inch. The force that develops the pressure is sum total of all the slight "nudges" on a surface generated by each molecule striking the surface; the greater number of impacts or the more violent each impact, the greater the pressure. Therefore, the pressure increases if the same number of molecules are contained in a smaller space (greater number of impacts per unit area) or if the molecules are heated (each impact is more violent).

Pressure-Volume Diagram—Also called a *PV* (pronounced *Pee-Vee),* or "*indicator*" *diagram*, the pressure-volume diagram plots indicated pressure against the displaced volume in the cylinder. A PV diagram has the remarkable feature of isolating the *work consumed* from the *work developed* by the engine. The area within the lower loop, drawn in a counterclockwise direction, represents the work consumed by the engine "pumping" the charge into the cylinder and forcing the exhaust gasses from the cylinder. The upper loop area, drawn in a clockwise direction, indicates the work produced by the engine from pressures generated after combustion.

PSI or **Pounds Per Square Inch**—A measure of *force* applied on a surface, e.g., the force on the wall of a cylinder that contains a compressed gas. A gas compressed to 100 psi would generate a force of 100 pounds on each square inch of the cylinder wall surface. In other words, psi equals the force in pounds divided by the surface area in square inches. Also see *pressure*.

Restriction—A measure of the resistance to flow for (usually) a liquid or gas. Exhaust or intake flow restriction can occur within tubing bends, within ports, manifolds, etc. Liquid restriction can occur in needle-and-seat assemblies, fuel pumps, etc. Some restriction is always present in a flowing medium.

Roller Lifter—See Lifters, Roller

Roller Tappet Lifter— See Lifters, Roller

RPM—Revolutions Per Minute. A unit of measure for angular speed. As applied to the IC engine, rpm indicates the instantaneous rotational speed of the crankshaft described as the number of crank revolutions that would occur every minute if that instantaneous speed was held constant throughout the measurement period. Typical idle speeds are 300 to 800rpm, while peak engine speeds can reach as high as 10,000rpm or higher in some racing engines.

Seat-To-Seat Cam Timing Method—See

Cam Timing, @ Seat to Seat.

Simulation and **Engine Simulation**—A engine simulation process or program attempts to predict real-world responses from specific component assemblies by applying fundamental physical laws to "duplicate" or simulate the processes taking place within the components.

Smallblock—A generic term that usually refers to a V8 engine with a displacement small enough to be contained within a "small" size engine block. Typical smallblock engines displace under 400 cubic inches.

Solid Lifter—See Lifter, Solid

Stroke—The maximum distance the piston travels from the top of the cylinder (at TDC) to the bottom of the cylinder (at BDC), measured in inches or millimeters. The stroke is determined by the design of the crankshaft (the length of the stroke arm).

Top Dead Center or **TDC**—The position of the piston in the cylinder bore at its uppermost point in the stroke. Occurs twice within the full cycle of a four-stroke engine; at the start of the intake stroke and 360 degrees later at the end of the compression stroke.

Torque—The static twisting force produced by an engine. Torque varies with the length of the "arm" at which the twisting force is measured. Torque is a force <u>*times*</u> the length of the measurement arm: *Torque = Force x Torque Arm*, where *Force* is the applied or the generated force and *Torque Arm* is the length through which that force is applied. Typical torque values are ounce-inches, pound-feet, etc.

Valve Head and **Valve Diameter**—The large end of an intake or exhaust valve that determines the diameter. Valve head temperature can exceed 1200 degrees F during engine operation and a great deal of that heat is transferred to the cylinderhead through the contact surface between the valve face and valve seat.

Valve Lift Rate—A measurement of how fast (in inches/degree) the camshaft raises the valve off of the valve seat to a specific height. If maximum valve lift is increased but the duration (crankshaft degrees) that the valve is held off of the valve seat is kept the same, then rate at which the valve opens must increase (same time to reach a higher lift). High lift rates can produce more horsepower, however, they also increase stress and valvetrain wear.

Valve Lift—The distance the valve head raises off of the valve seat as it is actuated through the valvetrain by the camshaft. Maximum valve lift is the greatest height the valve head moves off of the valve seat; it is the lift of the cam (lobe height minus base-circle diameter) multiplied by the rockerarm ratio.

Valve Motion Curve or **Valve Displacement Curve**—The movement (or lift) of the valve relative to the position of the crankshaft. Different cam styles (i.e., flat, mushroom, or roller) typically have different displacement curve acceleration rates. Engine simulation programs calculate a valve motion curve from valve event timing, maximum valve lift, and other cam timing specifications.

Volumetric Efficiency—Is calculated by dividing the mass of air inducted into the cylinder between IVO and IVC divided by the mass of air that would fill the cylinder at atmospheric pressure (with the piston at BDC). Typical values range from 0.6 to 1.2, or 60% to 120%. Peak torque always occurs at the engine speed that produced the highest volumetric efficiency.

Work and **Net Work**—Work is the energy required to move an object over a set distance. Both motion and force must be present for work to occur. In an IC engine, work is developed from pressures within the cylinder acting on the face of the piston (producing a force) times the distance through which the piston travels. In the internal combustion engine some of the work produces power output (such as pressures producing piston movement during the power stroke) and some of the work is negative (like compressing the fresh charge on the compression stroke). The difference between the positive and negative work is the net work produced by the engine.

Appendix-C Bibliography

The following books are generally not available in auto stores or "speed shops." These references can be found in well-stocked engineering libraries (particularly at state universities) or at technical book stores. Many of these books are serious engineering works requiring substantial math and physics background for complete understanding, however, all of these books have least some "readable" material that might be of interest to performance enthusiasts.

Anderson, Edwin P., Facklam, Charles G, ***Gas Engine Manual***, G.K. Hall & Co., 70 Lincoln Street, Boston, MA 02111, 800-343-2806. A very readable, non-engineering look at the four and two-cycle internal combustion engine. Includes a detailed look at individual components and their function. Published 1962. ISBN-0-8161-1707-1

Cummins Jr., C. Lyle, ***Internal Fire***, Internal Fire, SAE. In-depth history of the internal-combustion engine, from the early attempts at gunpowder engines in the 1600's to "modern" designs of the early 1900's. Easy, interesting reading. Orig. Published 1976, revised 1989. ISBN 0-89883-765-0

Benson, Roland S., ***The Thermodynamics and Gas Dynamics of Internal-Combustion Engines, Volume 1***, Oxford University Press, New York. Considered by many to be the original "bible" of gas dynamics as applied to the IC engine. This book has been the jumping off point for many engineers and scientists into the world of computer engine modeling. This book is mathematically rigorous and demands much of the reader. A knowledge of differential and integral calculus is required to get the most from this text. Published 1982 for the late Rowland Benson (1925-1978) by J.H. Horlock F.R.S and D.E. Winterbone, editors. ISBN 0-19-856210-1

Ferguson, Colin R., ***Internal Combustion Engines, Applied Thermosciences***, John Wiley & Sons, New York, An interesting combination of "nuts and bolts" and thermodynamics. A much more rigorous look at thermal science (encompasses two thirds of the book) than the design of IC engines. Requires knowledge of differential and integral calculus. Includes some computer code listings, however, calls are made to subroutines that may not be available to many readers. Published 1986. ISBN 0-471-88129-5

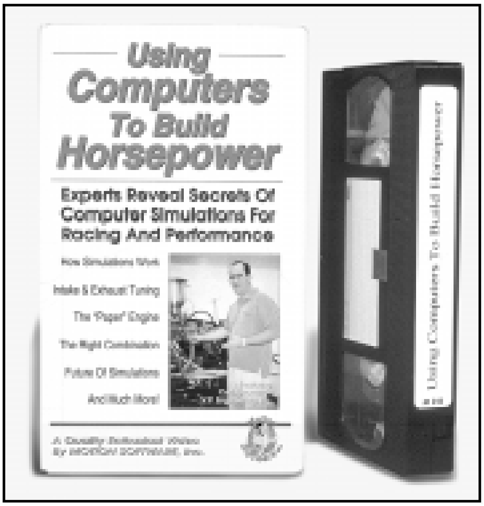
Stone, Richard, ***Introduction To Internal Combustion Engines***, Macmillan Publishing. A text for college-level students. Includes some high-level math, but much of the text is accessible to the advanced enthusiast. Covers thermodynamics (with a quick look at modeling theory), combustion, turbocharging, mechanical design, dynamometers. Published 1985; second edition published 1992. ISBN 0-333-37593-9

Heywood, John B., ***Internal Combustion Engine Fundamentals***, McGraw-Hill, Inc. A very thorough text for researchers and professionals. Includes both in-depth technical treatments (with thorough mathematical analysis) and a considerable amount of material that should be accessible to many enthusiasts. Nearly 1000 pages with many fascinating photos and drawings of research work. Published 1988. ISBN 0-07-028637-X

Markatos, N.C., ***Computer Simulation For Fluid Flow, Heat And Mass Transfer, And Combustion In Reciprocating Engines***, Hemisphere Publishing Corp., New York. A book that evolved from a course on computer simulation for fluid flow held in Dubrovnik, Yugoslavia in September 1987. Heavy mathematical treatment; very little light reading here. Published 1989. ISBN 0-89116-392-1

Ramos, J.I., ***Internal Combustion Engine Modeling***, Hemisphere Publishing Corp., New York. A text intended for researchers in fluid mechanics, combustion, turbulence and heat transfer at the graduate level. Rigorous mathematical treatment includes rotary, diesel, two- and four-stroke spark ignition engines. Published 1989. ISBN 0-89116-157-0

Ganesan, V., ***Internal Combustion Engines***, Tata McGraw-Hill Publishing Company Limited. A book written by V. Ganesan, a Professor of Mechanical Engineering at the Indian Institute of Technology in Madras, India. Readable text intended for the student beginning serious study of the I.C. engine. Good balance between textual explanations and mathematical/physics study. Contains many solved problems. First published 1994. ISBN 0-07-462122-X



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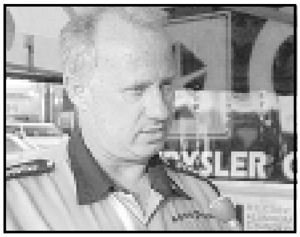

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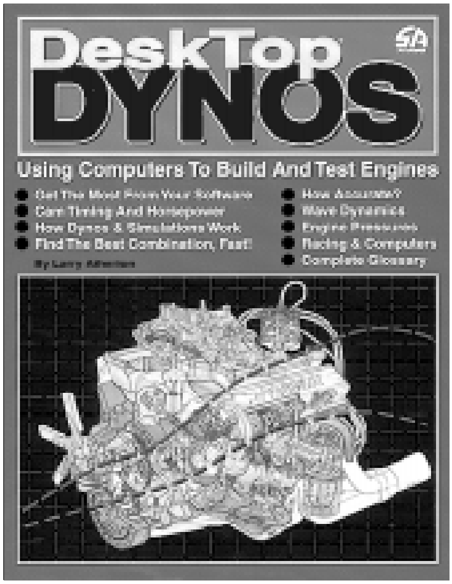

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  - **FILLING-AND-EMPTYING MODELS**
  - **GAS-DYNAMIC MODELS**
  - **MODERN ENGINE MODELING AND THE PC**
  - **DYNOMATION:**
  - **GLEAM IN A YOUNG MAN'S EYE**
  - **THE FUTURE OF IC MODELING**
- **GAS DYNAMICS AND DYNOMATION**
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