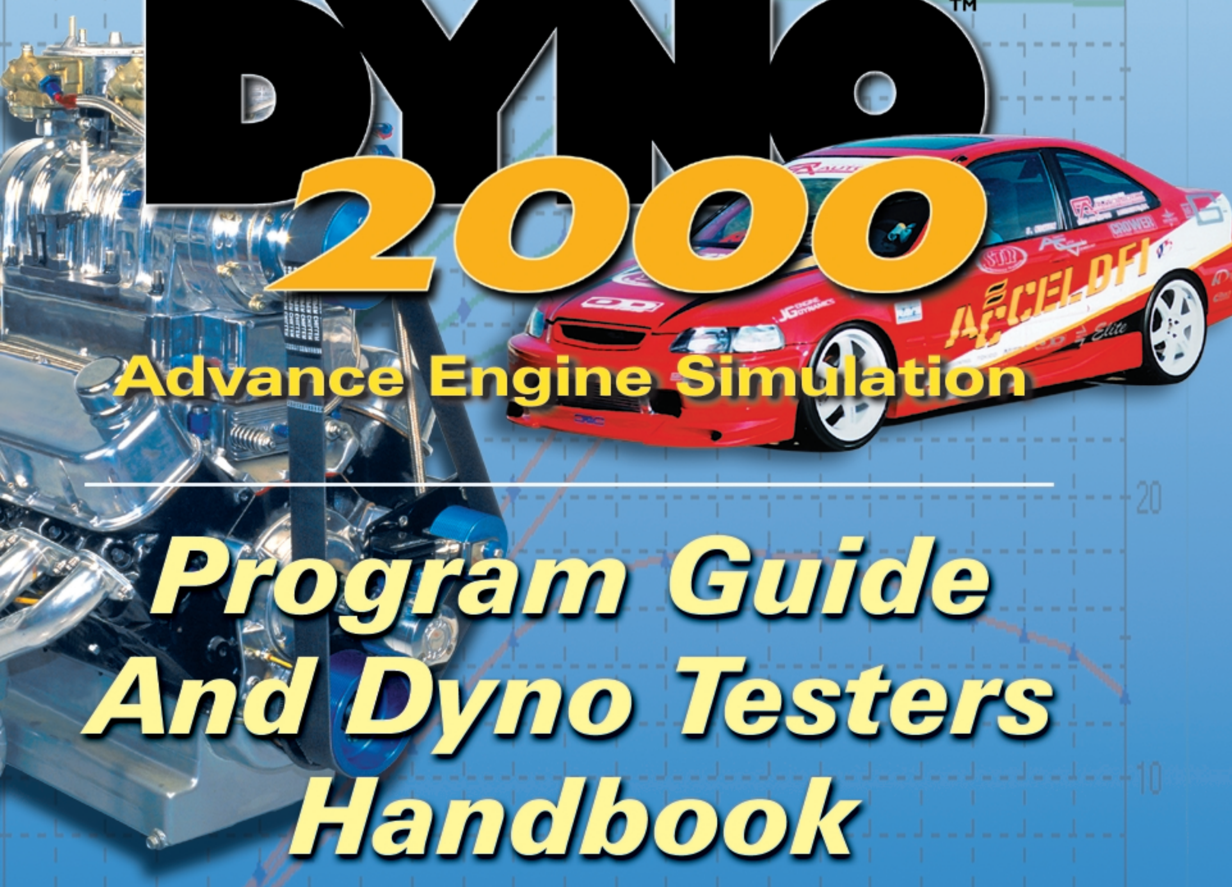
DYNO™ 2000

Advance Engine Simulation



Program Guide And Dyno Testers Handbook

Incorporates MotionPC™ Simulation Technologoy

MOTION PC SIMULATION

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Dyno2000 Simulation v3.10, 5-/01 Release 5

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Curtis Leaverton, Simulation Designer. My friend and college, Curtis Leaverton is the “brains” behind the Dyno2000. His engine computer simulations have changed the way performance enthusiasts approach engine building.

Brent Erickson, Simulation Designer, Programmer. Developed new simulation models for the Dyno2000. A brilliant programmer, Brent’s positive “can-do” attitude is backed up by his ability to accomplish what many dismiss as impossible.

Trent Noller, Marketing/Sales Manager. Trent excels at problem solving and there were more than a few problems that required his creative skills during the development and deployment of the Dyno2000. My friend for many years, Trent Noller can, simply, be credited with the success of the Dyno2000.

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And thanks to the many other individuals at Mr. Gasket who have contributed to the success of the DeskTop line, including:

Don McGee, Kirk Tinney, Mike Roth and others too numerous to mention.

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DYNO 2000™ Advanced Engine Simulation

INTRODUCTION

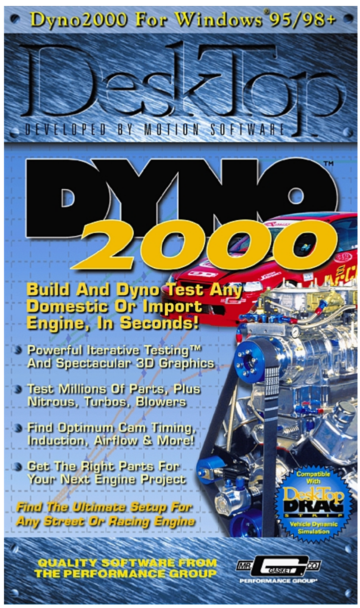
Note: *If you can't wait to start the Dyno2000™, feel free to jump ahead to* **INSTALLATION** on page 10, *but don't forget to read the rest of this manual when you have time. Also, make sure you mail in your registration card—it entitles you to receive a FREE upgrade and other information and support.*

Thank you for purchasing the Dyno2000™ for IBM®-compatible computers. This software is the result of several years of development and testing. It is just one of several quality software products developed by Motion Software, Inc., that can further your understanding and enjoyment of automobiles, performance, and racing technology.

HOW IT WORKS

The Dyno2000 is a Windows95/98 and WindowsNT/2000, 32-bit program based on the *Filling-And-Emptying* method of engine power simulation. We chose this family of mathematical models because of their excellent power prediction accuracy and fast processing times. The Dyno2000 is a *full-cycle* simulation. This means that it calculates the complete fluid-dynamic, thermodynamic, and frictional conditions that exist inside each cylinder throughout the entire 720 degrees of the four-cycle process.

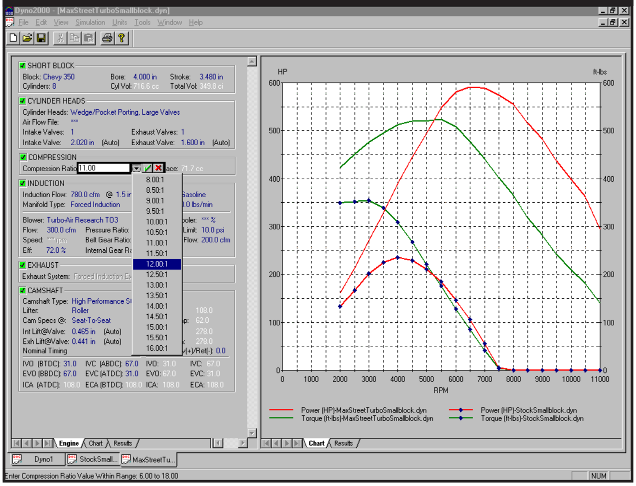
You will find that many other simulation programs on the market (even a few that sell for several times the price of the Dyno2000) are not true engine *simulations*. Rather, they calculate the volumetric efficiency (VE) and then derive an estimate



The Dyno2000 is the most advanced engine simulation ever offered to the performance enthusiast. It combines ease of use, rapid calculation times, powerful Iterative Testing™, and detailed graphics. The Dyno2000 is available from Mr. Gasket Performance and Motion Software, Inc.

Introduction To The Dyno2000

Dyno2000 Main Component Screen



The Dyno2000 incorporates a very clean, intuitive user interface. If you wish to change a component, simply click on the component name and select a new component from the drop-down list. A comprehensive data display is fully customizable. Multiple engine and/or data value comparisons are possible. All components and graphics displays can be printed in full color.

of torque and horsepower. There are many shortcomings to this technique. The two greatest drawbacks are: 1) since cylinder pressure is not determined, it is impossible to predict the pressure on the exhaust valve and the subsequent mass flow through the port when the exhaust valve opens, and 2) the inability to accurately determine the pumping horsepower (energy needed to move gasses into and out of the engine) from the predicted horsepower.

Since the Dyno2000 incorporates both filling-and-emptying *and* full-cycle modeling that includes frictional and pumping-loss calculations, extensive computation is required for each power point. In fact, the program performs several million calculations at each 500rpm test point on the power curve (a full power-curve simulation consists of 27 test points). This in-depth analysis offers unprecedented accuracy over a vast range of engines. The Dyno2000 has been successfully used to model single-cylinder “lawn mower” engines, light aircraft engines, automotive engines, modern Pro Stock drag-racing powerplants, and multi-thousand horsepower supercharged, nitrous-oxide injected “mountain motors.”

WHAT'S NEW IN THE DYNO2000

The Dyno2000 features a completely unique, easy-to use, point-and-click interface. Just click on any component, and drop-down menus offer alternative selections. Hundreds of components are available, including a wide selection of import engines. Instantly change between US and Metric measurements.

The Dyno2000 also models of forced induction systems, including turbocharging and roots/centrifugal supercharging. Set maximum boost, belt ratios, efficiencies, and more! Even model intercoolers.

Test engine power with alternate fuels, including Methanol, Ethanol, Propane,

Introduction To The Dyno2000

LNG, and even Nitrous Oxide injection. Graph cylinder pressures, frictional losses, and other engine variables.

And the Dyno2000 is the only engine simulation with exclusive *Iterative Testing™* that analyzes thousands of dyno tests, keep track of all the results, and displays the best setup for virtually any application, all automatically! Combine this power with uniquely versatile graphing capabilities, and the Dyno2000 is, simply, the best engine simulation you can buy. In fact, you will find no other software, even at many times the price, that offers so much capability and performance.

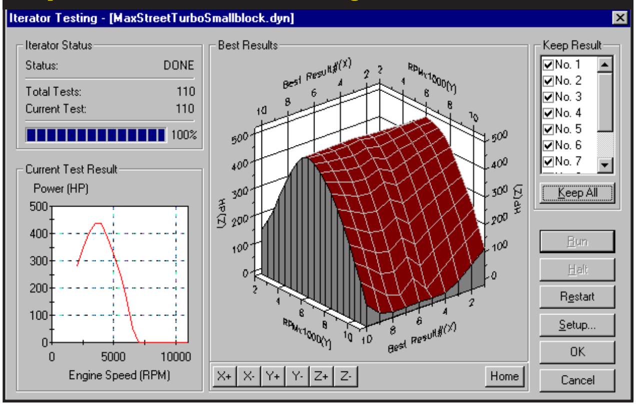
DYNO2000 REQUIREMENTS

The following list presents an overview of the basic hardware and software required to run the Dyno2000.

Minimum Requirements Overview:

- An IBM compatible “PC” computer with a CD-ROM drive
- At least 16MB of RAM (random access memory) for Windows95/98; 32MB for WindowsNT; 64MB for Windows2000
- Windows95/98 or Windows NT/2000 (recommend NT version 4.0 with SP4 or later)
- A video system capable of at least VGA (640 x 480 resolution). Recommend 800 x 600 or higher to optimize screen display and engine analysis
- A Pentium 200 or similar processor (Pentium II or III or faster processors will improve processing speeds; especially helpful for Iterative analysis)
- A mouse
- A printer (needed to obtain dyno-test printouts).

Dyno2000 *Iterative Testing™* Screen



***Iterative Testing™* is a powerful feature of the Dyno2000. This screen illustrates a test that just evaluated a series of components (over 100 dyno tests were performed). Using this powerful tool it is possible to automatically run thousands or even hundreds of thousands of tests to find the best combinations. The Dyno2000 keeps track of all the results and displays the best matches to your test criterion.**

Introduction To The Dyno2000

REQUIREMENTS IN DETAIL

Computer: An IBM-compatible “PC” computer with a CD-ROM disk drive is required. The Dyno2000 will operate on any computer system with an Intel-compatible processor, however, a Pentium-class microprocessor is recommended to minimize calculation times (Pentium II or III 400+Mhz processors will improve processing speeds; especially helpful for *Iterative* analysis where hundreds or thousands of dyno tests can be performed in a continuous series).

Windows95/98 and NT/2000: The Dyno2000 is a full 32-bit program designed for Windows95, Windows98 and later versions of Windows using the Win95 kernel. The Dyno2000 is also compatible with WindowsNT versions 3.51 or later and Windows2000 (Motion recommends that if you use WindowsNT, use version 4.0 with service pack 4 or later; and if you use Windows2000, make sure to install the latest service pack for both Windows2000 and for Internet Explorer).

System Memory: Your system should have a minimum of 16Mbytes of physical RAM memory for Windows95/98, 32Mbytes for WindowsNT, and 64Mbytes for Windows2000. The Dyno2000 may not operate on systems with less installed memory. To optimize Windows and Dyno2000 performance, 64Mbytes or more is recommended.

Video Graphics Card And Monitor: Virtually any Windows compatible monitor and display card will work with the Dyno2000. Systems with SVGA or better graphics (800 x 600 resolution or higher) provide more screen “real estate.” This additional display space is very helpful in component selection and power-curve analysis.
Note1: See FAQ on page 100 for help in changing the screen resolution of your monitor.
Note2: Specialized graphics cards and ultra-high resolution “workstation” displays may not be compatible with Dyno2000. If you encounter display incompatibilities with the Dyno2000, please contact Motion Software Tech Support, 535 West Lambert, Bldg. E, Brea, CA 92821-3911, 714-255-2931, or visit our website: www.motionsoftware.com.

System Processor: The Dyno2000 is extremely calculation-intensive. Over 25 million mathematical operations are performed for each complete power-curve simulation. While the program has been written in fast C++ and hand-tuned assembler to optimize speed, a faster processor will improve data analysis capabilities. Furthermore, the Dyno2000 incorporates a powerful *Iterative Tester* that can perform an analysis of hundreds of thousands of dyno tests. To reduce these calculation times and extend the modeling capabilities of the program, use the fastest processor possible.

The following table gives an approximation of the time required to complete a

Introduction To The Dyno2000

100 dyno-run *Iterative* test on various PC systems (this is a very short run; *Iterative* tests can consist of hundreds of thousands of simulated dyno runs or more):

Computer	Coprocessor	Calc. Time For 100-Test Run
Pentium 400Mhz	Built-In	17 Seconds
Pentium 200Mhz	Built-In	75 Seconds
Pentium 133Mhz	Built-In	112 Seconds
Pentium 60Mhz	Built-In	4.3 Minutes
80486DX 33Mhz	Built-In	13.5 Minutes
80386DX 25Mhz	Yes (added)	49 Minutes
80486SX 25Mhz	No	6.4 Hours
80386DX 33Mhz	No	9.4 Hours
80286 at 10Mhz	No	24 Hours
8088 at 8Mhz	Yes (added)	3.2 Hours

Mouse: A mouse (trackball, or other pointer control) is required to use the Dyno2000. While most component selections can be performed with the keyboard, several operations within the Dyno2000 require the use of a mouse.

Printer: The Dyno2000 can print a comprehensive “dyno-test report” of a simulated dyno run with any Windows-compatible printer. If you use a color printer, the data curves and selected information will print in color (see page 81 for more information about Dyno2000 printing).

DYNO™ 2000 Advanced Engine Simulation

INSTALLATION

Helpful Installation Tips

Dyno2000 installation is a quick and easy on virtually all computers. To minimize the likelihood of problems, review the following tips before you begin:

1) The Dyno2000 requires Windows 95/98® or Windows NT/2000® and at least 16MB of installed memory (see pages 7-8 for more information about system requirements).

2) The entire installation of the Dyno2000 and DeskTop Videos requires 110MB of free disk space. If you do not wish to install the Software Videos, select the “Compact” option presented during the installation process.

3) If at all possible, install the software onto the (default) drive and directory suggested by the SETUP program. This will speed the process of installing Dyno2000 software updates in the future.

Installing The Dyno2000

The installation programs included with the Dyno2000 will copy the appropriate files to your hard drive. Please read and perform each of the following instructions carefully.

1) Start Windows95/98 (or Windows NT/2000), if necessary.

2) Insert the Dyno2000 CD-ROM into your CD drive.

3) An installation Welcome screen will appear on your desktop within 5 to 30 seconds (depending on the speed of your CD drive). Proceed to **step 5**.

4) If the Dyno2000 installation Welcome screen does not automatically display on your desktop after 30 to 60 seconds, run the **Setup** program included on the Dyno2000 CD-ROM. (Open the *Windows Explorer*, switch to your CD Drive, then double click on **Setup**. Alternatively, choose ***Settings*** from the ***Start*** menu,

select ***Control Panels***, the double click on ***Add/Remove Programs***, finally click on ***Install***.)

5) Click ***Next*** to proceed to the second Installation screen. Click ***Next*** again to review the Motion Software License Agreement. Read the Agreement and if you agree with the terms, click ***Next*** to continue with the installation.

6) Enter your name and company name in the **User Information** screen (only enter your company name if the Dyno2000 is being registered to your company). Click ***Next*** again to continue the installation.

7) The **Choose Destination Location** window will suggest **C:\Dyno2000** as the installation path. We recommend that you accept this default. However, if you prefer another location for the Dyno2000, click on ***Browse...*** to select a new path. When you are finished, click on ***Next*** to continue the installation.

8) The **Setup Type** window will present three installation options:
Typical—Installs Dyno2000, sample files, user manual, and software videos.
Compact—Installs Dyno2000, sample files, and user manual only.
Custom—Allows you to select the installed elements.
We recommend you select **Typical**, then press ***Next*** to continue the installation.

9) The **Select Program Folder** screen indicates that the **Dyno2000** program folder will be added to the list of Windows Program choices displayed on the ***Start***, ***Programs*** menu. You may change the name of the program folder. Press ***Next*** to continue.

10) The **Start Copying Files** screen gives you a chance to review all the installation choices that you've made. Press ***Back*** to make any changes; press ***Next*** to being copying files to your system.

11) When main installation is complete, the **Setup Complete** screen provides a checkbox option (defaults unchecked) that allows you to start the Dyno2000 immediately after installation. (Note: If you do not check this box and click ***Finish***, you can start the Dyno2000 at any time by selecting ***Programs***, ***Dyno2000 Engine Simulation*** from your Windows ***Start*** menu.) Click ***Finish*** to complete the installation.

Starting The Dyno 2000

12) To start the Dyno2000, open the Windows ***Start*** menu, select ***Programs***, then choose ***Dyno2000 Engine Simulation***, and finally click on the ***Dyno2000 Engine Simulation*** icon that opens from the folder.

Installing & Starting The Dyno2000

13) A video of the new DeskTop DragStrip2000, has been included with the Dyno2000. Start the demo by opening the ***Start*** menu, select ***Programs***, then choose the ***Dyno2000 Engine Simulation*** folder, finally click on ***DragStrip2000 Demo NEW***.

14) You can also access considerable additional information on the DeskTop software line and technical support by opening the ***Start*** menu, select ***Programs***, then choose the ***Dyno2000 Engine Simulation*** folder, finally click on ***DeskTop Software Info***.

15) Please review the remainder of this user guide for more information on menu selections, program functions, and simulation tips.

16) If you have installation problems with the Dyno2000, please review program requirements on pages 7-9, and take a few minutes and look over the following sources of information before you contact technical support:

- The FAQs starting on page 100 in this booklet contain detail installation and operational questions and answers.

- Visit the Tech Support section of the Motion Software website for additional tips and FAQs.

If you cannot find a solution to your problem, use the fax-back form in this manual. Fax or mail the completed form to:

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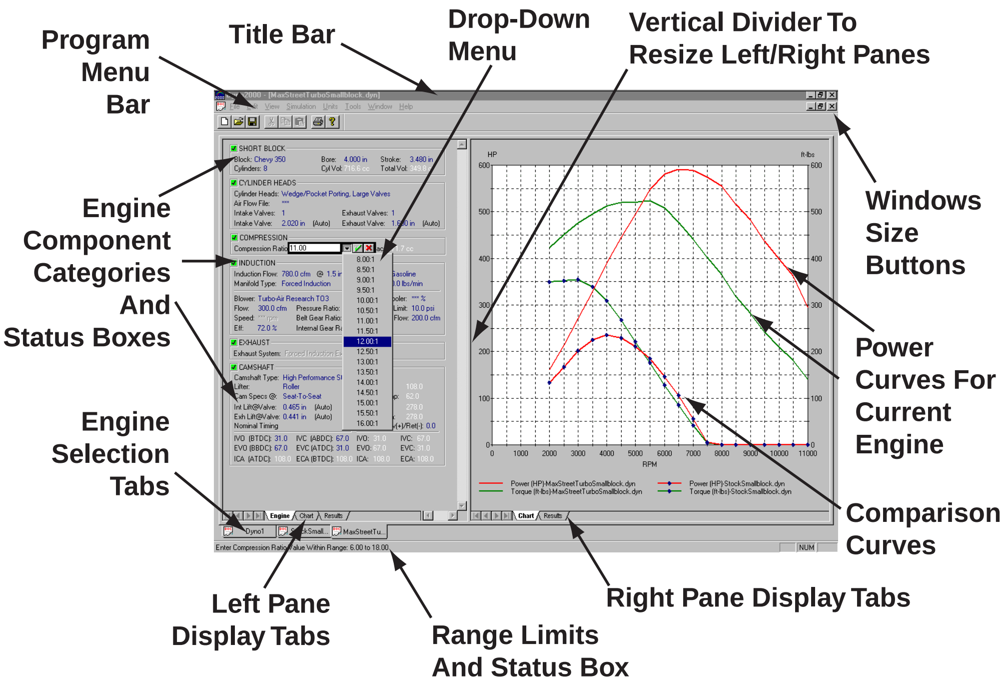
DYNO 2000™ Advanced Engine Simulation

OVERVIEW



THE MAIN PROGRAM SCREEN

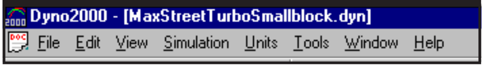
The **Main Program Screen** allows you to select engine components, dimensions, and specifications. In addition, engine power curves and/or simulation data is displayed in graphical and chart form. The Main Program Screen is composed of the following elements:

1) The **Title Bar** displays the program name followed by the name of the currently-selected engine.

2) The **Program Menu Bar** contains eight pull-down menus that control overall program function. Here is an overview of these control menus, from left to right

Program Overview

Program Menu Bar



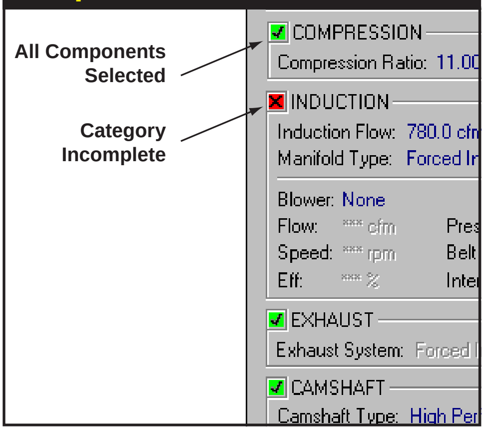
Program Menu Bar contains eight pull-down menus that control overall program function.

(detailed information on menu functions is provided in the next section, beginning on page 20):

File—Opens and Saves dyno test files, exports DOS Dyno files to other DeskTop software, prints engine components and power curves, allows the quick selection of the most recently used Dyno files, and contains an exit-program function.
Edit—Clears all component choices from the currently-selected engine (indicated by the *Engine Selection Tab* currently in the foreground; see **Engine Selection Tabs**, below).
View—Allows you to turn the ***Toolbar***, ***Status Bar*** and ***Workbook*** layout on (default) or off.
Simulation—***Run*** forces an update of the current simulation. ***Auto Run*** enables or disables (toggles) automatic simulation updates when any engine component is modified.
Units—Selects between US and Metric units.
Tools—Opens the *Iterative Testing* window or selects one of the build-in, engine-math calculators.
Window—A standard Windows menu for arranging and selecting engine display windows.
Help—Gives access to this Users Guide, and other program help features.

3) The **Engine Component Categories** are made up of the following groups:

Component Status Boxes



A Status Box is located in the upper left corner of each Component Category. These boxes either contain a red boxed X, indicating that the category is not complete (inhibiting a simulation run), or a green-boxed check-mark ✔, indicating that all components in that category have been selected

Program Overview

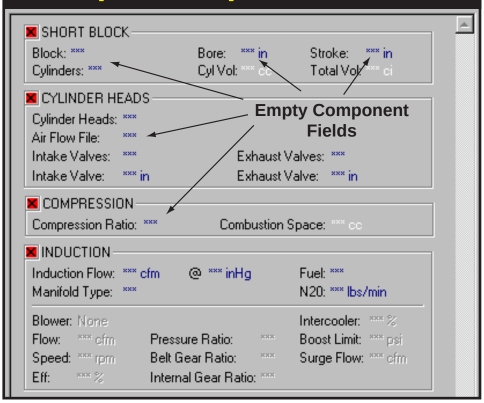
SHORTBLOCK—Select the bore, stroke, and number of cylinders in this category (see page 20).
CYLINDER HEADS—Select the cylinder head type, port configuration, and valve diameters. Direct entry of flow-bench data is also supported (see page 22).
COMPRESSION—Select the compression ratio (see page 30).
INDUCTION—Selects the airflow rate through the induction system, the type of fuel, nitrous flow rate, intake manifold, and a forced induction system (see page 38).
EXHAUST—Selects the exhaust-system configuration (see page 59).
CAMSHAFT—Selects the camshaft type, lifter type, and allows direct entry of valve timing and lift data (see page 65).

Note: Each component category contains a **Status Box** located in the upper left corner. These boxes either contain a red boxed **X**, indicating that the category is not complete (inhibiting a simulation run), or a green-boxed check-mark ✔, indicating that all components in that category have been selected. When all component categories have green checks, a simulation will be performed using the current data values and the results will be displayed in the graph on the right pane of the Main Program Screen (the simulation run and data plot will occur automatically providing **Autorun** is checked in the **Simulation** drop-down menu [default], see **Simulation Menu** described on the previous page).

4) The **Drop-Down Component Menus** contain components and specifications for each of the Component Category choices. Click on any component specification to open its menu. The menu will close when a selection is complete. If you wish to close the menu before making a new selection, click the red **X** next to the drop-down box or press the **Escape** key until the menu closes.

Component fields that do not yet contain valid entries are marked with a series of asteristics. This indicates that the field is empty and can accept data input. Most numeric fields accept direct keyboard entry or selections from provided drop-down menus. Text selection fields (like the Cylinder Head choice menu) only accept selections from the associated drop-down menu. When a valid selection has been made, it will replace the asteristics and be displayed next to the field names.

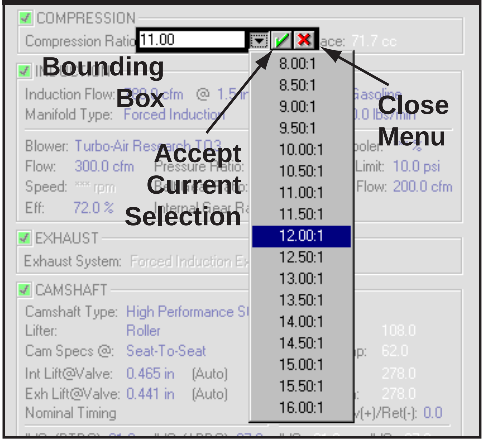
Incomplete Component Fields



Program Overview

The Direct-Click™ Component Menus contain components and specifications for each Component Category choice. Click on any component specification to open its menu. The menu will close when a selection is complete (or accept the current selection by clicking on the green ✔). If you wish to close the menu before making a new selection, click the red X next to the drop-down box or press the Escape key until the menu closes.

Direct-Click™ Component Menus



5) Several Component Category menus allow direct numeric entry. During this data entry, the range of acceptable values will be displayed in a **Range Limit Line** within the **Status Box** at the bottom of the screen.

6) The Dyno2000 can simulate several engines at once. Switch between "active" engines by selecting any Tab from the **Engine Selection Tabs**, just above the **Status Box** (see photo, page 13). The currently-selected engine is indicated on the foreground Tab. The name of the currently-selected engine is also displayed in the **Title Bar**.

7) The Main Program Screen window is divided into two panes (the width of these panes is adjustable; drag the vertical screen divider to resize). Each pane contains a **Screen Display Tab** group. Use these tabs to switch the display in each pane to component lists and other data displays.

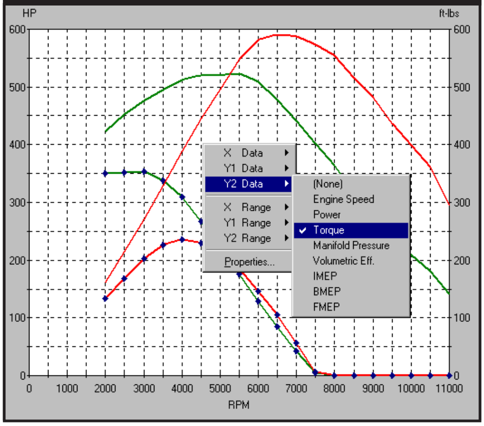
8) The **Current Engine Power Curves** window displays the horsepower and torque for the currently-selected engine. Horsepower and torque are the default curves, however, any graphic data display can be changed by right-clicking on the graph and reassigning each curve in the **Graph Options Box**. Use **Properties...** in the Options Box setup list to create comparisons between any "active" engines.

9) The Main Program Screen also incorporates **Windows Size Buttons**. These buttons provide standard maximizing, minimizing, and closing functions common to all windows. Refer to your Windows documentation for more information on the use of these buttons.

Program Overview

The Right-Hand Power Curves Box displays the horsepower and torque for the currently-selected engine. Horsepower and torque are the default curves, however, the data displayed can be modified by right-clicking on the graph and reassigning each curve in the Graph Options Box**. In addition, you can use the** Properties... **choice available at the bottom of the** Options Box **to setup comparisons between any "active" engine. Note: A second, Left-Hand graph is available under the component selection screen (to activate this display, use the Left-Pane tabs at the bottom of the component screen).**

Graph Options Box



USING THE MOUSE OR KEYBOARD TO BUILD A TEST ENGINE

Begin using the Dyno2000 by "assembling" a test engine from component parts. For example, select a bore and stroke by using the **Block** pull-down menu. Activate the menu by:

Mouse

1) Start the Dyno2000 or select ***New*** from the **File** menu. All component categories start off empty, indicated by strings of asterisks (****) next to each incomplete selection.

2) Move the mouse cursor into the SHORTBLOCK category and double click the left mouse button on the asterisks in the Block component category.

3) When the component-menu bounding box appears (see photo, page 16), click on the ▼ symbol to open the Shortblock selection menu.

4) Move the mouse pointer through the menu choices.

5) When a submenu opens, move the mouse cursor over your selected choice in the submenu.

6) Click the left mouse button on your selection. This loads the engine name, bore, stroke, and number of cylinders into the SHORTBLOCK category. Note that the red boxed **X** (Status Box) on the left of the SHORTBLOCK category changed to a green-boxed check-mark ✔, indicating that all components in that category

have been selected.

7) Alternatively, to close the menu without making a selection, click the red **X** on the right of the bounding box or press the **Escape** key until the menu closes.

8) Continue making component selections until all the category Status Boxes have switched to green. At this point an engine simulation will be performed and the results will be displayed on the graph or chart on the right of the Main Program Screen.

Keyboard

1) Press and release the **Alt** key followed by the **F** key to highlight and open the File menu. Use the cursor-arrow keys to select ***New***, then press **Enter** to create a new, blank component screen. All component categories start off empty, indicated by strings of asterisks (********) next to each incomplete component selection. **Note:** You can activate other menu choices—e.g., *Edit*, *View*, *Simulation*, etc., by pressing the **Right-Arrow** or **Left-Arrow** keys or by using the menu shortcuts (e.g., open the *Edit* menu by pressing **Alt E**).

2) A component menu bounding box is positioned around the **Block** choice in the SHORTBLOCK category.

3) Press Enter to activate the box. Then press Tab to move the highlight (focus) to the ▼ symbol. Then press the **Spacebar** to open the **Block** selection menu.

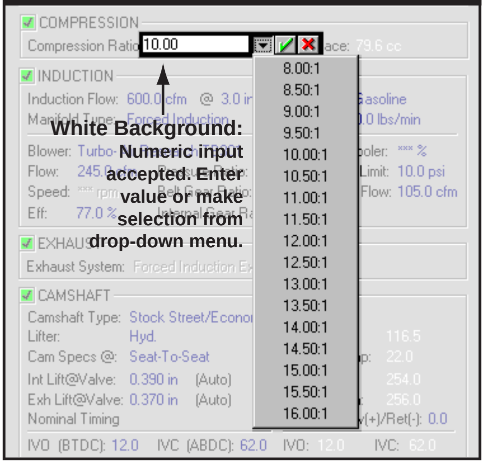
4) Use the **Up-Arrow** or **Down-Arrow** keys to scroll through the menu choices. When the menu selections include submenus (a small arrow points to the right at the end of the menu line), use the **Right-Arrow** key to open the submenu.

5) When you have highlighted your choice, press **Enter** to make the selection. This loads the engine name, bore, stroke, and number of cylinders into the SHORTBLOCK category. Note that the red boxed **X** (Status Box) on the left of the SHORTBLOCK category changed to a green-boxed check-mark ✔, indicating that all components in that category have been selected.
Note: Alternatively, to close the menus without making a selection, press the **Escape** key.

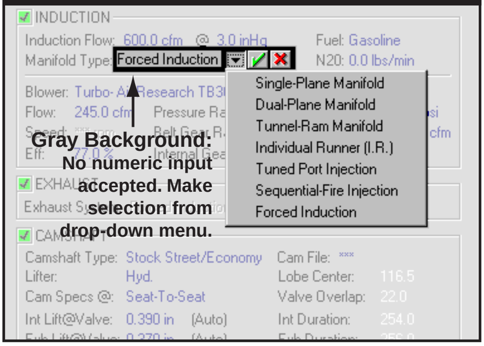
6) Use the **TAB** key to move the component-selection bounding box to the next blank field (Compression Ratio). Continue making component selections until all the main component category Status Boxes have changed to green. At this point an engine simulation will be performed and the results will be displayed on the graph or chart in the right pane of the Main Program Screen.
Note: The **Shift Tab** key combination will move the bounding box backwards to the previous component field.

Program Overview

Fields Accepting Direct Input



Fields Not Accepting Direct Input



DIRECT-ENTRY MENU CHOICES

The Bore, Stroke, Number Of Cylinders, Valve Size, Compression Ratio, Induction Airflow, and several other menus permit direct numeric entry. When a component field supports direct entry, the bounding box will have a white interior. If the only entry possible is a choice from the drop-down menu, the bounding box will have a gray interior (see above photos). Choosing a new numeric value will replace the currently displayed value. When you press **Enter** the new value will be tested for acceptability, and if it passes, it will be used in the next simulation run. If you press **Enter** without entering a new value, the currently displayed value is left unchanged.

Data entry into any field in the component-selection screen is limited to values over which the Dyno2000 can accurately predict power. The range limits are displayed in the **Range Limit Line** within the **Status Box** at the bottom-left of the Main Program Screen. If you enter an invalid number, the Dyno2000 will play the Windows error sound and wait for new input.

THE MEANING OF SCREEN COLORS

The colors used on the component-selection screen provide information about various engine components and specifications. Here is a quick reference to screen color functionality:

White Numeric Values: White engine specifications indicate that they are automatically calculated by program and cannot be directly altered.

Dark Blue: All engine specifications that can be changed by the user through pull-down menus are displayed in dark blue.

DYNO 2000 Advanced Engine Simulation

COMPONENT MENUS

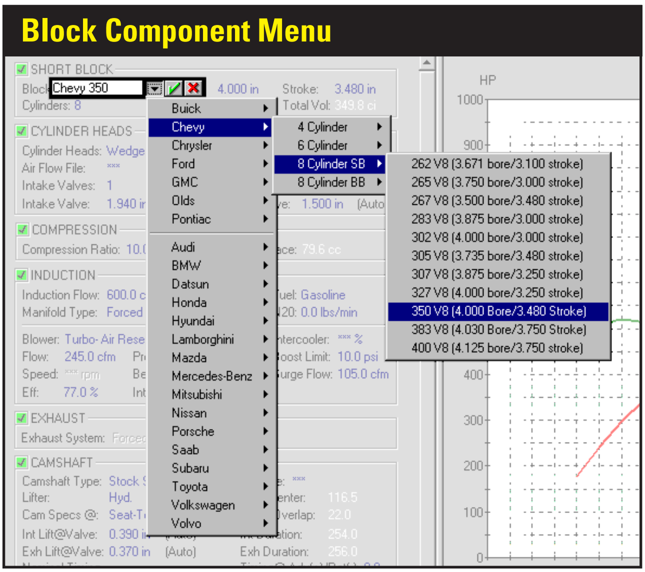
THE BORE, STROKE, AND NUMBER-OF-CYLINDER MENUS

The **Block** menu is located on the upper-left of the SHORTBLOCK component category on the Main Program Screen. By opening this menu, you are presented with a variety of domestic and import “pre-defined” engine shortblock configurations. If any one of these choices is selected, the appropriate bore, stroke, and number of cylinders will be loaded in the SHORTBLOCK category. In addition to selecting any predefined engine configuration, you can directly enter any **Block** name, **Stroke**, **Bore**, and **Number Of Cylinder** numeric values (within the acceptable range limits of the program indicated at the bottom of the screen in the **Status Bar**).

What’s A SHORTBLOCK

When a particular engine combination is selected from the Block menu, the bore, stroke, and the number of cylinders are “loaded” into the SHORTBLOCK category. These values are subsequently used in the simulation. The SHORTBLOCK menu

The Block component menu contains over 200 bore and stroke combinations of popular domestic and import engines that you can instantly use in a simulation.



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 cylinder heads (hemi vs. wedge) or any other engine characteristics. The Bore/ Stroke menu only loads **Bore**, **Stroke**, and the **Number Of Cylinders** into the program database.

Bore And Stroke And Its Effects On Compression Ratio

After making a Bore, Stroke, and Number-Of-Cylinder selection, the swept cylinder volume and the total engine displacement will be calculated and displayed in the SHORTBLOCK component category. 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 values required in calculating compression ratio. We'll discuss compression ratio in more detail later, but for now let's take a quick look at how compression ratio is calculated:

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

The total volume that exists in the cylinder when the piston is located at BDC (this volume includes the Swept Volume of the piston plus the Combustion Space Volume) is divided by the remaining volume that exists when the piston is positioned at Top Dead Center.

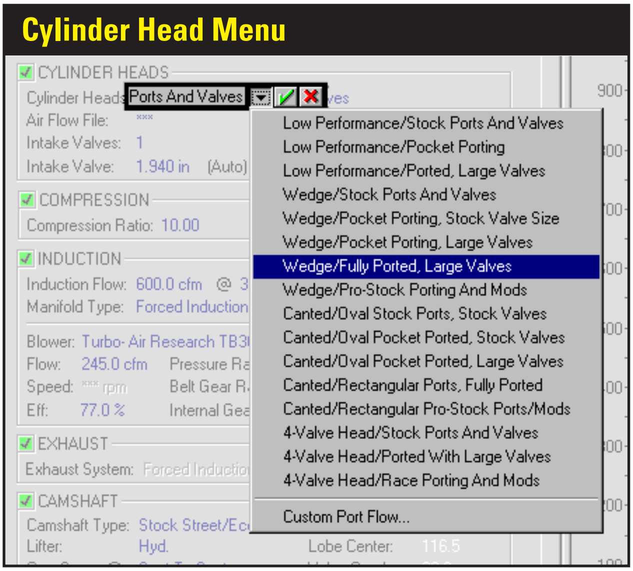
Bore and stroke dimensions greatly affect cylinder volumes and, therefore, compression ratio. When the stroke, and to a lessor degree the bore, is increased while maintaining a fixed combustion-space volume, the compression ratio will rapidly increase. And, as is the case in the Dyno2000 simulation, if the compression ratio is held constant—because it is a fixed component selected by you—the <u>combustion space</u> volume (not necessarily the same as the combustion-chamber volume, see page 31) must increase to maintain the desired compression ratio.

This may seem more understandable when you consider that if the combustion-space volume did <u>not increase</u>, 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.

THE CYLINDER HEAD AND VALVE DIAMETER MENUS

The **Cylinder Head** pull-down menu is located in the CYLINDER HEAD category, and selection from this menu allows the Dyno2000 to simulate various cylinder head designs and a wide range of airflow characteristics. The menu lists general cylinder head characteristics, including restrictive low-performance ports, typical wedge- and

Cylinder Head Menu



The Cylinder Head menu contains a wide range of head/port choices, from stock to all-out racing. In addition, Custom Port Flow allows the direct entry of flow bench data. This feature allows the simulation and testing of any cylinder head for which flow data is available.

canted-valve configurations, and 4-valve cylinder heads. Each type of head/port includes several stages of modifications from stock to all-out race configurations.

In addition, the **Custom Port Flow** choice at the bottom of the Cylinder Head menu allows the direct entry of flowbench data, allowing the Dyno2000 to model any cylinder head for which flow data is available. This option will be described in more detail later.

Basic Flow Theory

A selection from the Cylinder Head menu is the first part of a two-step process used by the simulation to accurately model cylinder head flow characteristics. The initial cylinder head selection determines the airflow <u>restriction</u> generated by the ports. That is, this choice establishes *how much less air than the theoretical maximum peak flow will pass through each port*. What determines peak flow? That's selected from the remaining CYLINDER HEAD category menus: **Intake** and **Exhaust Valve Diameter** Menus. The valve-diameter menus allow the selection of valve sizes that fix the theoretical peak flow (called *isentropic* flow) of each port. Most cylinder heads flow only about 50% to 70% of this value.

Note: You can enable the **Auto Calculate Valve Size** feature to allow the Dyno2000 to automatically determine valve diameters based on bore size and the degree of cylinder head porting/modifications. The various **Cylinder Head** menu choices load airflow data into the simulation, but this flow data is <u>not directly</u> used to determine the airflow capacity of the cylinder heads.

There are several reasons for this. 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 steady-state flow, measured at a fixed pressure drop (it's also dry flow, but a discussion of that feature 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. The Dyno2000 calculates these internal pressures at each degree of crank rotation throughout the four-cycle process. To determine mass flow into and out of the cylinders at any instant, the flow that occurs as a result of these changing pressure differences is also calculated. 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.

While it is impractical to use cylinder head flow data directly in an engine simulation, measured cylinder head flow figures are, nonetheless, a good starting point. 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 predicted mass flow moving into and out of the cylinders. Furthermore, the discharge coefficient also can be used to predict the changes in flow for larger or smaller valves and for various levels of port modifications. In other words, the discharge coefficient provides a practical method to simulate mass flow within a large range of engines under a wide range of operational conditions.

Sorting Out Cylinder Head Menu Choices

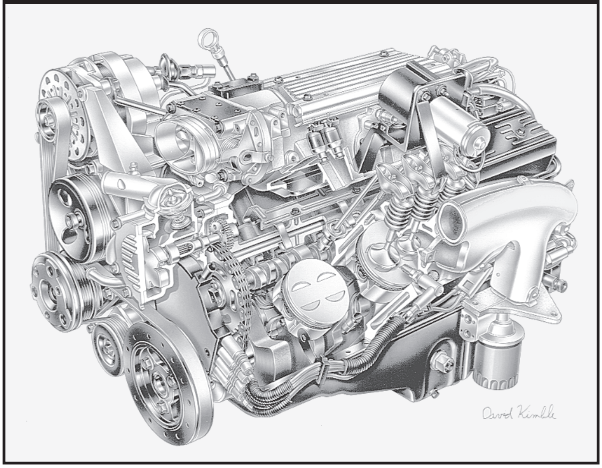
Now that some of the basic flow theory behind the choices in the CYLINDER HEAD category menus has been exposed, here's some practical advice that may

Typical Low-Performance Cylinder Heads

The "Low Performance" cylinder head choices are intended to model cylinder heads 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 early smallblock Chevy castings fall into this category.

Cylinder Head Menu

Typical Wedge Cylinder Heads



The "Wedge Cylinder head" menu choices model cylinder heads that have ports and valves sized with performance in mind, like the heads on this LT1 smallblock Chevy.

help you determine the appropriate selections for your application.

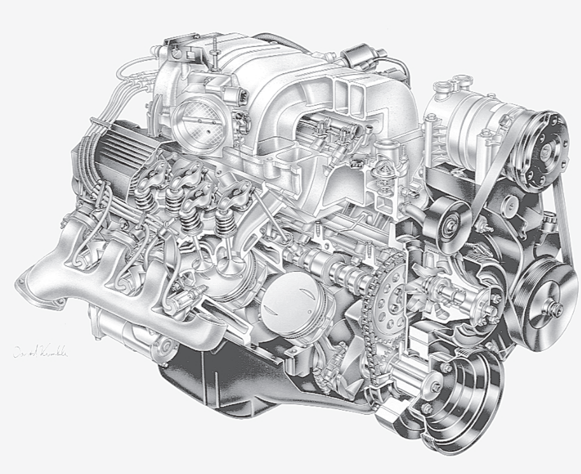
Low Performance Cylinder Heads—There are three "Low Performance" cylinder head selections listed at the top of the Cylinder Head menu. Each of these choices is intended to model cylinder heads that have unusually small ports and valves relative to engine displacement. Heads of this type were often designed for low-speed, economy applications, with little concern for high-speed performance. Early 260 and 289 smallblock Ford and to a lessor degree 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, 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 "Low Performance/Pocket Porting" choice 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 plus slightly larger intake and exhaust valves and some modest work in the port runners. Auto-calculate valve size increases vary, but they are always scaled to a size that will 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 even more restrictive, by selecting Low-Performance and manually entering the exact valve sizes, the simulation will, at least, give you an approximate power output usable to evaluate changes in cam timing, induction flow, and other components.

Cylinder Head Menu

Typical Canted-Valve Cylinder Heads



The "Canted-Valve Cylinder Head" selections have ports with generous cross-sectional areas and valves that angle toward the port mouths. The first three menu choices model <u>oval-port</u> designs. The final two selections simulate performance <u>rectangular-port</u> heads. This L29 big-block Chevy would be best modeled by the second or third menu choice—the fourth menu choice models a head with flow capacity beyond the capabilities of L29 castings.

Wedge Cylinder Heads—The wedge-chamber and canted-valve choices comprise the two main cylinder head categories. Choices from these two groups are applicable to 90% of all performance engine applications.

The first three basic wedge 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 best for high-performance street to modest racing applications.

The fourth wedge head "Wedge/Fully Ported, Large Valves" 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 that has additional modifications to provide optimum flow for racing applications. It does not incorporate "exotic" modifications, like raised and/or welded ports that require custom-fabricated manifolds.

The last choice in the wedge group is "Wedge/Pro-Stock Porting And Mods." This selection is designed to model state-of-the-art, high-dollar, Pro-Stock drag-racing cylinder heads. These custom pieces are designed for one thing: Maximum power. They usually 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.

Canted-Valve Cylinder Heads—All canted-valve selections are modeled after heads with "canted" valves. That is, the valve stems are tilted toward the outside of the cylinder heads to improve the discharge coefficient and overall airflow. All ports have generous cross-sectional areas for excellent high-speed performance.

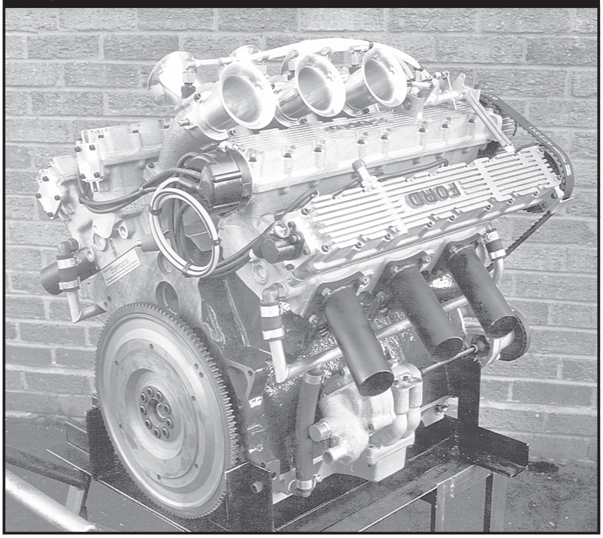
The first three choices are based on an oval-port configuration. These smaller cross-sectional area ports provide a good compromise between low restriction and high flow velocity for larger displacement engines. The stock and pocket-ported choices are suitable for high-performance street to modest racing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

The final two selections simulate extensively modified rectangular-port heads. These choices model, primarily, all-out, big-block heads, however, they closely model other extremely aggressive high-performance racing designs, like the Chrysler Hemi head and all-out ProStock designs. As with the smallblock category, the “Canted/Rectangular Ports/Fully Ported” 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 canted-valve group is “Canted/Rectangular ProStock Ports/Mods.” This selection is designed to model state-of-the-art, ProStock drag-racing cylinder heads. These custom pieces, like their wedge-design counterparts, are built from the ground-up 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 Cylinder Head menu, except for 4-valve heads (discussed next). These specially fabricated cylinder heads only develop sufficient airflow for good cylinder filling with large displacement engines at very high engine speeds.

4-Valve Cylinder Heads—The next three selections in the Cylinder Head submenu

Typical 4-Valve Cylinder Heads



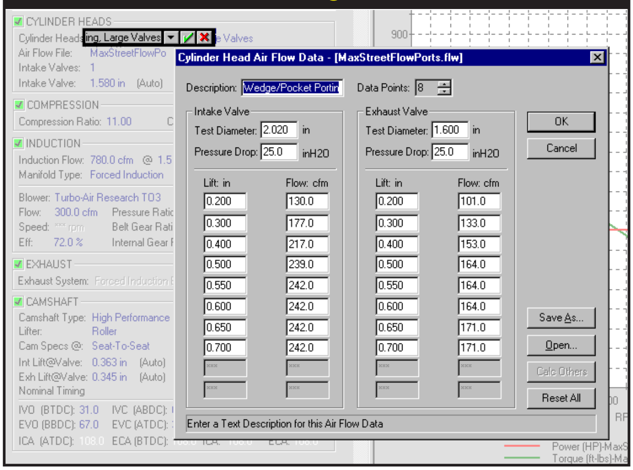
The “4-Valve Cylinder Head” selections model cylinder heads with 4-valves per cylinder. These heads can offer more than 1.5 times the curtain area of the largest 2-valve heads. This large valve area, combined with high-flow, low-restriction ports greatly improves air and fuel flow into the cylinders 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 V8s.

model 4-valve cylinder heads. These are very interesting choices since they simulate the effects of very low-restriction ports and valves used in many import 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 low valve lifts and 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, unless camshaft timing is carefully designed to complement the low-lift flow capabilities of these cylinder heads. On the other hand, the outstanding flow characteristics of the 4-valve head put it in another "league" when it comes to horsepower potential on high-speed racing engines.

The first choice in the 4-valve group is "4-Valve Head/Stock Ports And Valves." This simulates a 4-valve cylinder head that would be "standard equipment" on factory high-performance or "sports-car" engines. These 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" incorporates mild performance modifications. Larger valves have been installed and both intake and

Custom Port Flow Dialog



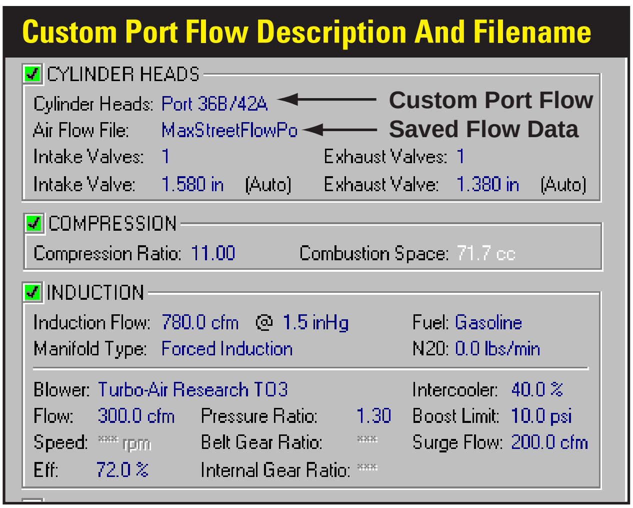
The Custom Port Flow dialog box allows the direct entry of flow bench data. From 4 to 10 data points for each port can be entered. Virtually any test valve diameter and pressure drop can be used.

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 most engines.

The final choice, “4-Valve Head/Race Porting And Mods,” like the other “Race Porting And Mod” choices in the Cylinder Head menu, models an all-out racing cylinder head. 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 with late IVC timing on low-speed, small-displacement engines.
Note: These heads, like all choices provided in the Cylinder Head menu, 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.
Tip: If you would like to know what “hidden” power is possible using any particular engine combination, try this cylinder head choice. It is safe to say that the only way to find more power, with everything else being equal, would be to add forced induction, nitrous-oxide injection, or use exotic fuels.

Custom Port Flow—The Dyno2000 will accept flowbench data, taken from measuring virtually any port, with any valve size, at any pressure drop. Selecting **Custom Port Flow** opens the airflow-bench dialog box (see photo on previous page). If you open this dialog after you have selected one of the other cylinder head menu choices, the Custom Port Flow dialog will display the flow data for that head configuration.

To enter flow-bench data, first provide a short description of the flow-bench/cylinder head test in the **Description** field. Then select the number of data points



When Custom Port Flow is used, the port Description name (entered in the Port-Flow Dialog Box) is displayed in the CYLINDER HEAD category. In addition, if the flow data was saved to disk, the filename is also displayed. You can double-click on the filename (or asteristics in that field) and load and save airflow data.

in your flowbench test into the **Data Points** field (click up to increase, down to decrease). Then enter the test-valve diameters and the pressure drop (in inches of H_20) at which the tests were performed. Finally enter flow and valve-lift test data. **Note 1:** You may press the **Calc Others** button at any time to fill in the remaining lift fields with the same "step" value that was established in the previous fields. The **Calc Others** button is smart enough to change step values at higher valve lifts. **Note 2:** If you have fewer data points for one of the valves, simply repeat the highest measured flow value to "flush out" the remaining data points. This technique has been shown to produce accurate simulations.

You can save the flow data to your hard drive at any time by pressing the **Save As** button. Recall previously saved flow data with the **Open** button.

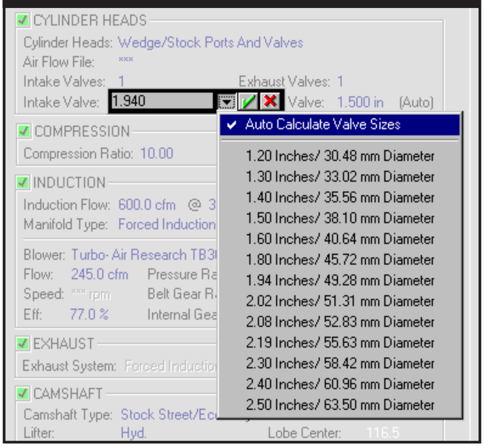
Pressing **OK** will load the new test data into the engine database and display the custom flow **Description** in the CYLINDER HEAD category.

Valve Diameters

The **Valve Diameter** menus are located in the lower portion of the CYLINDER HEAD category. The first selection is **Auto Calculate Valve Size**. This 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 current bore diameter and the Cylinder Head selection. When the Auto Calculate function is activated **Auto** will be displayed next to the calculated sizes, and it remains active on the current engine until turned off (by selecting it a second time). Auto Calculation is turned off by default when the Dyno2000 is started and whenever ***Clear Components*** is chosen from the **Edit** menu.

Auto Calculate Valve Size is especially helpful if you are experimenting with several different bore and stroke combinations or comparing different engine configurations. Auto Calculate will always select valves of appropriate diameter for the

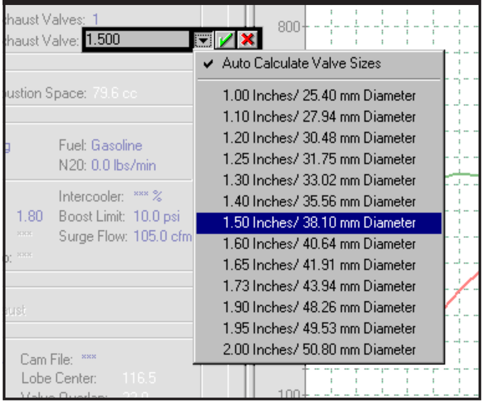
Intake And Exhaust Valve Sizes



Select valve sizes for the intake and exhaust valves from drop-down menus. If you choose Auto Calculate Valve Size **from either the intake or the exhaust menu, the Dyno2000 will size all valves appropriately, based on the cylinderhead type and the bore diameter. Deselecting** Auto Calculate Valve Size **on either the intake or the exhaust valve-size menus will disable this feature on all valves.**

Selecting a specific valve size will disable Auto Calculate Valve Size**. You can select from the provided sizes (displayed in both US and Metric measurements), or you can directly enter any valve dimension within the range limits of the Dyno2000.**

Manually Selecting Valve Sizes



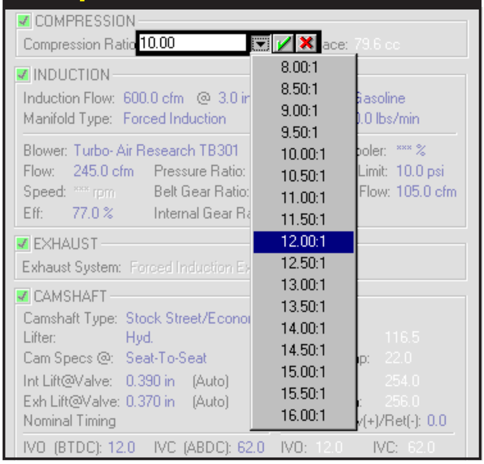
cylinder heads under test and it will never use valve sizes that are too large for the current bore diameter (also, see page 69 for information on the related **Auto Calculate Valve <u>Lift</u>** feature).

While the **Auto Calculate Valve Size** 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 situations refer to the lower choices on the Valve Diameter menus. Here you will find a list of exact valve sizes consisting of common intake and exhaust dimensions. In addition, you can directly enter any valve diameters within the acceptable range limits of the program.

THE COMPRESSION RATIO MENU

The **Compression Ratio** menu is located in the COMPRESSION category. A

Compression Ratio Menu



The Compression Ratio of the engine 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). Passenger car engines often have 8 to 10:1. While racing engines can have as high as 18:1 compression ratio.

Compression Ratio Menu

Basic Compression Ratio Equation

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

Compression ratio is calculated by dividing the total volume that exists in the cylinder when the piston is located at BDC by the volume that exists when the piston is positioned at Top Dead Center.

selection from this menu establishes the compression ratio for the simulated engine (the Dyno2000 range of compression ratios is 6:1 to 18:1). As mentioned earlier, 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).

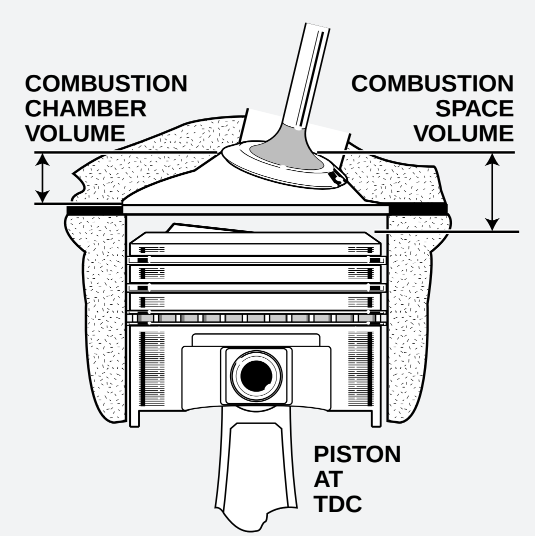
Let’s take a close look at this relationship to discover exactly what compression ratio is and how compression ratio affects performance in the IC engine.

Compression Ratio Basics

The 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 volumes, so the first step in exploring compression ratio must be to understand these volumes.

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 and the head-gasket thickness, less any volume due to the piston or piston dome protruding above the top of the bore.

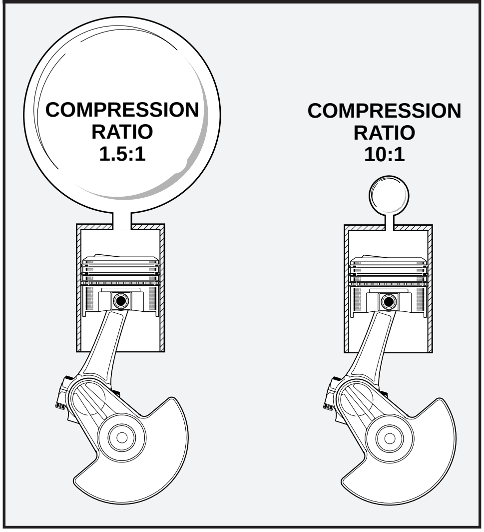
Top Dead Center Volumes



Compression Ratio Menu

A combustion space containing twice as much volume as the cylinder produces a 1.5:1 compression ratio. Peak cylinder pressures after fuel ignition 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 reach about 1500psi. The higher compression ratio generated much higher cylinder pressures throughout the first half of the piston's travel from TDC to 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.

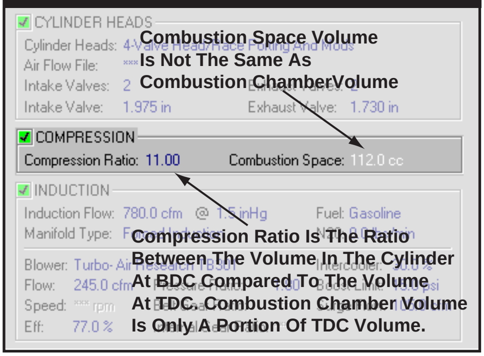
Compression Increases Power



Swept cylinder volume is the most straightforward of the two. As you discovered previously, swept volume is calculated by the Dyno2000—and displayed in the SHORTBLOCK category—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 positioned at TDC. This space includes the volume in the combustion chamber, the volume taken up by the thickness of the head gasket, 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. However, the following explanation and illus-

Combustion Space Volume

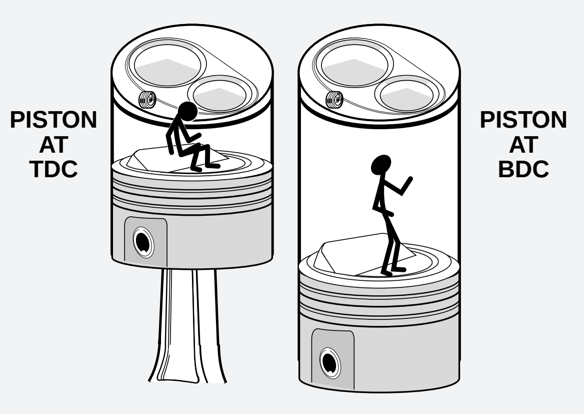


An 11:1 compression ratio (as shown here) means that the sum of the Cylinder Volume and the Combustion Space Volume is eleven times greater than the volume in the Combustion Space alone.

Compression Ratio Menu

A good way to visualize compression ratio volumes is to imagine yourself a "little man" wandering around inside the engine. You would see the combustion chamber above you like a ceiling. Your floor would be the top of the piston (see text for additional description of cylinder volumes).

Compression Space And Volumes



trations should clarify these important engine variables.

A good way to visualize these volumes 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 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 trim 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 room divider 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 room divider (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 combustion space, 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 described in the numerator of the compression-ratio 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 decrease 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" 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 rapidly. This is due, in part, to the increase in combustion volume that often accompanies a larger bore (again, using our "little man," a larger bore adds more floor space by increasing the diameter of the room—and it can also increases the size of the ceiling), partially offsetting the compression-ratio increase from swept cylinder volume.

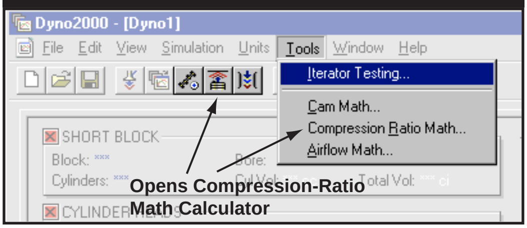
Changing combustion space, the other element in the equation, will also alter the compression ratio. Anything that reduces the combustion volume, while maintaining or increasing the swept volume of the cylinder, will increase the compression ratio. Some of the more common methods are decreasing the volume 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. These modifications and others can be explored further using the built-in Compression Ratio Math Calculator, described in the next section.

THE COMPRESSION-RATIO MATH CALCULATOR

The Dyno2000 advanced engine simulation allows the selection and testing of virtually any compression ratio. But many users have requested the ability to directly enter combustion-chamber volumes, head-gasket thickness, etc., to determine their effects on compression ratio. The **Compression-Ratio Math Calculator** built-in to the Dyno2000 quickly performs these functions. But it is not another "enter-the-numbers-into-the-equation" calculator. This tool "intelligently" adjusts itself to the needs of the engine builder, changing the way it handles volumes for flattop or domed piston configurations.

After you have specified the bore, stroke, and number of cylinders for the engine under test, activate the **Compression-Ratio Math Calculator** by selecting either ***Compression-Ratio Math*** from the **Tools** menu or by clicking on the **Compression-Ratio Math Icon** in the Toolbar. When the calculator is first activated, it defaults to

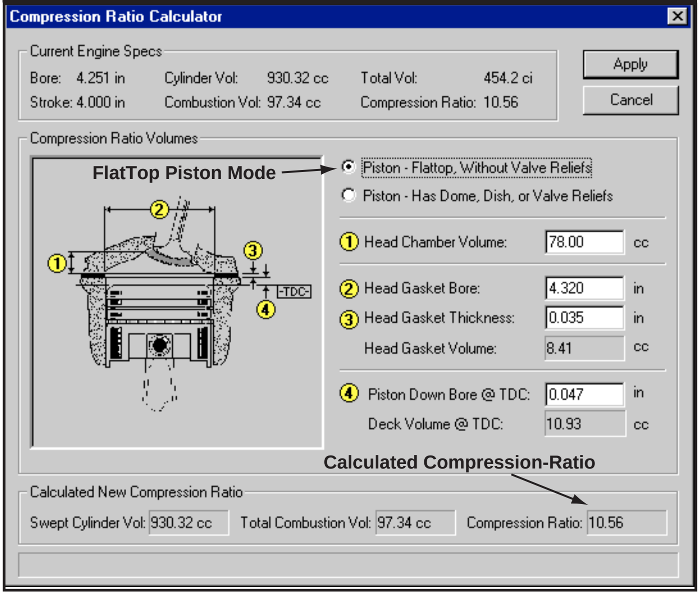
Compression-Ratio Math Calculator



After you have specified the bore, stroke, and number of cylinders in the engine, activate the Compression-Ratio Math Calculator by either selecting *Compression Ratio Math* from the Tools menu or by clicking on the Compression-Ratio Math Icon in the toolbar.

Compression-Ratio Math Calculator

CR Math Calculator—FlatTop Piston Mode



When the Compression-Ratio Math Calculator is first activated, it defaults to the Flattop Piston Mode. This is the simplest model for calculating compression ratio, since the combustion-volume can be calculated by the simple sum of the chamber volume, head-gasket volume, and deck volume (*deck volume* is the remaining space in the cylinder with the piston at TDC).

flattop-piston mode. This is the simplest model for calculating compression ratio, since the combustion volume (the space above piston at TDC) can be calculated by the simple sum of the chamber volume, head-gasket volume, and deck volume (volume remaining in the cylinder with the piston at TDC).

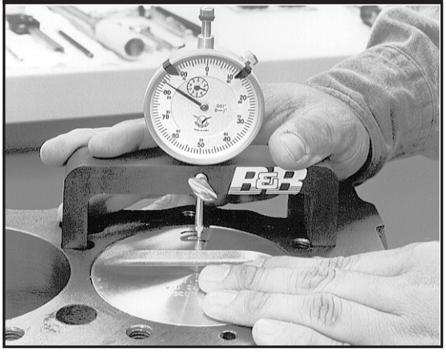
Using The Calculator With FlatTop Pistons Without Valve Reliefs

Flattop pistons do not require measuring or calculating the volume of any domes, dishes, or valve reliefs, so calculating the final compression ratio is considerably simplified. Here is the procedure to use the **Flattop-Piston Mode**; the domed-piston model will be discussed in the next section. Start off by verifying that the calculator is in the Flattop Mode by checking the upper radio button *Piston—Flattop, Without Valve Reliefs*. Next, enter the combustion-chamber volume (in cubic centimeters—cc's) in the first **(1)** *Head Chamber Volume* data box (refer to the above photo). Combustion chamber volume is typically measured with a burette containing a colored liquid (for more information on compression ratio basics, refer to page 31).

The next two data-entry boxes are used to calculate the volume added to the combustion space by the compressed head gasket. The data box marked **(2)** accepts the *Head Gasket Bore* diameter in the appropriate (Metric or US) units system. Most head gaskets have a bore-circle larger than the cylinder-bore diameter. For gaskets with bore-circle diameters of odd shapes, simply estimate the bore size by averaging the larger and smaller dimensions. Next, enter the head gasket compressed thickness in the **(3)** *Head Gasket Thickness* field. This dimension is often

Compression-Ratio Math Calculator

Measuring Deck Height



Use a dial indicator and stand to measure how far down the bore the piston is positioned at TDC. Enter a positive number for "down-the-bore" distances and a negative number if the piston protrudes above the deck surface. A typical value might be +0.040, indicating that the piston comes to a rest at TDC 0.040-inch <u>below the deck surface</u>.

available from the gasket manufacturer. When the thickness is entered, the *Head Gasket Volume* is calculated.

For flattop pistons, the next data entry field is **(4)** *Piston Down Bore @ TDC* that allows you to enter how far down the bore the piston is positioned at TDC. Enter a positive number for "down-the-bore" distances and a negative number if the piston protrudes above the deck surface (see photo, above). A typical value might be +0.040, indicating that the piston comes to a rest 0.040-inch <u>below the deck surface</u> at TDC. As soon as this last value is entered, both the *Deck Volume @ TDC* and the *Compression Ratio* are calculated.

Note: A positive *Deck Volume @ TDC* indicates the piston is positioned below the deck surface and this volume adds to the combustion space at TDC; a negative number indicates the piston protrudes above the deck surface at TDC and subtracts from the combustion space.

At this point, you can move to any of the previous fields (by clicking in them or using the Tab and/or the SHIFT-Tab keys) and modify any values to determine their effect on compression ratio. At any time, you can click on the **Apply** button to load the new calculated compression ratio into the Component Screen and save all entered values with the simulated engine. Alternately, you can press the **Cancel** button to discard all entries and leave any previously entered compression ratio specifications intact.

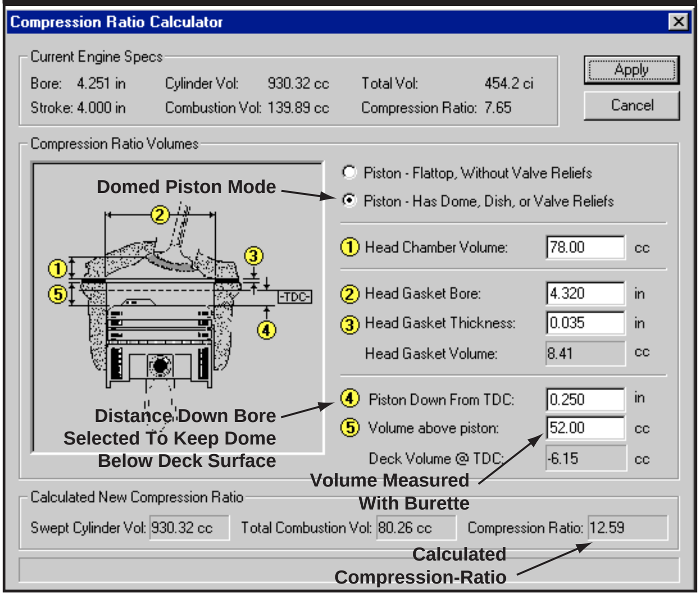
Using The Calculator With Domed/Dished Pistons Or Pistons With Valve Reliefs

Pistons with domes, dishes, pockets, or valve reliefs complicate the compression ratio issue. Each of these volumes must be accurately determined so that the net effect of all "positive" (domes) and "negative" (pockets, reliefs) can be calculated.

Start off by verifying that the calculator is in the Domed Mode by checking the lower radio button *Piston—Has Dome, Dish or Valve Reliefs*. Enter the combustion chamber volume (in cubic centimeters—cc's) in the first **(1)** *Head Chamber Volume* data box. As described earlier, combustion chamber volume is typically measured

Compression-Ratio Math Calculator

CR Math Calculator—Domed Piston Mode



When the Compression-Ratio Math Calculator is switched to the Domed Piston Mode, field (4) is redefined and an additional field (5) is displayed. These fields allow the engine builder to calculate a volume (Deck Volume @ TDC) equivalent to the sum of all the dome, dish, and relief volumes of the piston. The piston is lowered down the bore until the dome is located below the deck surface (4). A direct measurement is taken of the cylinder volume (5). The Deck Volume @ TDC is then calculated and displayed.

with a burette.

The next two data-entry boxes allow the program to calculate the volume added to the combustion space by the compressed head gasket. The data box marked **(2)** accepts the *Head Gasket Bore* diameter in the appropriate (Metric or US) units system. Most head gaskets have a bore-circle larger than the cylinder-bore diameter. For gaskets with bore-circle diameters of odd shapes, simply estimate the bore size by averaging the larger and smaller dimensions. Next, enter the head gasket compressed thickness in the **(3)** *Head Gasket Thickness* field. This dimension is often available from the gasket manufacturer. When the thickness is entered, the *Head Gasket Volume* is calculated. Up to this point, the function of the Compression Math Calculator is identical to the flattop-piston mode, however, the next two data entry fields are unique to the domed-piston model.

The next entry **(4)** *Piston Down From TDC* allows you to enter a distance down the bore (measured from the deck surface) that positions the highest part of the piston dome below the deck. Typical values may be 0.100-inches or 0.250-inches depending on the height of the piston dome (any distance is acceptable as long as the entire dome resides below the deck surface). At this depth, a direct measurement is made of the *Volume Above The Piston* in the cylinder. This measurement is taken by the engine builder (see photo on next page) using a burette with a petcock to fill the space above the piston (grease is often used to “seal” the piston to the bore and a flat Plexiglas plate covers and seals the top of the bore). The volume liquid dispensed will be less (typically) than the volume for a simple cylinder of the same height as the distance the piston is positioned below the deck surface.

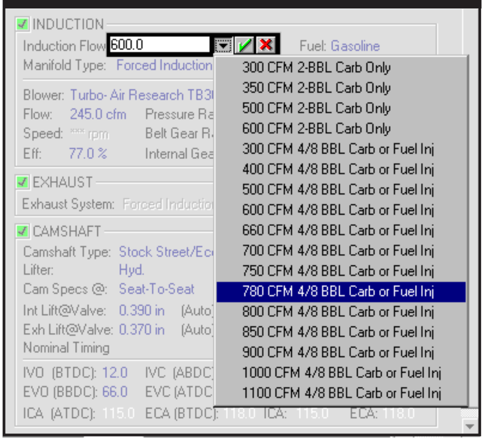
Induction Airflow Menus

Measuring Dome/Deck Volume

Measure the volume above the piston while the highest portion of the piston dome is positioned below the deck surface. Enter this value in field (5) *Volume Above Piston*. The difference between this volume and the volume of a simple cylinder [of a height equal to the value entered in field (4)] is the *Deck Volume At TDC*. This volume is equivalent to the sum of all the dome, dish, and relief volumes of the piston. A negative *Deck Volume At TDC* indicates that the dome reduces the combustion space and will increase the compression ratio over a flattop piston. A positive value indicates that the sum of all dome/deck/dish/relief volumes will increase the combustion volume and decrease the compression ratio over a flattop piston.

The liquid volume dispensed from the burette is entered in field **(5)** *Volume Above Piston*. The difference between this volume and the volume of a simple cylinder [of a height equal to the value entered in field **(4)**] is the *Deck Volume At TDC*, a volume equivalent to the sum of the dome, dish, and relief volumes of the piston.
Note: A negative *Deck Volume At TDC* indicates that the dome reduces the combustion space and will increase the compression ratio over a flattop piston. A positive value indicates that the sum of all dome/deck/relief/dish volumes will increase the combustion space volume and decrease the compression ratio over a similar flattop piston (with the same deck height at TDC).

Induction Airflow Menu



The Induction Airflow menu establishes the airflow restriction for the induction system and the pressure drop at which the airflow is measured. For the purposes of the simulation, everything upstream of the intake ports, including the intake manifold, carburetor/fuel-injection system, venturis, any supercharger or turbocharger, and the openings to the atmosphere is considered the induction system. The Airflow menu consists of four 2-barrel-carburetor selections and thirteen 4-barrel-carburetor/fuel injection choices. In addition, you can directly specify the rated airflow from 100 to 3000cfm and the pressure drop at which this airflow is measured.

Induction Airflow Menus

THE INDUCTION MENU

The next main component category establishes an INDUCTION system for the simulated test engine. An induction system, as used in the Dyno2000, is everything upstream of the intake ports, including the intake manifold, common plenums (if used), carburetor/fuel-injection throttle body, venturis (if used), any supercharger or turbocharger, and openings to the atmosphere. Dyno2000 induction menus are divided into two groups: 1) Airflow, pressure drop, fuel type, and manifold type, and 2) forced induction.

Airflow Selection And Pressure Drop

The first two Induction menus are used to select the rated airflow for the induction system and the pressure drop at which this airflow is measured. The **Induction Flow** menu consists of four 2-barrel-carburetor selections and thirteen 4-barrel-carburetor/fuel-injection choices. In addition, you can directly specify any rated airflow from 100 to 3000cfm.

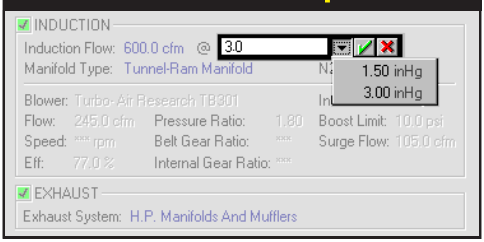
The two-barrel selections “install” either a 300-, 350-, 500-, or 600-cfm 2-bbl carburetor on the test engine. These are the only 2-barrel choices directly available in the menu. The remaining **Induction Flow** choices range from 300 to 1100cfm, 4- or 8-barrel carburetors and fuel-injection applications.

The flow ratings for 2-barrel carburetors are measured at a pressure drop twice as high as the pressure used to rate 4-barrel carburetors and most fuel-injection systems. Rated airflow for 2-barrels is typically measured at a pressure drop of 3 inches of mercury, while the pressure drop for 4-barrel carburetors is 1.5-inches of mercury (this is the pressure differential maintained across the carburetor during airflow measurement at wide-open throttle). This is displayed as **3-inHg** in the **Pressure Drop** menu (**Hg** is the symbol for mercury as used in the periodic table of elements).

Note: 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 pressure-drop convention, it is possible to simulate virtually any 2-barrel or 4-barrel induction. By manually entering, say, 460cfm into the **Induction Flow** menu and 3.0 for **Pressure Drop**, the program will accurately model a 460cfm

Induction Pressure-Drop Menu



Use the Induction Airflow Pressure Drop menu to select between 1.5-inches of Mercury, a measurement standard for 4-barrel carburetors and injection systems, and the two-barrel carburetor standard of 3.0-inches of Mercury.

2-barrel.

Note: See the Airflow Math Calculator (see page 40) for quick conversions between any airflow measured at any pressure drop.

The last thirteen choices in the **Induction Flow** menu are labeled with **4/8-BBL Carb Or Fuel Inj**. These selections designate airflow ratings that were measured at 1.5-in/Hg. **4/8-BBL** indicates that the induction system can consist of single or multiple carburetors or a fuel-injection system capable of the rated airflow. The important thing to remember about airflow selection is that the program *makes no assumption about the type of restriction* used in the carburetor or injection system. The airflow is simply a rating of the total restriction of the induction system.

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. To keep things consistent, 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 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 the *total rated airflow into the engine*. On dual-inlet or multiple-carburetor systems, the Induction 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).

Note: Keep in mind the unique way airflow capacities are handled on Individual Runner (I.R.) manifolds (discussed in an upcoming section). 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, enter 2000cfm directly into the Induction Airflow field. When an **I.R.** manifold is selected from the **Manifold Type** menu, the airflow is equally divided into all cylinders.

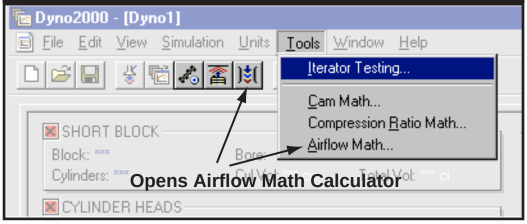
THE AIRFLOW MATH CALCULATOR

The Dyno2000 will simulate virtually any engine with an induction airflow rating measured at a pressure drop of either 1.5-Inches of mercury (In/Hg), widely accepted as the standard 4-barrel airflow pressure-drop rating system, or at 3.0-In/Hg,

Airflow Math Calculator

The Airflow Math Calculator is a general-purpose tool that will convert airflow to/from any pressure-drop standard. Activate the Airflow Math Calculator by either selecting *Airflow Math* from the Tools menu or by clicking on the Airflow Math Calculator Icon in the toolbar.

Airflow Math Calculator



the standard pressure drop used to rate 2-barrel carburetors. For those instances where an induction system, injector, or carburetor was flow tested at a different pressure drop, or whenever you would like to convert flow values from one pressure-drop to another, the Dyno2000 **Airflow Math Calculator** easily performs these conversion functions. The Airflow Math Calculator can also convert flow ratings measured in inches-of-mercury (in/Hg) to and from airflow values rated in inches-of-water (in/H_2O).

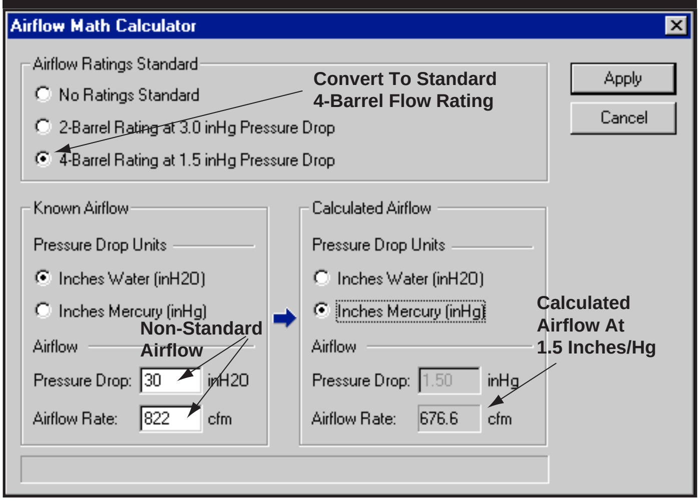
Note: A pressure drop of 1.5-in/Hg is equivalent to 20.3-in/H_2O.

The **Airflow Math Calculator** has three basic modes of operation: 1) Convert to the 4-Barrel Standard, 2) convert to the 2-Barrel Standard, and 3) calculate airflow between any pressure drop ratings. Each of these methods are described below. Activate the Airflow Math Calculator by either selecting *Airflow Math* from the **Tools** drop-down menu or click on the **Airflow Icon** located in the Toolbar.

Using The Airflow Math Calculator
Mode 1: Convert To The 1.5-in/Hg, 4-Barrel Standard.

When the calculator is first activated, the *Airflow Ratings Standard* "radio button"

Airflow Math Calc—Convert To 4-Bbl Standard



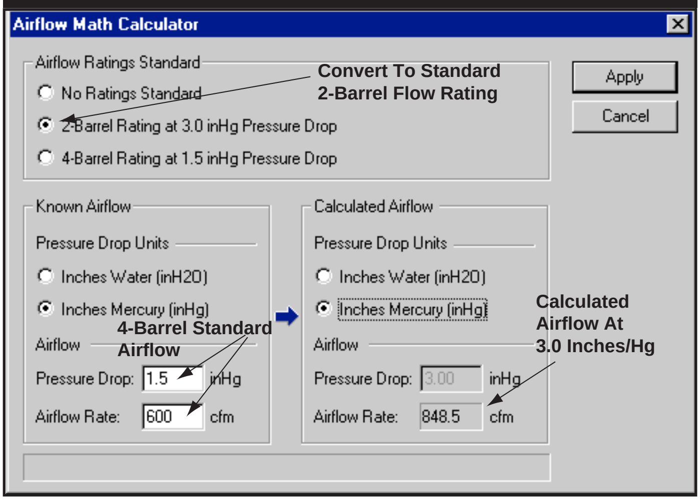
When the calculator is first activated, the *Airflow Ratings Standard* is set to *1.5-in/Hg*. This forces the *Calculated Airflow* to default to a pressure drop of 1.5-in/Hg or 20.3-in/H_2O. To convert any known airflow to the 1.5-in/Hg, 4-barrel standard, enter the measured airflow and pressure drop in the *Known Airflow* category. The calculated airflow will be displayed in the *Airflow Rate* field.

of 1.5-in/Hg is selected. This forces the result, or *Calculated Airflow* category to default to a pressure drop of 1.5-in/Hg or 20.3-in/H_2O (these pressure drops are identical). To convert any known airflow measured at any pressure drop to the 1.5-in/Hg, 4-barrel standard, enter the measured airflow and pressure drop in the *Known Airflow* category (you can switch between Inches-of-Mercury(Hg) and Inches-of-Water (H_2O) buy clicking on the appropriate radio buttons in the *Known Airflow* and *Calculated Airflow* categories). The calculated airflow will be displayed in the *Airflow Rate* field (see photo, previous page). You can move to any of the previous fields (by clicking on them or using the Tab or SHIFT-Tab keys) to make changes and determine their effect on the calculated airflow. At any time, you can click on the **Apply** button to load the new calculated airflow into the Induction Airflow field on the Component Selection screen, saving all entered values. Alternately, you can press the **Cancel** button to discard all entries and leave any previously entered values intact.

Using The Airflow Math Calculator
Mode 2: Convert To The 3.0-in/Hg, 2-Barrel Standard.

Switch the *Airflow Ratings Standard* category selection to the radio button marked *2-Barrel Rating of 3.0-in/Hg Pressure Drop*. This forces the "result," or *Calculated Airflow* category to default to a pressure drop of 3.0-in/Hg or 40.7-in/H_2O (these pressure drops are identical). To convert any known airflow measured at any pressure drop to the 3.0-in/Hg, 2-barrel standard, enter the measured airflow and pressure drop in the *Known Airflow* category (you can switch between Inches-of-Hg and Inches-of-H_2O buy clicking on the appropriate radio buttons in the *Known Airflow* and *Calculated Airflow* categories). The new, equivalent airflow at the new 3.0-in/Hg pressure drop will be displayed in the *Airflow Rate* field (see photo, below). You can

Airflow Math Calc—Convert To 2-Bbl Standard

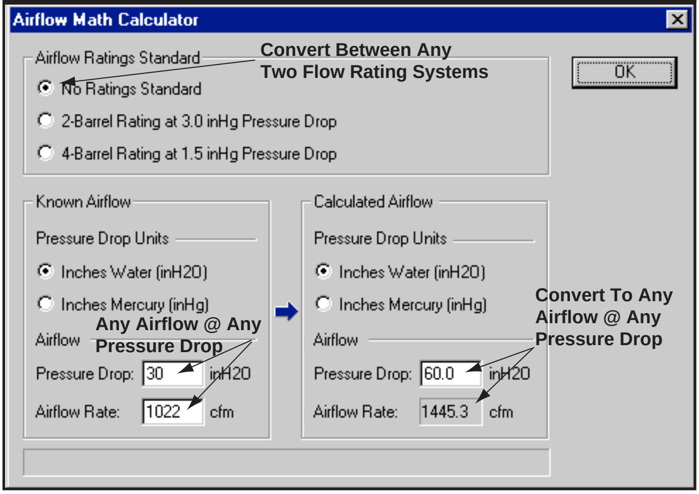


Switch the *Airflow Ratings Standard* to *3.0-in/Hg*. This forces the *Calculated Airflow* to default to a pressure drop of 3.0-in/Hg or 40.7-in/H_2O (a pressure drop used for 2-barrel carburetors). Enter the measured airflow and pressure drop in the *Known Airflow* category (a 4-barrel airflow is shown here, but any airflow at any pressure drop may be entered). The new calculated airflow is displayed in the *Airflow Rate* field.

Airflow Math Calculator

Switch the ***Airflow Ratings Standard*** to *No Ratings Standard*. The ***Calculated Airflow*** can now be set to any pressure drop measured in Inches of Hg or H_2O. Select the desired Pressure Drop Units and enter the known airflow and pressure drop. Enter the desired pressure drop in the ***Calculated Airflow*** category. The equivalent airflow will be displayed in the ***Airflow Rate*** field.

Airflow Math Calc—Convert To Any Pressure Drop



move to any of the previous fields (by clicking on them or using the Tab or SHIFT-Tab keys) make changes and determine their effects on the calculated airflow. At any time, you can click on the **Apply** button to load the new calculated airflow into the Induction Flow field on the Component Selection screen, saving all entered values. Alternately, you can press the **Cancel** button to discard all entries and leave any previously entered values intact.

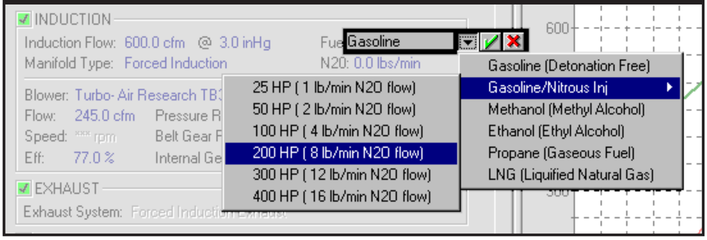
Using The Airflow Math Calculator
Mode 3: Convert To Equivalent Flow At Any Pressure-Drop.

Note: Since the Dyno2000 Induction Flow field only accepts induction airflow rated at either 1.5- or 3.0-in/Hg (20.3- or 40.7-in/H_2O), the **Apply** button is not shown when the No Ratings Standard is selected. If you wish to use the new calculated values in a dyno test, select either the *4-Barrel Rating at 1.5-in/Hg Pressure Drop* or *2-Barrel Rating at 3.0-in/Hg Pressure Drop* choices in the *Airflow Ratings Standard* category.

Switch the *Airflow Ratings Standard* category selection to the radio button marked *No Ratings Standard*. This allows the “result,” or *Calculated Airflow* category to be set to any pressure drop measured in Inches of Hg or Inches of H_2O. To convert any known airflow measured at any pressure drop to any other pressure drop equivalent flow, enter the starting airflow and pressure drop in the *Known Airflow* category (you can switch between Inches-of-Mercury(Hg) and Inches-of-H_2O). Then enter the new pressure drop in the *Calculated Flow* category. The calculated equivalent airflow will be displayed in the *Airflow Rate* field (see photo, above). You can move to any of the previous fields (by clicking on them or using the Tab or SHIFT-Tab keys) to make changes and determine their effects on the calculated airflow.

Fuel Menu

Fuel And Nitrous-Oxide Selection Menu



The Dyno2000 allows a wide range of possible fuels for dyno testing. When any of these fuels have been selected, the air/fuel ratio is adjusted to ensure optimum power.

FUEL MENU

The Dyno2000 can model five automotive fuels and Nitrous-Oxide injection during dyno testing.

To select any of the available fuels, make a choice from the FUEL menu:

- Gasoline (Detonation Free),
- Methanol (Methyl Alcohol),
- Propane (Gaseous fuel),
- Gasoline W/Nitrous Injection
- Ethanol (Ethyl Alcohol)
- LNG (Liquefied Natural Gas)

When any of these fuels have been selected, the Dyno2000 readjusts the air/fuel proportion for optimum power. Since combustion *flame-travel* is not modeled in the Dyno2000 (a flame-travel model would require a full 3D map of the combustion chamber and piston shape), detonation and/or variations in combustion efficiency are not calculated. However, the predicted power will accurately match dyno figures on engines that are setup properly to use these fuels.

Nitrous-Oxide Injection

There are many ways to boost engine power. However, nitrous-oxide injection is a uniquely effective method. Developed during World War II for piston-driven fighter aircraft, nitrous-oxide gas—an oxygen-releasing substance—allows an engine to burn more fuel while maintaining optimum air/fuel ratios. When injected into the cylinders with additional fuel, the effect is similar to instantaneous supercharging but without the losses from a belt- or exhaust-gas-driven device. Remarkable as it may seem, you can add about as much horsepower as you want, with the limitations being excessive cylinder pressure, detonation, and component failure. There are no subtleties here: Add more nitrous and fuel; get out more horsepower.

Most nitrous systems inject a fixed amount of nitrous and fuel, regardless of engine speed. In other words, when the nitrous “switch” is turned on, the engine will immediately produce a boost in power determined by the amount of injected fuel and nitrous. A nitrous injection system designed to add 100 horsepower (flowing about

4 pounds per minute of nitrous oxide), will produce a 100 horsepower boost instantly upon triggering the system, and continue to produce that horsepower increase across the entire rpm range. In other words, a 100hp nitrous system activated at 2000rpm (when the engine may have been producing only about 70hp) can virtually double or even triple power output.

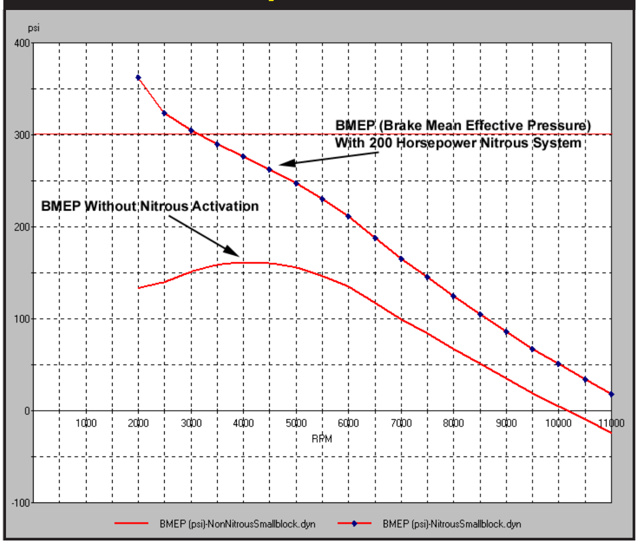
But these huge power boosts at low engine speeds (when each cylinder must ingest a large "dose" per power cycle) can send cylinder pressures through the roof. So, fixed-flow-rate systems are often designed to delay activation until the engine reaches sufficient speed to reduce each cylinder's nitrous load, thereby reducing cylinder pressures and preventing detonation and mechanical failure.

The Dyno2000 models a constant-flow nitrous/gasoline system. You should monitor cylinder pressures (BMEP) to make sure dangerously high pressures are avoided at lower engine speeds (a BMEP greater than 300psi is usually considered excessive). For example, the test graph shown above illustrates a 350 smallblock equipped with a 200hp nitrous system. Note that BMEP cylinder pressures below 3000rpm exceed 300psi.

One of the ways you can reduce low-speed cylinder pressures is by altering cam timing. Increasing valve duration and overlap will lower cylinder pressures at lower engine speeds. While this phenomenon is normally a hindrance to power at low speed, combined with a nitrous-oxide injection system it can permit earlier nitrous flow while optimizing power at higher rpms. Other variables that will decrease low-speed cylinder pressures are reduced compression ratios, increased exhaust-system back pressure, reduced induction airflow, less efficient induction manifolding, and larger engine displacement.

Using the Dyno2000 it is a simple matter to simulate and test a variety of com-

Nitrous-Oxide vs Cylinder Pressure (BMEP)



This graphic comparison was set up in the Dyno2000. It shows how cylinder pressure increases after a 200-horsepower nitrous system is activated. Since this is a fixed-flow nitrous system, cylinder loads of nitrous increase as engine speed decreases. Below 3000rpm, BMEP exceeds 300psi. To maintain engine reliability, nitrous-system activation should be delayed to keep cylinder pressures below this critical level.

ponent combinations to determine the maximum nitrous load that can be injected into any specific engine at any engine speed.

You can choose to add nitrous by selecting **Gasoline/Nitrous Injection** from the induction menu. This will open the following submenu:

- 25 HP (1 lb/min N_2Oflow),
- 50 HP (2 lb/min N_2Oflow)
- 100 HP (4 lb/min N_2Oflow),
- 200 HP (8 lb/min N_2Oflow)
- 300 HP (12 lb/min N_2Oflow),
- 400 HP (16 lb/min N_2Oflow)

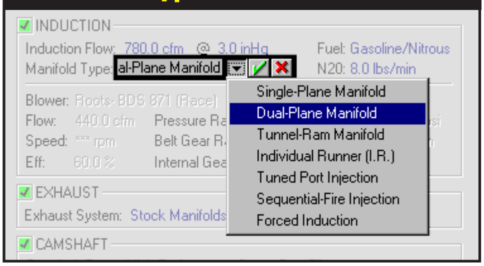
These six choices allow nitrous flow selections from 25hp (1-lb/min flow) to 400hp (16-lb/min flow). You can also directly enter flow values from 0.1- to 20-lb/min into the **Nitrous Flow Rate** field.

MANIFOLD TYPE MENU

The **Manifold Type** menu consists of six naturally-aspirated manifold choices and a Forced Induction selection (forced induction is discussed in the next section). Each of the six naturally-aspirated manifolds applies a unique tuning model to the induction system. These six manifolds are only a small sample of the comprehensive list of all the intake manifolds available for IC engines. The list should be interpreted as six discrete designs that, in fact, cover most manifolds available to the engine builder. If you are interested in a manifold 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 set up a comparison and 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 Dyno2000 (look into the Motion Software *Dynomation Lite™* engine simulation program series, available 2001), keep in mind that the data you obtain might not match real-world dyno data with some unique 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.

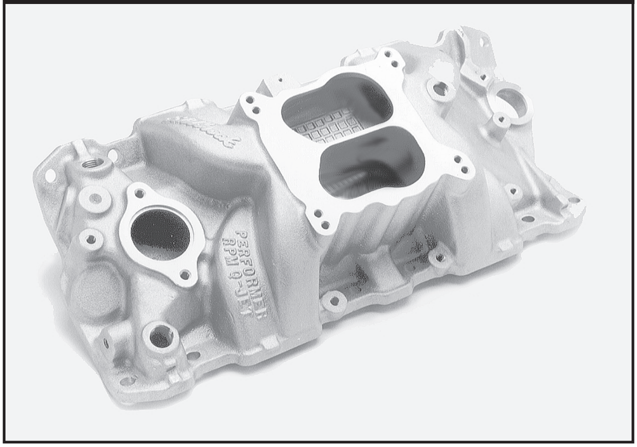
Manifold Type Menu



Each of the six naturally-aspirated manifolds in the Manifold Type menu applies a unique tuning model to the induction system. While these six manifolds are only a small sample of the intake manifolds available for IC engines, these six discrete designs cover most manifolds available to the engine builder.

Dual-Plane Manifold Modeling

Dual-Plane Manifold



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

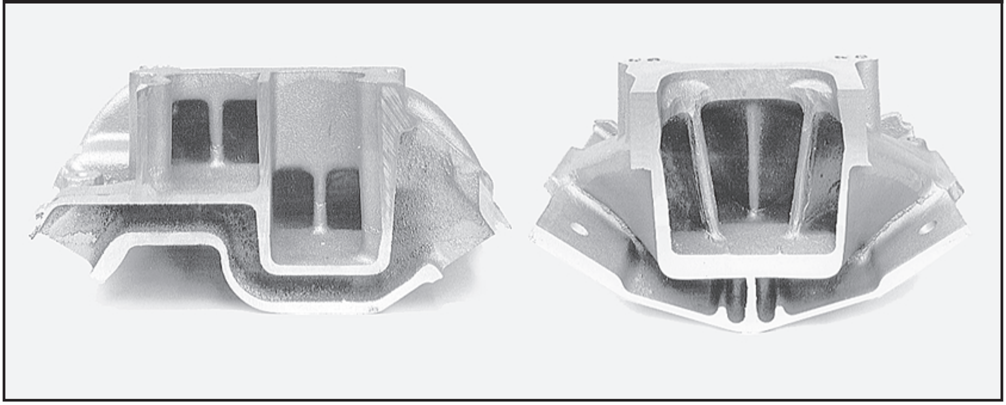
Dual-Plane Manifold—Remarkably, the well-known and apparently straightforward design of the dual-plane manifold is, arguably, the most complex manifold on the list. An intake manifold is considered to have a dual-plane configuration when 1) the intake runners can be divided into two groups, so that 2) 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 producing oscillations within the entire manifold. 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

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 only a slight sacrifice in low-speed torque. 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.

Hybrid Dual-Plane Design

Dual-Plane Manifold Modeling

Dual-Plane vs. Single-Plane Design



The basic differences between single- and dual-plane manifolds are clearly illustrated here. The dual-plane (left) divides the plenum in half, with the runners grouped 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 that is laid flat across the top of the engine. The manifold has excellent high-speed performance, but its design prevents full-manifold resonance. That reduces low-speed torque, driveability, and fuel economy.

known for its ability to boost low-end power.

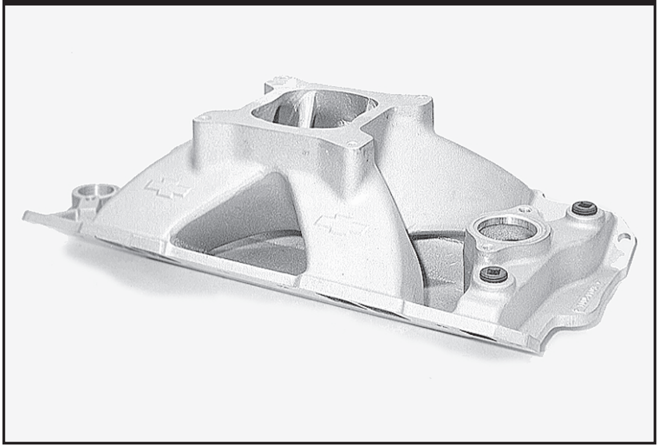
The divided plenum is another common feature of dual-plane manifolds that further boosts low-end power. Since each side of the plenum is connected to only one-half of the cylinders (4-cylinders in a V8), each cylinder in the engine is "exposed" to only one-half of the carburetor. This maximizes wave 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, limiting peak horsepower.

The main benefits of the dual-plane design 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 configuration. 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 these designs, you can successfully use the “trend” method described earlier to estimate engine torque and power. Unfortunately, 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 Dyno2000 simulation. Only a simulation that models intake passages, including the complex effects of multicylinder interference, can perform this analysis (Motion’s upcoming *Dynomation* engine simulation series is capable of this analysis). Unless you can perform actual dyno testing on these manifolds to determine 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. The runner design prevents full-manifold resonance. That reduces low-speed torque, and depending on the size of the plenum and runners, single-plane manifolds can also reduce driveability and fuel

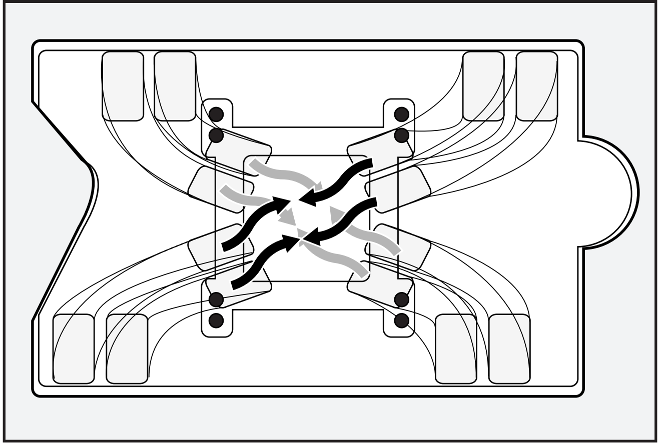
Single-Plane Manifold



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.

Single-Plane Manifold Modeling

Single-Plane Pulse Interference



The typically compact, low-profile design of the single-plane manifold has some 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.

economy. Furthermore, a large-volume, undivided plenum often contributes to low-speed performance 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 on fuel-injection systems). 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, however, 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 brew. 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 cylinder heads (to some degree, these effects are present in all manifold designs). Predicting these will-o'-the-wisp anomalies requires rigorous modeling. Currently, pinning down these problems requires dyno testing with exhaust temperature probes to measure fuel distribution accuracy.

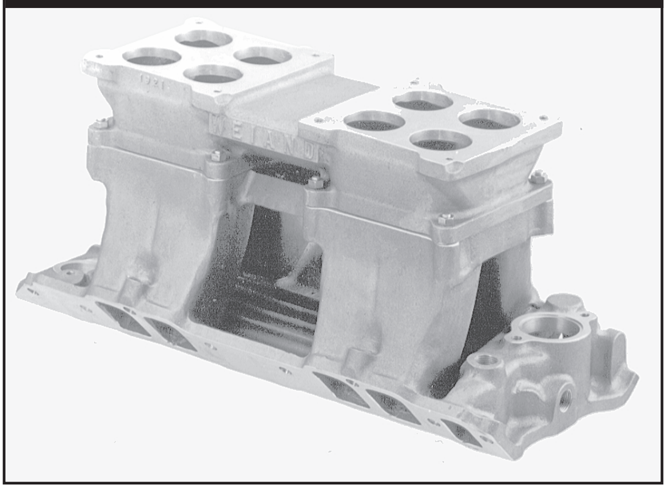
Designers and engine testers have experimented with hybrid single-plane manifold designs that incorporate various dual-plane features. One common modification is dividing the plenum into a pseudo dual-plane configuration. While this does increase signal strength at the carburetor, uneven firing does 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 typically 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 through 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 transform 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., 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 compact manifold design for applications where wide-open-throttle engine speed rarely falls below 4000rpm. 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 derive from its combination of a large common plenum and short, straight, large-volume runners. The large plenum has plenty of space for two carburetors, potentially flowing up to 2000+cfm. 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 usually 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 somewhat more compact 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,

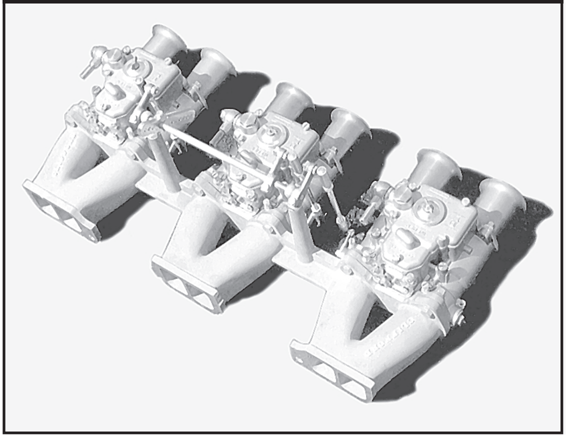
Tunnel-Ram Manifold



This Weiand/Holley 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 naturally-aspirated manifolds listed in the *Induction* menu.

Individual-Runner Manifold Modeling

Individual Runner Manifold



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.

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. 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 has the potential to produce the highest peak horsepower of all the naturally-aspirated 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 sub-

Tuned-Port Injection Manifold Modeling

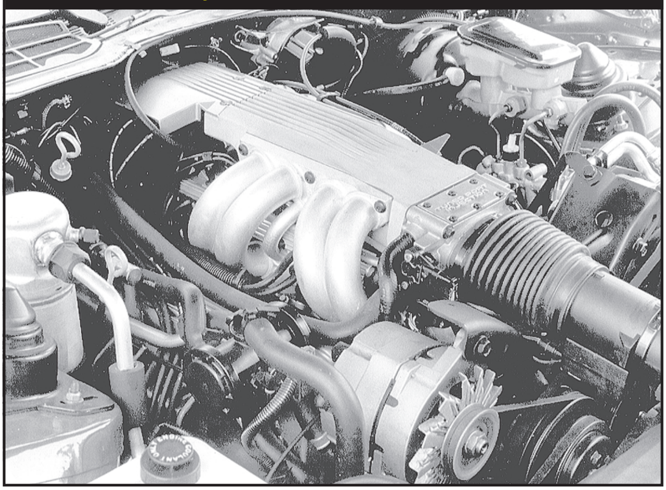
stantial low-end performance benefits (more on that next), at 5000rpm and higher on typical smallblock installations, power can fall below the levels of an average single four-barrel, 780cfm induction system!

Taken as a whole, multiple-carburetor, I.R. induction may seem to offer so much flow capacity, that it must be 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 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 emissions regulations—prevents their wider acceptance.

The simulation model for the **Individual Runner** choice in the Manifold Type 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 systems, the tunnel-ram manifold selection provides a better induction model (see the previous tunnel-ram description).

Tuned-Port Injection Manifold



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.

Sequential-Fire Injection Modeling

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 incorporates 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 are slightly to significantly above a dual-plane up to about 5000rpm, then runner restriction and an out-of-tune condition substantially lowers peak power.

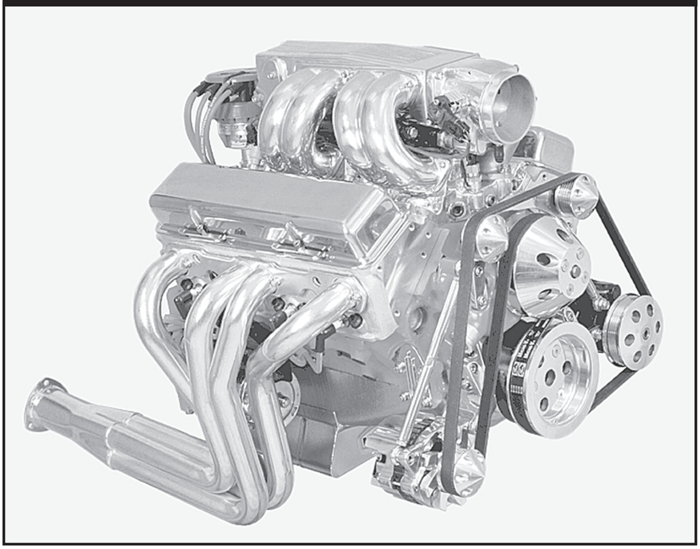
The TPI is a single-plane design that functions like a long-runner tunnel ram. Each runner is completely isolated until it reaches the central plenum. This design tends to maximize pressure-wave tuning and minimize wave interactions. Since fuel is injected near the valve, the TPI system delivers precise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

There is a wide range of aftermarket parts available for the TPI, including enlarged and/or Siamesed runners, improved manifold bases, high-flow throttle bodies, and sensor/electronic modifications. The Tuned-Port Injection selection in the Manifold Type 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" 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 the sequen-

Many TPI and EFI (electronic fuel injection) packages are based on short-runner, high-flow tunnel ram bases and use sequential-fire electronic injectors. Some longer-runner systems, like this manifold from Induction Technology, allow much greater airflow than the original factory TPI and still provide substantial low-speed torque. Model this induction systems with the *Sequential-Fire* manifold selection.

Sequential-Fire Injection System



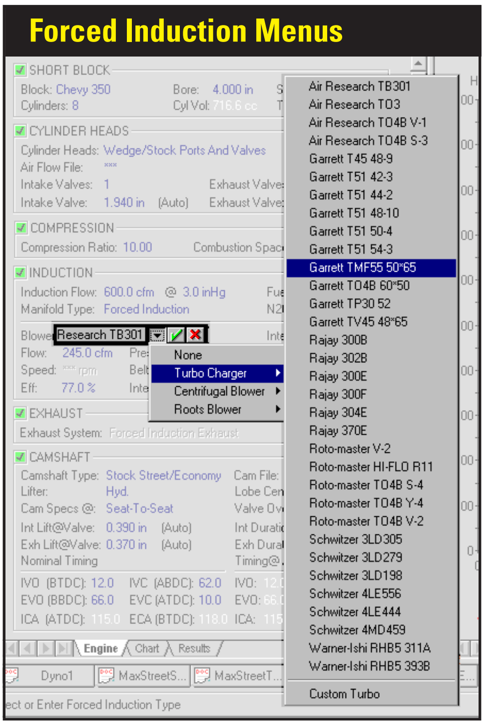
tial-fire manifold (for large-runner packages) to obtain more realistic power curves. Only choose a TPI manifold when the induction system uses a typical small-diameter, long-runner TPI configuration.

Sequential-Fire Injection—The sequential-fire injection manifold models the current state-of-the-art in high-performance manifolds used on many street muscle cars and in some racing applications. Aside from the near perfect fuel distribution provided by sequential injectors, this manifold model is, typically, built around a "tunnel ram" design, with larger and shorter runners than a TPI.

The torque produced by this design is somewhat lower than long-runner TPI in the low rpm ranges, but higher than you might expect since the fuel delivery is so precise. The sequential-fire system really shines at higher engine speeds. Large cross-sectional area, short runners promote cylinder filling and increase horsepower.

FORCED INDUCTION MENUS

The ***Forced Induction*** choice included in the **Manifold Type** menu considerably expands the modeling power of the Dyno2000. In an instant, you can add a positive displacement Roots-type blower, a centrifugal blower (like a Paxton or Vortech), or a turbocharger to any engine. In addition, you can vary maximum boost—or blow-off (wastegate) pressure—pulley ratios, and you can even change blower pressure



The Dyno2000 includes nearly 100 forced induction choices (Turbos, shown here, Centrifugal, and Roots blowers). Selecting a supercharger from any of the three submenus will load the specifications for that device into the INDUCTION category. You may edit these values at any time to determine their affect on engine power. In addition, you can select *Custom* from the bottom of any of the supercharger menus and directly enter of all supercharger specifications.

Forced Induction Modeling

Belt Gear Ratio

Both centrifugal and roots blowers are mechanically driven by the engine. The Belt Gear Ratio (external drive) is the mechanical connection between the engine crankshaft rpm and blower input rpm. This bigblock Chevy pulley setup provides a slight overdrive (a Belt Ratio of 1.20:1).

ratios, surge cfm, and more. And finally, you can test the effects of an intercooler on any of the forced-induction systems.

Because a positive-pressure induction system changes the flow dynamics within the engine, forced induction cannot be used with other manifold types. And, as is detailed in the next section, the exhaust system choices are also overridden with a ***Forced Induction Exhaust*** display. The modeling applied to both the intake and exhaust systems for forced induction will simulate many free-flowing, high-performance designs. The exhaust model is designed to simulate open exhaust, or at least a low-restriction muffler system with large diameter tubing.

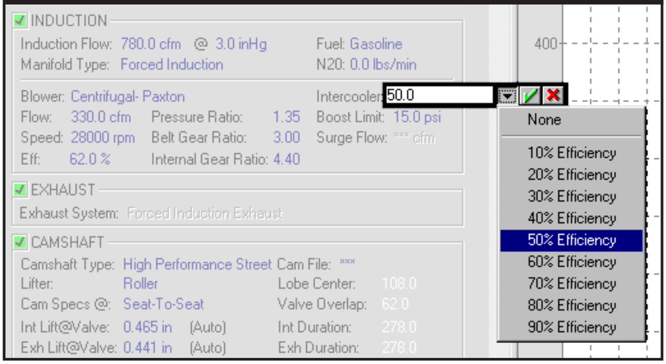
Selecting ***Forced Induction*** activates the lower half of the INDUCTION category. Double-click on the **Blower** field to open a menu containing Turbocharger, Centrifugal, and Roots blower choices. Select from any of the nearly 100 forced induction devices. Specific fields will become active depending on the type of supercharger that was selected. Here is a quick overview of these fields, the superchargers to which they apply, and how they affect forced induction performance:

INDUCTION Category With Forced Induction

The INDUCTION Category displays naturally-aspirated and forced-induction selections. Directly click on any field to change values and evaluate the effects on power, torque, and manifold pressure.

Forced Induction Modeling

Intercooler Menu



The Dyno2000 includes an intercooler model that can be used with any forced induction system. An intercooler reduces induction temperatures from compressing the intake charge that, otherwise, can substantially reduce performance.

Flow—(Turbos, Centrifugals, Roots) This is the flow rate at which the supercharger is most efficient, also called the Island Flow. Typically, the smaller the turbo the lower the Island Flow. Small turbos will spin up faster but have lower overall flow potential.

Pressure Ratio—(Turbos, Centrifugals) This is the ratio of compressor pressures at the Island flow point (ambient vs. output). The higher this number, the more efficient the device performs as an “air compressor.” Roots blowers are a positive-displacement device, so pressure ratio, as used here, does not apply.

Boost Limit—(Turbos, Centrifugals, Roots) This is the pressure at which the wastegate or blow-off valve is activated, maintaining induction pressure at or below this value.

Speed—(Centrifugals) This value is the rotational speed (rpm) at which centrifugal superchargers reaches peak efficiency. There is a similar speed value applicable to turbochargers, however, the model currently incorporated in the Dyno2000 does not support this variable for turbochargers.

Belt Gear Ratio—(Centrifugals, Roots) Both centrifugal and roots blowers are mechanically driven by the engine. The Belt Gear Ratio (external) is the mechanical connection ratio between the engine crankshaft rpm and blower input rpm. This value is multiplied by the Internal Gear Ratio on centrifugal superchargers to determine internal rotor speed.

Surge Flow—(Turbos) The surge flow is the airflow within the Island (most efficient) pressure ratio at which turbocharger flow and internal momentum can “resonate” and produce a pulsing in the induction system. This phenomenon reduces efficiency and engine power output, and it can even damage the turbocharger.

Intercooler Modeling

Efficiency—(Turbos, Centrifugals, Roots) This is a measure of the power consumed by the supercharger compared to the increase in induction pressure at the point of highest efficiency. Roots blowers are often the least efficient, however, they generally deliver substantial induction pressure increases at low speeds. On the other hand, centrifugal and especially turbochargers are more efficient, but require more time to "spin up" to an efficient operating speed.

Internal Gear Ratio—(Centrifugal) Centrifugal superchargers are driven by a mechanical connection to the engine crankshaft. Internal rotor speed is increased by the external Belt Gear Ratio (described earlier), but this speed increase is not a sufficient for most centrifugal superchargers to reach their optimum operating speeds (35,000rpm and higher). An internal gear train is commonly used to further increase rotational speed. The ratio of this internal gearing determines how much faster the turbine rotates over input-shaft rpm. To determine the internal speed of the centrifugal turbine, multiply crankshaft rpm by the **Belt Gear Ratio**, then multiply that by the **Internal Gear Ratio**.

Selecting a supercharger listed in any of the three submenus will load the specifications for that device into the INDUCTION category. You may edit these values at any time to determine their effect on engine power. In addition, you can select ***Custom*** from the bottom of any of the supercharger menus. This option permits direct entry of all supercharger specifications.

Intercoolers

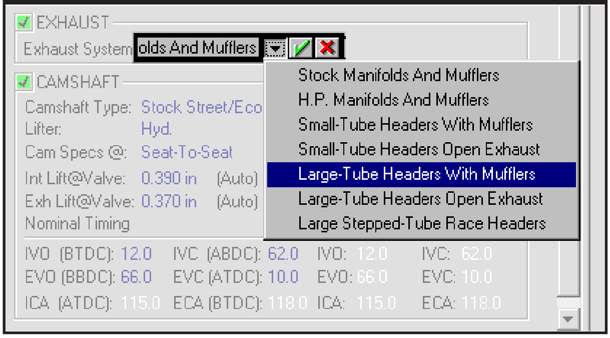
One of the drawbacks to any method of supercharging is increased induction temperatures. High boost pressures can quickly raise charge temperatures more than 200-degrees(F)! These higher temperatures, common on blowers with pressure ratios of 2.0 or higher, can cost more than lost horsepower. Higher temperatures can lead to detonation, increase in octane requirements, and a required reduction in overall ignition timing advance. While induction cooling can improve performance directly from increased charge density (more oxygen and fuel per volume of inducted charge), the additional benefits of reduced detonation and increased reliability make charge cooling an attractive addition to any supercharged high-performance or racing engine.

Charge cooling is accomplished much in the same way that heat is removed from the engine itself. A radiator, called an intercooler, is placed in the air ducting between the supercharger and the intake manifold. Everything from ducted outside air to ice water and even evaporating pressurized liquefied gas (like Freon or nitrous oxide) have been used to remove heat from an intercooler. The average efficiencies for these devices are:

Air-To-Air	**25%,**	**Air-To-Cooler Ducted Air**	**50%**
Air-To-Water	**75%,**	**Air-To-Cooled Water**	**100%**

Exhaust System Modeling

Exhaust System Menu



Flow restriction (back pressure) is accurately modeled using "pressure-drop" techniques. The Dyno2000 can accurately predict engine power changes from various exhaust manifolds and headers of large and small tubing diameters (sizes are relative to the engine under test).

Air-To-Ice Water 120%, Air-To-Evaporating Liquid 120+%

The Dyno2000 includes an intercooler model that can be activated with any forced induction system. Simply double-click on the **Intercooler** field and select an intercooler efficiency from the drop-down list (or directly enter a custom value).

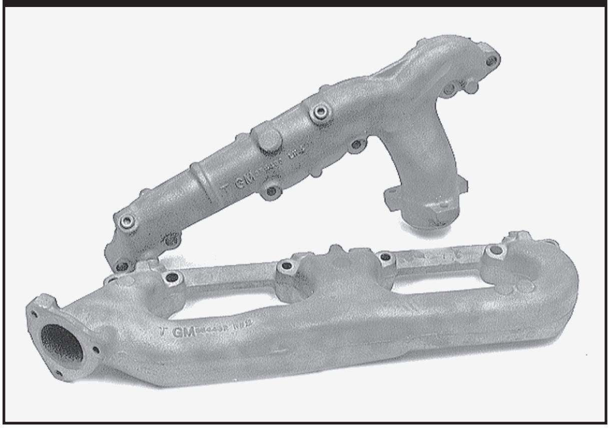
Note: When methanol evaporates it cools the intake charge more than gasoline (the latent heat of vaporization of methanol is greater than gasoline). Therefore, intercooling is somewhat less effective with methanol.

THE EXHAUST MENU

The EXHAUST category establishes an exhaust manifold or header configuration for the simulated test engine. The menu includes seven selections, four of which include mufflers. Since the Dyno2000 is designed to simulate the power levels for an engine mounted on a dyno testing fixture, the exhaust system for muffled engines

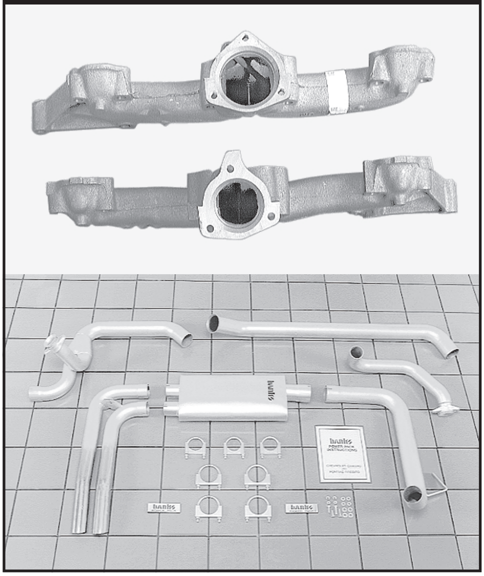
The first choice in the Exhaust menu simulates typical, production, cast-iron, "log-type" exhaust manifolds, where all ports connect at nearly right angles to a common "log" passage. These manifolds are designed to provide clearance for various chassis and engine components and provide much less than optimum exhaust flow.

Stock Exhaust System Manifolds



Exhaust System Modeling

HP Manifolds And Mufflers



The HP Manifolds And Mufflers exhaust-system choice offers a measurable improvement over the stock-exhaust selection. High-performance exhaust manifolds 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.

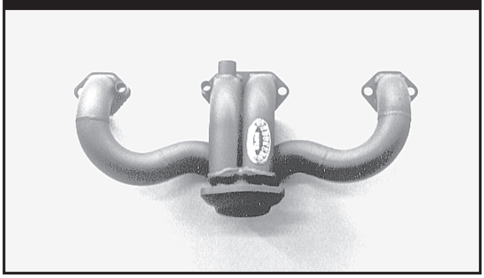
ends at the outlet of the muffler and does not include additional tubing commonly used to route exhaust gasses to the rear of a vehicle.

Each of the exhaust system selections apply a unique tuning model within the simulation. (Refer to the complete *DeskTop Dynos* book available from Motion Software for a more rigorous look at the theory of exhaust-system tuning.)

Exhaust Menu Selections

The exhaust system—perhaps more than any other single part of the IC engine—is a virtual "playground" for high-pressure wave dynamics. These interactions can be solved only by sophisticated, computationally-intensive methods that are only partially modeled in the Dyno2000 (a much more detailed modeling of these interactions is done in the *Dynomation* engine simulation series available from Motion Software

Custom "Manifolds"



Here are excellent examples of high-performance "manifolds" from Hooker Headers. The low-restriction manifolds fit 1992-1995 Corvettes with an LT1 engine. When used with mufflers, model this system using the *H.P. Manifolds And Mufflers* menu choice.

in 2001). While flow restriction (back pressure) is accurately modeled using "pressure-drop" techniques, the Dyno2000 does not resolve specific header dimensions. However, the Dyno2000 can accurately predict engine power changes from various exhaust manifolds and headers of large and small tubing diameters (sizes relative to the displacement of the engine under test).

The exhaust menu choices are described in the following sections. Use this information to make the most appropriate choice for your test engine.

Stock Manifolds And Mufflers—The first choice in the Exhaust menu simulates the most restrictive exhaust system. It 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. These manifolds are designed more to minimize clearance problems with 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.

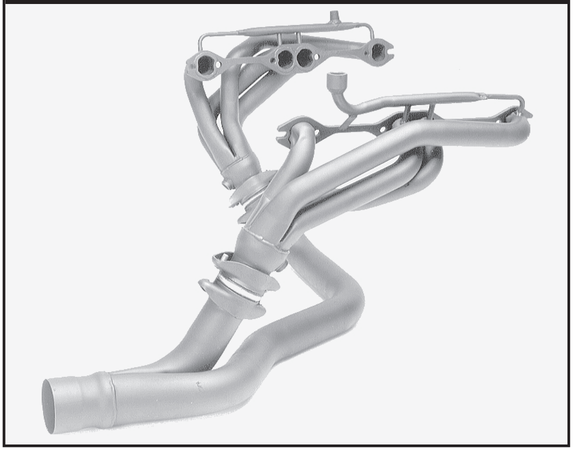
H.P. Manifolds And Mufflers—This choice offers a measurable improvement over the stock exhaust system modeled in the previous selection. The high-performance exhaust manifolds simulated here are designed to improve exhaust gas flow and reduce system restriction. They are usually a "ram-horn" or other "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 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 ABOUT ALL HEADER CHOICES: *Some engines, in particular,*

Exhaust System Modeling

Small Tube Headers



This is the first exhaust-system selection that begins to harness the tuning potential of wave dynamics in the exhaust system. While the system pictured here is not a "true" header, this tubular exhaust system from Edelbrock for late model cars and trucks offers some wave-dynamic scavenging.

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 Dyno2000 simulation. Instead, the headers are assumed to deliver a scavenging wave only to the cylinder that generated the initial pressure wave.

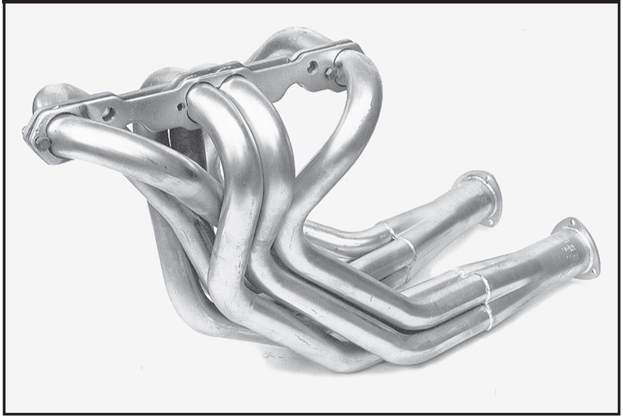
Note About Tubing Sizes For All Header Choices: The following rules of thumb give approximations of tubing diameters used by the simulation: Headers with tubes that measure 95% to 105% of the exhaust-valve diameter are considered "small" for any particular engine; tubes that measure 120% to 140% of the exhaust-valve diameter are "large" tube headers.

Small Tube Headers 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.

Note: 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 Exhaust 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 below peak-torque engine speeds. These headers typically show optimum benefits on smaller displacement engines, and may produce less power on large displacement

Exhaust System Modeling

Large Tube Headers



Typical large-tube headers are designed for high-performance street and racing applications in mind. The better pieces have 3- to 4-inch collectors and 1-3/4- to 2-3/8-inch primary tubes (depending on whether they were designed for smallblocks or bigblocks).

engines.

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 show benefits on smaller displacement engines but may produce less power on large-displacement, big-block engines.

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.

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 engines. These headers may produce less power on small-displacement engines operating in the lower rpm ranges.

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.

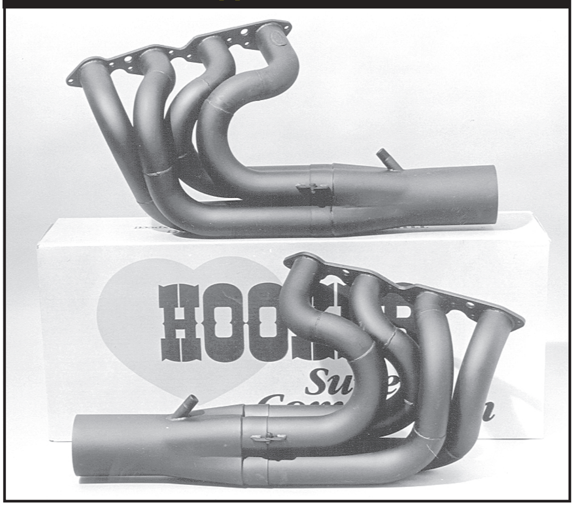
The primary tubes modeled by this menu selection are considered "large," and should be interpreted to fall within a range of dimensions that are commonly associated with applications requiring optimum power at peak engine speeds. These headers typically show benefits on high-rpm racing smallblocks or large displacement big-block engines. These headers may produce less power on small-displacement engines, particularly those operating in the lower rpm ranges.

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.

The "stepped" design of the primary tubes can reduce pumping work on some engines. As high-pressure compression waves leave the port and encounter a step in the primary tube, they return short-duration rarefaction waves. These low-pressure "pulses" moves back up the header and assists the outflow of exhaust gasses. When rarefaction waves reach the open exhaust valve, they help 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

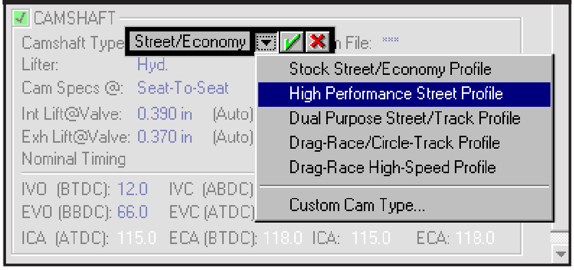
Large Tube Stepped Racing Headers



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 ProStock 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.

Camshaft Modeling

Camshaft Menu



The Dyno2000 can test the effects of cam timing changes in seconds. Several cam profiles are included in the drop-down menu, and you can easily input any custom timing and valve lift specs. Test cams from manufacturer catalogs or load camfiles directly from the Motion Software *CamDisk*™ containing over 1200 read-to-test cams.

should be interpreted to fall within a range of dimensions that are commonly associated with applications requiring optimum power at peak engine speeds.

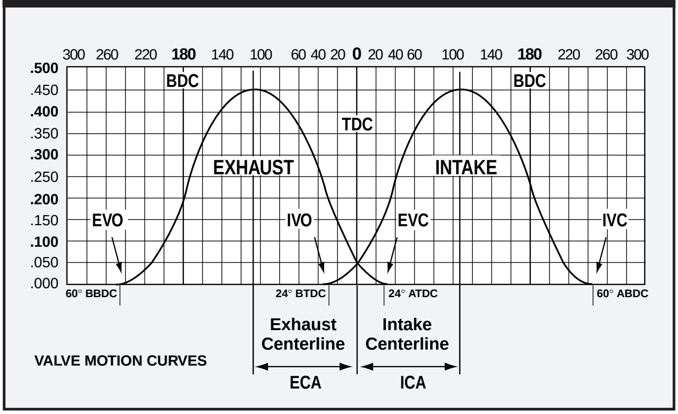
CAMSHAFT MENU

The final component category allows the selection of the single most important part in the IC engine: the camshaft. For many 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 physics within 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 accurately *predict* the outcome.

The Dyno2000 makes it possible to test the effects of cam timing in seconds. The

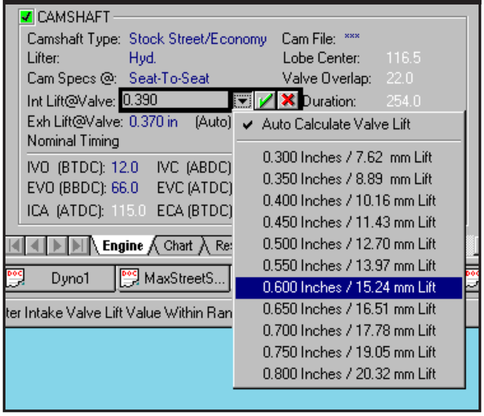
The best way to visualize camshaft timing is to use this "twin-hump" event drawing. It shows valve motion for the exhaust lobe on the left and the intake lobe on the right, positioning the valve overlap and TDC at the center. Study this picture. It will help you quickly evaluate cam timing specs and visualize how they relate to one another.

Valve Motion Diagram



Camshaft Modeling

Valve Lift Menu



Selecting (placing a check mark next to) *Auto Calculate Valve Lift* will automatically calculate appropriate valve lifts for camshafts listed in the Camshaft Type drop-down menu. To manually select valve lift from the drop-down menu, or to directly enter a custom value, make sure that the Auto Calculate Valve Lift feature is turned off (no check mark next to Auto Calculate).

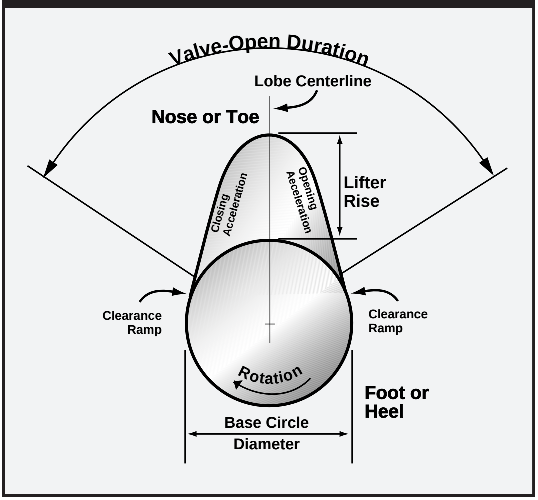
ability of the program to take multiple elements into consideration and "add up the effects over time" is key to analyzing the effects of camshaft timing changes.

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.

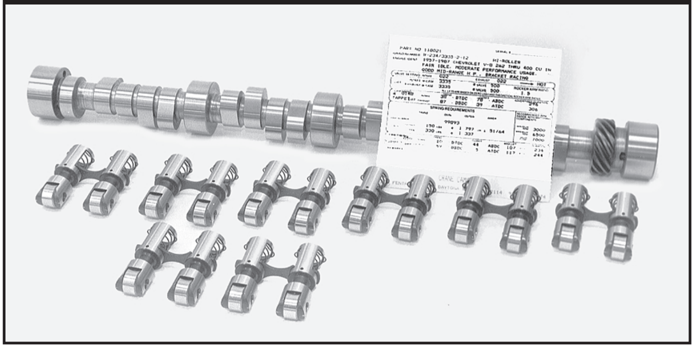
The camshaft is a round shaft incorporating cam *lobes*. The *base circle diameter* is the smallest diameter of the cam lobe. *Clearance ramps* form the transition to the *acceleration ramps*. The lifter accelerates up the *clearance ramp* and continues to rise as it approaches the *nose*, then begins 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*. *Valve-open duration* is the number of crankshaft degrees that the valve or lifter is held above a specified height (usually 0.006-, 0.020-, or 0.050-inch). A symmetric lobe has the same lift curve for opening and closing.

Cam Basics



Camshaft Modeling

Common "Cam Card" Timing



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.

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 several 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 rates at which the valves are accelerated open and then returned to their 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.

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 the basic six events. Derivative events are:

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

Camshaft Modeling

The first four basic timing points (IVO, IVC, EVO, EVC) pinpoint the "true" beginning and end of the four engine cycles. These valve opening and closing points indicate when the function of the piston/cylinder mechanism changes from intake to compression, compression to power, power to exhaust, and exhaust back to intake. For much more in-depth information about cam timing, refer to the complete book *DeskTop Dynos* available from Motion Software.

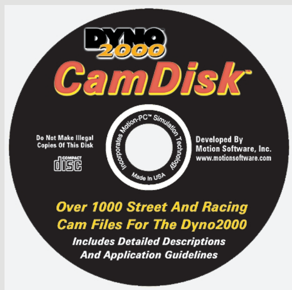
Camshaft Menu Choices

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 (described later in this chapter) 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, EVC, Intake Centerline, Intake Lobe Center Angle, Intake Duration, and Exhaust Duration are loaded into the CAMSHAFT category along with the camshaft description. Valve lifts can be manually entered or automatically calculated by the program (refer to the following Notes).

Note 1: *The* **intake** *and* **exhaust valve lifts** *for any of the camshafts listed in the* **Camshaft Type** *drop-down menu can be automatically calculated if you choose*

ISKY Cam Catalog

CamDisk™ With 3500+ Camfiles

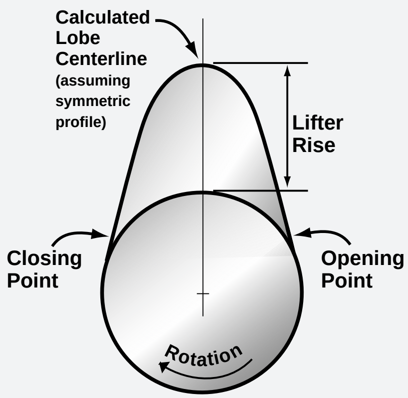


The new CamDisk2™ from Motion Software includes over 3500 camfiles with application info. It's the fastest way to load and test cams in the Dyno2000.

Several of the "generic" grinds included in the Dyno2000 Camshaft menu were modeled after profiles from the ISKY CAMS catalog.

Camshaft Modeling

Developing Valve Motion Curves



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

The Dyno2000 models symmetric valve motion curves from six data points, three for each lobe: 1) the opening point, 2) the closing point, and 3) the point of maximum lobe lift. Although some cam grinds are asymmetric, performance differences between a symmetric model and actual asymmetric valve motion is quite small.

Auto Calculate Valve Lift *available from either* **Valve Lift** *menu (intake or exhaust). Calculated valve lifts are based on the valve-head diameters and camshaft timing. The auto-lift calculation feature will be suspended and, instead, permanent lift values will be used for any camshaft when you re-select* **Auto Calculate Valve Lifts** *to "uncheck" the feature (turn it off).*
Note 2: *If* **valve diameters** *are also being automatically calculated by the Dyno2000 (see page 29)—cylinder-bore diameter and a cylinder head selection must be complete before the program can calculate valve diameters and, consequently, valve lifts.*

Stock Street/Economy Profile—This first cam selection is designed to simulate a typical factory-stock cam. All cam timing events displayed are seat-to-seat measurements.

The EVO timing maintains 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 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.

High Performance Street Profile—This profile is designed to simulate a high-performance "street" camshaft. All cam timing events displayed in the CAMSHAFT category 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 overlap allows 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. The *High Performance Street Profile* choice can be used with hydraulic, solid, or roller lifters, and the simulation will accurately model this cam with any lifter-acceleration rate (choose hydraulic lifters for increased driveability and solid or roller lifters for more high-performance oriented applications). This cam is nearly identical to the *ISKY Hi-Rev Flat-Tappet* cam part 201025.

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

EVO timing on this camshaft is beginning to move away from specs that would be expected for optimum combustion pressure utilization, with more of an emphasis on 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 designed to harness exhaust scavenging. The modestly aggressive overlap allows 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 hydraulic, solid, or roller lifters, and the simulation will accurately model this cam with any lifter-acceleration rate (choose hydraulic lifters for more street-oriented applications and solid or roller lifters for more high-performance oriented applications). The profile of this cam is nearly identical to the *ISKY Hydraulic Series* cam part 201281.

Drag-Race/Circle-Track Profile—This profile is designed to simulate a competition aftermarket camshaft. All cam timing events displayed 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, intended to optimize exhaust scavenging effects. This aggressive overlap is designed for higher engine speeds with open headers and allows exhaust gas reversion into the induction system at lower rpm, 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 substantial power at higher engine speeds and is especially effective in lightweight vehicles. The *Drag-Race/Circle-Track Profile* choice can be used with solid or roller lifters, and the simulation will accurately model this cam with either lifter-acceleration rate (choose solid lifters for less valvetrain punishing applications and roller lifters for higher power drag-racing applications). The profile of this cam is similar to the *ISKY Oval Track Flat Tappet Series* cam part 201555.

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

All timing events on this camshaft are designed to optimize power on large displacement 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 for fuel consumption. This *Drag-Race High-Speed Profile* is typically used with roller lifters. The profile of this cam is similar to *ISKY Roller Series* cam part 201600.

Note: Each of the previous application-specific cams can be modified in any way by directly entering valve-event or other cam-timing specs (more on this in the next few sections).

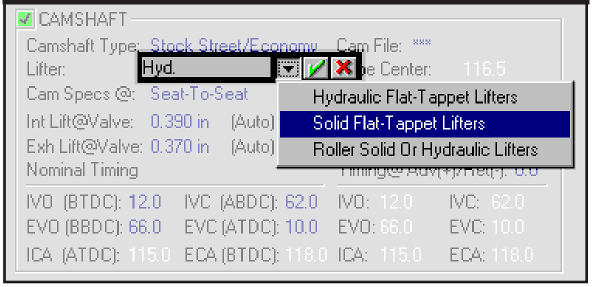
LIFTER MENU

The Dyno2000 uses sophisticated modeling to simulate camshaft and valvetrain motion, but you should keep in mind that valve motion curves for both the intake and exhaust valves are being calculated from only six data points, three for the intake valve and three for the exhaust valve.

The three points for each simulated motion curve are the opening point, closing point, and the maximum lobe lift. From these points, and the lifter-type selection, the program creates motion curves that pinpoint valve lifts at each degree of crank position. While the results are remarkably accurate, the Dyno2000 cannot model subtle differences between cam grinds that use the same event timing and valve lift specs. Furthermore, the Dyno2000 develops a symmetric valve motion curve (meaning that the "opening" side of the lobe has an identical shape as the "closing" side). Asymmetric modeling is impossible with only three data input points, luckily, perfor-

Camshaft Modeling

Lifter Menu



The Dyno2000 uses increasing valvetrain acceleration to model hydraulic, solid, and roller-lifters. This is a good assumption, since most cam profiles have predictable valve acceleration rates. However, some roller-lifter street cams do not to have high acceleration, but instead use roller lifters to optimize reliability. Refer to the accompanying text for help in selecting a lifter choice.

mance differences between symmetric and asymmetric valve motions are often quite small.

The **Lifter** menu offers three choices. Each choice instructs the simulation to apply a unique "ramp-rate" model to the valve motion curve:

Hydraulic Flat-Tappet Lifters—The lowest acceleration is assigned to the first menu choice. Hydraulic lifters incorporate a self-adjusting design that maintains zero lash 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.

Solid Flat-Tappet Lifters—The next highest acceleration rate is assigned to Solid 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 characteristics are used by the Dyno2000 to derive a more aggressive valve-motion curve.

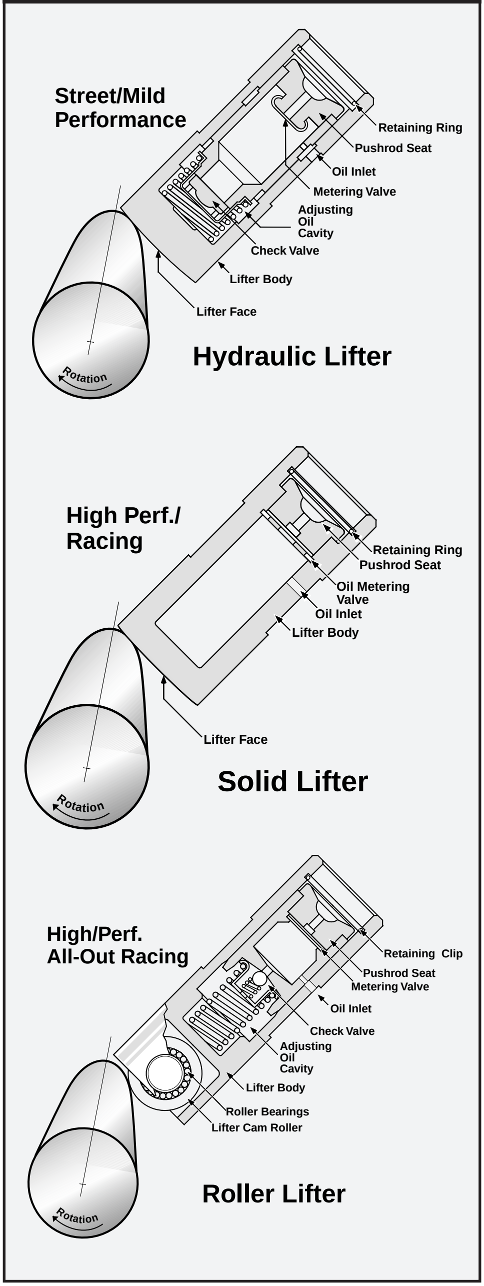
Roller Solid Or Roller Hydraulic Lifters—The highest acceleration rates are applied to Roller Lifters. This choice generates very aggressive ramp acceleration rates and derives valve motion curves appropriate for most high-performance and racing, roller-lifter camshafts.
Note: Please refer to the next section for additional information on selecting the proper lifter choice for your application, especially for roller-lifter camshafts.

Making The Best Lifter Choice

The simulation uses increasing valvetrain acceleration to model hydraulic, solid,

Camshaft Modeling

Basic Lifter Choices



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 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 will produce optimistic simulated power curves. 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:

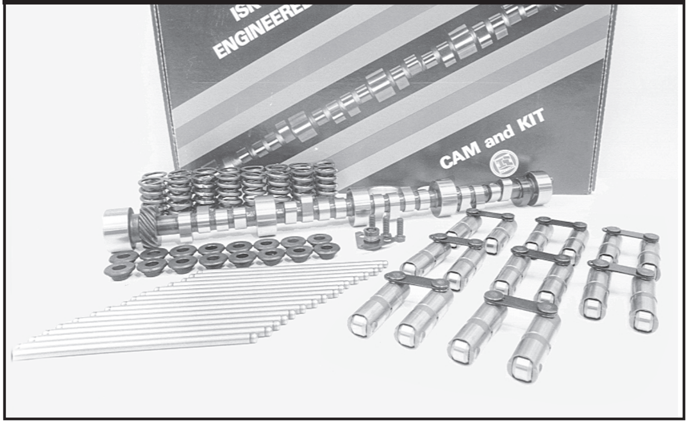
Menu Choice	Application
Hydraulic Flat-Tappet	Street/Mild Perf.
Solid Flat-Tappet	HP/Mild Racing
Roller	Very HP/Racing

If the cam you're modeling is a roller-lifter grind but incorporates a mild-street profile, select *Hydraulic* or *Solid Flat-Tappets* from the menu, since this choice will produce a lift curve that best matches a mild street camshaft. On the other hand, if the cam is a high-performance—or high acceleration—grind, select *Solid Lifters* since this will model the faster acceleration rates of aggressive performance grinds. If you are modeling a 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 pre-

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*.

Camshaft Modeling

Lifter Choices In The Dyno2000



If your cam uses roller lifters but is a mild street profile, select *Hydraulic* or *Solid Flat-Tappets* 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.

dictions from the *Roller Lifter* selection.

TIMING METHOD MENU

The Dyno2000 will simulate camshaft motion for both *Seat-To-Seat* and *0.050-inch cam timing specifications*. Whenever you change the **Cam Specs** field and alter the camshaft-timing method, <u>any currently displayed timing events are NOT changed; it's up to you to modify/enter the correct timing values</u>. A warning message will be displayed indicating that the timing method has been changed. In addition, the new selected timing method will be displayed next to **Cam Specs @:** in the CAMSHAFT category.

The basic cam timing events are affected by changes in timing methods. Changing the cam timing method affects IVO, IVC, EVO, EVC, and the calculated intake and exhaust durations. 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 changes in the timing measurement method because none of these specs are derived from any of the basic four valve events.

Cam Timing Method Menu

The Dyno2000 will simulate camshaft motion for both Seat-To-Seat and 0.050-inch cam timing. However, the internal simulation model requires seat-to-seat event timing to accurately calculate the beginning and end of mass flow in the ports and cylinders and must *derive* seat-to-seat timing from 0.050-inch figures. Unfortunately, this cannot be done with high accuracy. So, whenever possible enter seat-to-seat timing to obtain the most accurate simulation results.

Camshaft Modeling

Seat-to-seat timing method—This 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 seat-to-seat timing methods:

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

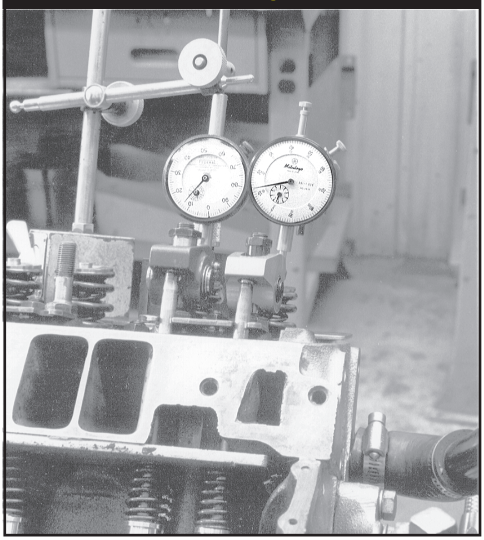
The timing specs measured using these methods are meant to approximate the actual valve opening and closing points that occur within the running engine. Because of this, seat-to-seat valve events are often called the *advertised* or *running* timing. The Dyno2000 needs this information to calculate the 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, directly entering seat-to-seat timing specifications will produce the most accurate simulation results.***

0.050-inch cam timing—This timing method is widely used by cam manufacturers. 0.050-inch cam timing points are always measured at:

0.050-inch LIFTER rise for both intake and exhaust.

This measurement technique is based on the movement of the cam follower (lifter) rather than the valve. Since the lifter is rapidly opening or closing at 0.050-inch lift, this technique provides an accurate "index" for cam-to-crank position, and

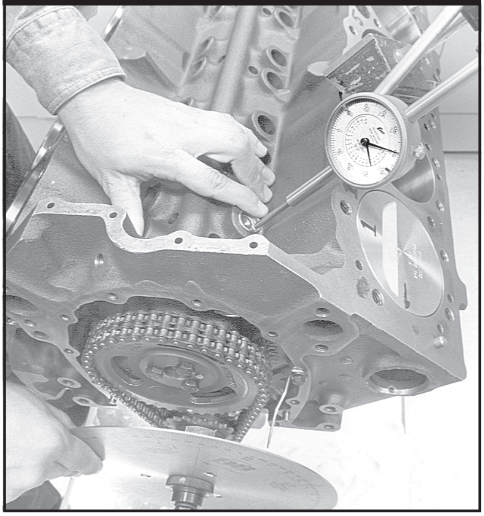
Seat-To-Seat Timing Method



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 <u>valve rise</u>, 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.

Camshaft Modeling

0.050-Inch Timing Method



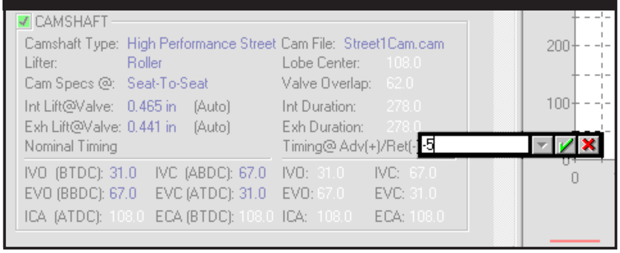
The 0.050-inch lifter rise cam timing method measures valve timing when the lifter has risen 0.050-inch off of the base circle of the cam. In the setup pictured here, the dial indicator is positioned on an intake lifter; the 0.050-inch valve timing point can now be read directly off of the degree wheel attached to the crankshaft. Timing specs measured using this method are not meant 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 Dyno2000 are internally converted 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.

is a wonderful way to accurately check the installation (index) of a camshaft. However, 0.050-inch timing does not pinpoint when the intake and exhaust valves open or close; essential data needed to perform any engine simulation. While you will always find 0.050-inch lifter rise timing points published on the cam card and in many cam manufacturer's catalogs, the Dyno2000 must internally convert 0.050-timing to seat-to-seat figures. And unfortunately, this can introduce some degree of error into valve-motion calculations. So to emphasize a point: *When ever possible, use seat-to-seat timing specifications; they produce the most accurate simulation results.*.

ADVANCE/RETARD MENU

The Dyno2000 allows direct entry of a camshaft advance or retard value. Changing this specification 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

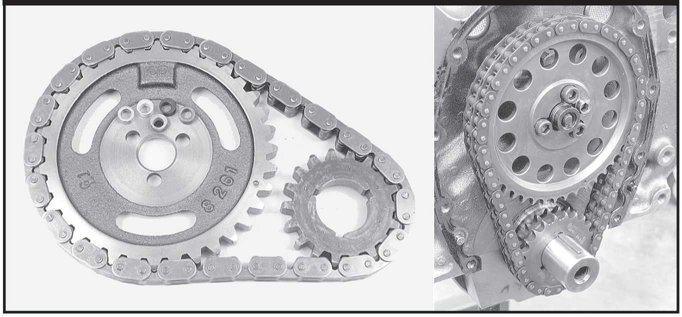
Advance/Retard Menu



The Dyno2000 allows direct entry of camshaft advance or retard. Changing this specification from zero (the default) to a positive value advances the cam; negative values retard the cam. See text for more information on how these changes affect engine output.

Camshaft Modeling

Changing Camshaft Advance/Retard



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

relative to the crankshaft. Why is this done? It is just about the only valve-timing change available to the engine builder after the camshaft has been purchased. 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.

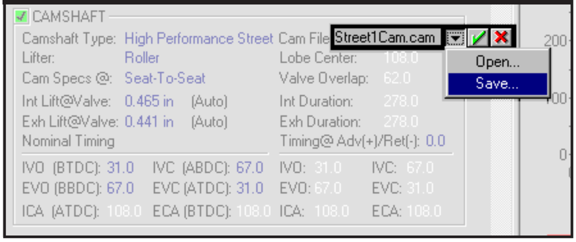
SAVING CAM SPECS MENU

The Dyno2000 can save and retrieve cam file specifications. Simply select the **Cam File** field in the CAMSHAFT category. Choose **Save** from the drop-down menu. Enter a filename then click **OK**. A ".cam" is automatically added to the filename and the file will be saved in the selected directory.

Note 1: If you change any of the on-screen cam specs after saving or retrieving cam files, the cam file will <u>not</u> be automatically updated (if desired, choose the **Save** function again to update the cam files on disk).

Cam Math Calculator

Saving And Retrieving Cam Files



The Dyno2000 can save and retrieve cam file specifications. Simply select the Cam File field in the CAMSHAFT category. A ".cam" is automatically added to the file name you select to save the cam specs.

Note 2: Dyno2000 cam files are not compatible with cam files from the Motion-PC Dyno Shop v.2.8.7. If you wish import cam files from this previous version of the Dyno, print out a test sheet from v.2.8.7, then manually load the cam specs into the Dyno2000, finally re-save the cam specs to your hard drive.

THE CAM MATH CALCULATOR

As discussed previously, the basic four valve events (IVO, IVC, EVO, EVC) are required by the Dyno2000 to pinpoint when the intake and exhaust valves open and close. The IVO and EVO signal the beginning of mass flow in the intake and exhaust ports. The closing points, IVC and EVC, mark the end of mass flow. Unfortunately, many cam catalogs and other printed materials ONLY publish the lobe center angles and duration values, leaving the conversion to IVO, IVC, EVO, and EVC up to the frustrated simulation user.

Now, the "frustration factor" has been reduced. The **Cam Math Calculator,** included in the Dyno2000, instantly converts the lobe-center angle, intake centerline, and the duration values into IVO, IVC, EVO, and EVC events. These values can be loaded into the main Component Selection screen (in the CAMSHAFT Category) and used in the next simulation. In order for the Cam Math Calculator to determine all four valve events, BOTH the lobe-center angle AND the intake centerline must be available. Without the intake centerline, there is no way to determine how the cam is "timed" or "indexed" to the crankshaft. Many, unfortunately not all, cam manufacturer catalogs provide sufficient information to use the Cam Math Calculator to determine valve event timing. If you have a catalog that does not provide this information, try another cam manufacturer, or consider purchasing the ***CamDisk*** from Motion Software that provides over 1200 read-to-load cam files for the Dyno2000. **Note:** You can even obtain cam (and flow and engine) files from several sources on the Internet. The popularity of the Dyno2000 has engendered "unofficial" support sites that you may find helpful in your engine development (Motion Software, Inc., does not endorse, guarantee, or accept any responsibility for the accuracy or usability of any of the information obtained from "non-official" sources.)

Before you open the Cam Math Calculator, select the appropriate cam timing method from the **Cam Specs @** menu located in the CAMSHAFT category on the

Cam Math Calculator

Cam Timing Method Menu

Before you open the Cam Math Calculator, select either the Seat-To-Seat or 0.050-inch cam timing methods. Remember, seat-to-seat event timing will produce the most accurate simulation results.

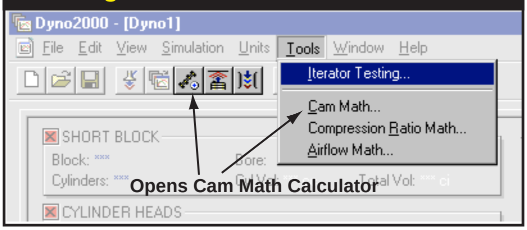
main Component Screen. This will establish how the timing points are applied to the simulated engine by the Calculator.

Important Note: The Cam Math Calculator screen indicates whether the displayed specs used in cam-math calculations and, potentially, applied to the simulated engine are based on the 0.050-inch or Seat-to-Seat timing methods (see photo, page 80). You can switch between these timing methods by closing the Cam Math Calculator and choosing a timing specification from the **Cam Specs @** field in the CAMSHAFT category. Remember, whenever you have a choice of cam specs, always use seat-to-seat timing values; the simulation results will have the highest accuracy.

The Cam Math Calculator is activated by selecting **Cam Math** from the **TOOLS** drop-down menu or by clicking on the Cam Math Calculator button on the Toolbar. If IVO, IVC, EVO and EVC cam timing values were already entered in the CAMSHAFT category (on the main Component Selection Screen), the Cam Math Calculator will display the lobe-center angle, intake centerline, and duration values for the current cam and accept any changes. On the other hand, if you have not yet entered valve-event timing, the Cam Math Calculator will display blank fields, and allow the input of centerline, duration, and valve-lift specs. As you fill in the fields, the corresponding IVO, IVC, EVO and EVC points will be calculated and displayed. You may then either accept the calculated values and transfer them to the CAMSHAFT category (on the Component Selection Screen) by pressing the **Apply** button or discard

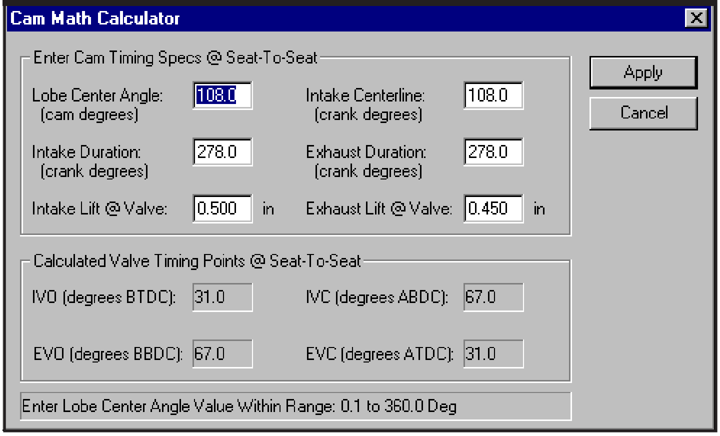
The Cam Math Calculator is an easy-to-use tool that converts LCA, ICA and duration values to the IVO, IVC, EVO, and EVO timing points needed to perform a simulation. Open the calculator by selecting Cam Math from the Tools menu or clicking on the Cam Math Calculator button in the Toolbar.

Starting The Cam Math Calculator



Cam Math Calculator

Cam Math Calculator

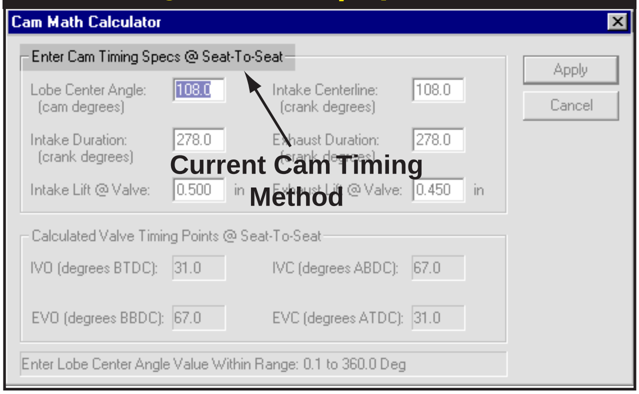


The *Cam Math Calculator* allow direct entry of cam data from cam manufacturer's catalogs. It also simplifies changing lobe-center angle, intake centerline, intake and exhaust duration, and valve lift specifications.

the new values and close the Calculator by pressing **Close**.

You also will find the Cam Math Calculator a handy tool for testing changes made to the lobe-center angle, the intake centerline, intake duration, and exhaust duration values. Combined with the ability to change IVO, IVC, EVO, EVC, and overall advance and retard from the Component Selection screen, the Dyno2000 with the Cam Math Calculator allows quick, "what-if" manipulation of EVERY cam timing event.

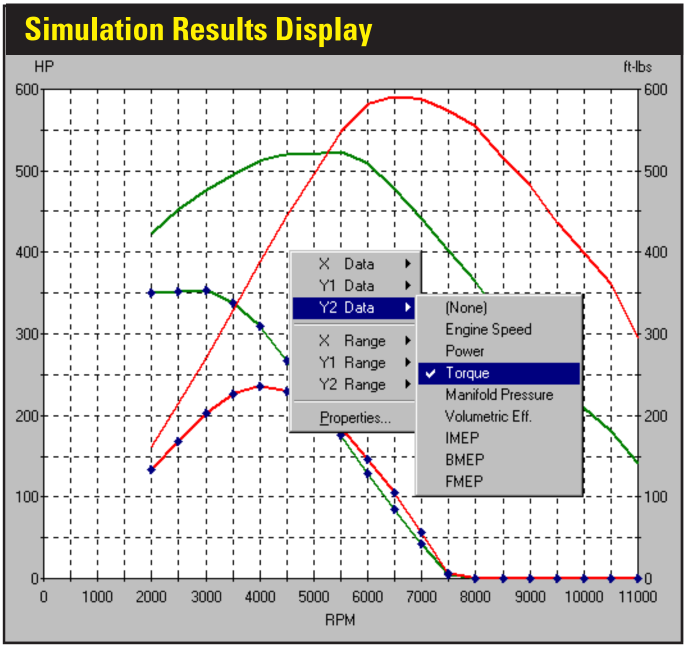
Cam Timing-Method Display



The *Cam Math Calculator* screen indicates whether the displayed specs will be used in the simulation as 0.050-inch or Seat-to-Seat timing values. You can switch between these methods by closing the Cam Math Calculator and choosing the timing specification from the *Cam Specs @* field in the CAMSHAFT category.

Dyno2000 Advanced Engine Simulation

RESULTS SCREEN



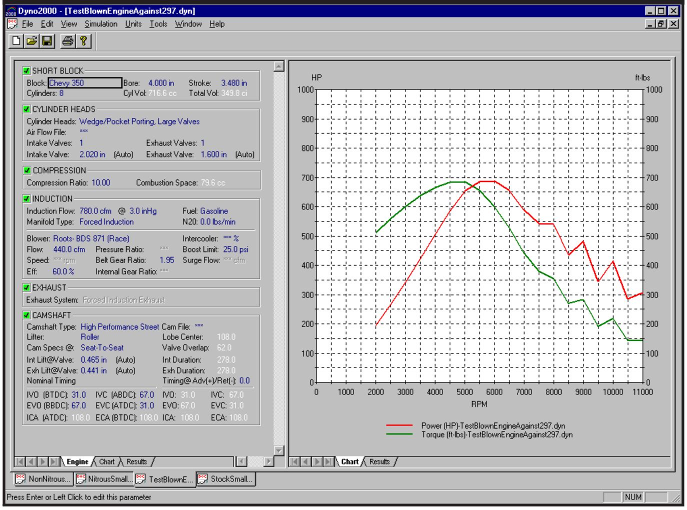
The speed and ease of engine component entry in the Dyno2000 is complemented by the power and versatility of the simulation results displays. Almost the same instant that all the component categories have been completed (all categories have green Status Boxes) the simulation results will be displayed on a fully-scalable precision graph. The display graph can be customized to display virtually any engine variable on any axis. Auto scaling or manual axis scaling are easily setup by right-clicking on the graph (shown here). Up to four engines can be compared at once. And a comprehensive "table" display shows exact horsepower, torque, rpm, induction pressure, cylinder pressure, engine friction, and more! The Dyno2000 will show you what you are looking for, fast!

The **Simulation Results** display is composed of several elements that will help you retrieve the most information from any simulation as quickly and easily as possible:

1) The Main Program Screen (photo, next page) is divided into two sections (called panes), with the component selection categories on the left and the results screen on the right (by default). The center divider between each pane can be moved (click and drag) to resize the results screen to suit your requirements. The graph will redraw and re-scale to take advantage of changes in display

Simulation Results Display

Two Panes Of Main Program Screen



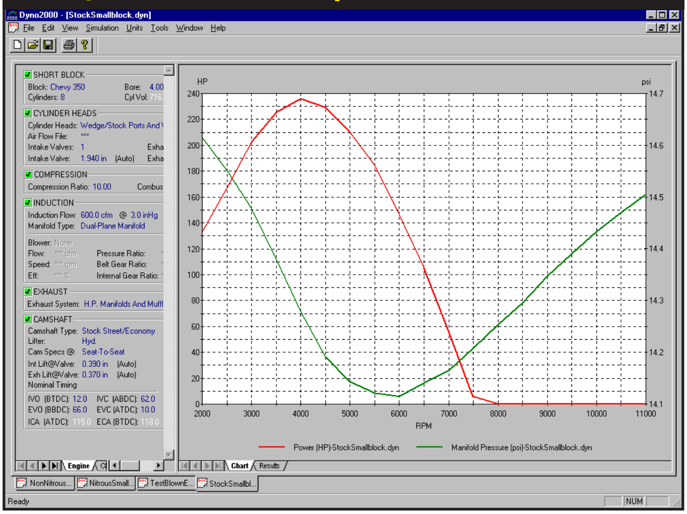
The Main Program Screen is divided into two sections (called panes), with the component selection categories on the left and the results screen on the right (by default). The center divider between each pane can be moved (click and drag) to change the size of the results screen to suit your requirements. The graph will redraw and re-scale.

area.

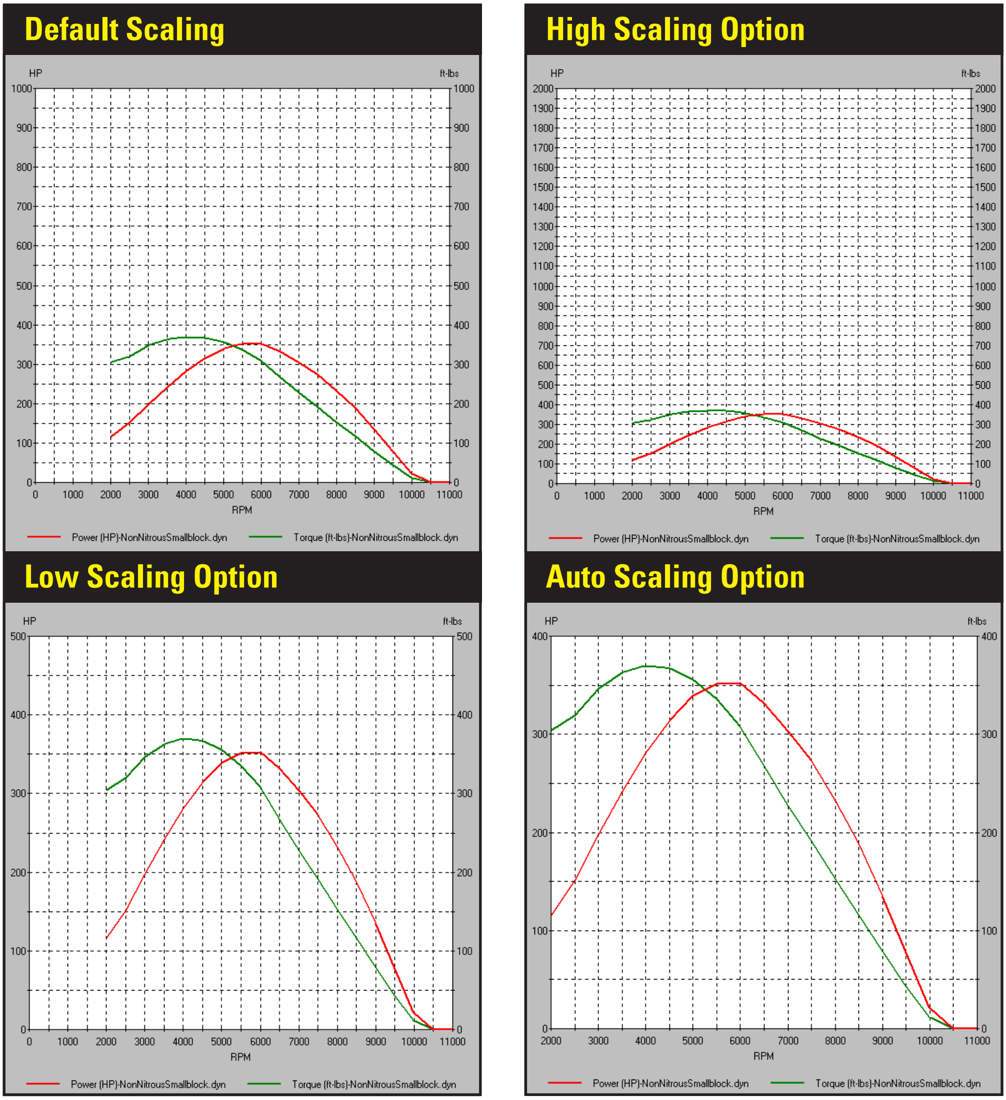
2) The results graph consists of three axis, a left, right, and bottom (horizontal) axis. Each of these axis can be assigned an engine variable. Currently the Dyno2000 will graph the following variables: Rpm, Horsepower, Torque, Intake Manifold Pressure, Volumetric Efficiency, Imep (Indicated Mean Effective Pressure), Bmep (Brake Mean Effective Pressure), and Fmep (Friction Mean Effective Pressure). Right click on the graph to display the **Graph Options** menu to

The results graph consists of three axis, a left, right, and bottom (horizontal). Right click the graph and assign any variable to each curve (see photo on previous page). The graph on the right shows how horsepower (red) and manifold pressure (green) varied throughout a test run. Also note that the screen divider has been moved to allow the graph more display area.

Assign Variables To Graph Curves



Simulation Results Display



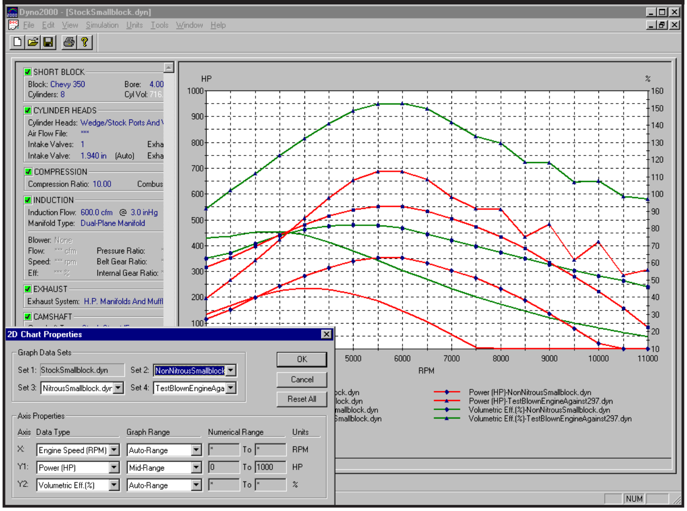
The results graph supports several methods of axis scaling. Each axis will scale to a low, medium, high, and auto-scale ranges.

assign engine variables to graph axis.

3) The results graph supports several methods of axis scaling. Each axis will scale to a low, medium, and high value. Plus auto-scaling can be enabled for any axis. By default, auto-scaling is turned off. This maintains the axis values constant, establishing a fixed baseline so that changes in power or torque are easily distinguished. However, when component changes dramatically alter power (like nitrous-oxide injection or forced induction), the auto-scaling feature will ensure

Simulation Results Display

Compare Up To Four "Open" Engines



A comparison of four engines was setup using the Properties Box. Up to four "open" engines can be compared on any graph. This graph shows how horsepower (red) and volumetric efficiency varied for all four test engines.

that the data curves are always visible and display at 80 to 90% of full graph height for maximum resolution.

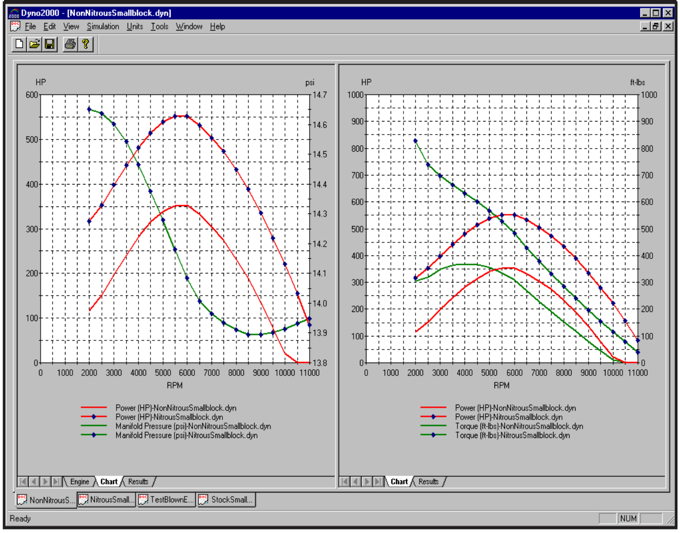
4) The **Graph Properties** dialog screen allows on-graph comparison of up to four engines at once. The engines you wish to include in the comparison must be "open" with active tabs in the Engine Selection Tabs display. Right click the

Ensure Supercharger Boost Pressure

The ability to select the variables that you would like to compare is an extremely power tool. Here horsepower is compared with manifold pressure on a supercharged engine. Notice that the wastegate opens at 7000rpm when manifold pressure reaches about 25psi (above ambient).

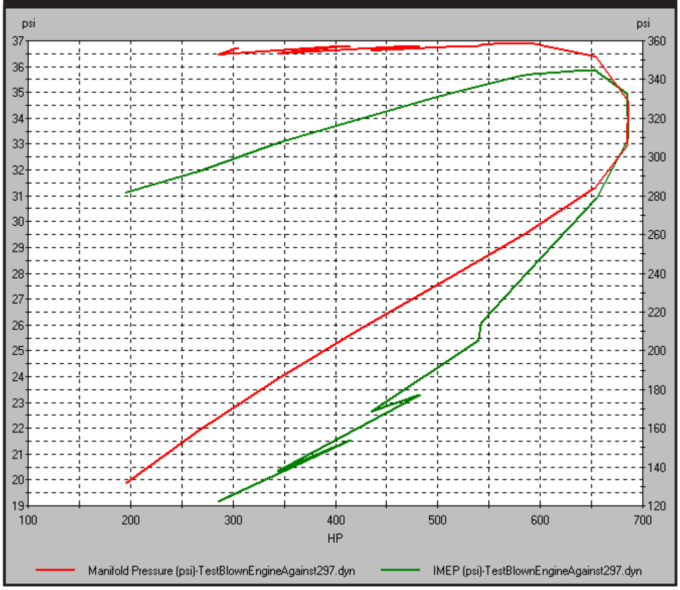
Simulation Results Display

Dual Displays Of Same Engine



Select a Graph display for both panes and set the Options to plot different variables for each graph. View more simulation data and get better insight into the performance potential of any component combination.

Research Nature Of I.C. Engine

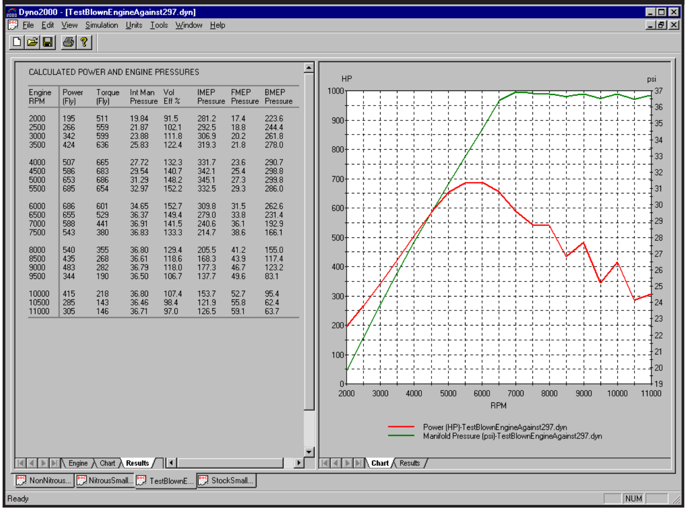


The graphing capability of the Dyno2000 is not limited to standard "power" curves. Here is a display of how induction pressure and Brake Mean Effective Pressure (Bmep) varied in relationship to engine output. It's graphs like this that can lead to new insights and optimum component combinations.

graph to display the **Graph Options** menu then select **Properties**. Use the Graph Data Sets drop-down menus to select from currently-open engines. When you click on **OK**, the graph will redraw with the desired data comparisons. A legend at the bottom of the graph provides a key to all graph curves.

Simulation Results Display

Table Shows Exact Test Results



Engine RPM	Power (Fly)	Torque (Fly)	Int Man Pressure	Vol Eff %	IMEP Pressure	FMEP Pressure	BMEP Pressure
2000	195	511	19.84	91.5	281.2	17.4	223.6
2500	266	559	21.87	102.1	292.5	18.8	244.4
3000	342	599	23.88	111.8	306.9	20.2	261.8
3500	424	636	25.83	122.4	319.3	21.8	278.0
4000	507	665	27.72	132.3	331.7	23.6	290.7
4500	586	683	29.54	140.7	342.1	25.4	298.8
5000	653	686	31.29	148.2	345.1	27.3	299.8
5500	685	654	32.97	152.2	332.5	29.3	286.0
6000	686	601	34.65	152.7	309.8	31.5	262.6
6500	655	529	36.37	149.4	279.0	33.8	231.4
7000	588	441	36.91	141.5	240.6	36.1	192.9
7500	543	380	36.83	133.3	214.7	38.6	166.1
8000	540	355	36.80	129.4	205.5	41.2	155.0
8500	435	268	36.61	118.6	168.3	43.9	117.4
9000	483	282	36.79	118.0	177.3	46.7	123.2
9500	344	190	36.50	106.7	137.7	49.6	83.1
10000	415	218	36.80	107.4	153.7	52.7	95.4
10500	285	143	36.46	98.4	121.9	55.8	62.4
11000	305	146	36.71	97.0	126.5	59.1	63.7

In addition to 2D graphing capability described in the text, a chart display is available by clicking on Table tabs located at the bottom of either pane. The chart lists all engine variables recorded during the simulated dyno run. The exact data values are displayed in 500rpm increments from 2000 to 11,000rpm.

5) In addition to 2D graphing capability described above, a chart display is available by clicking on **Table** tabs located at the bottom of either display pane. The chart lists all engine variables recorded during the simulated dyno run. The exact data values are displayed in 500rpm increments from 2000 to 11,000rpm.

DYNO 2000™ Advanced Engine Simulation

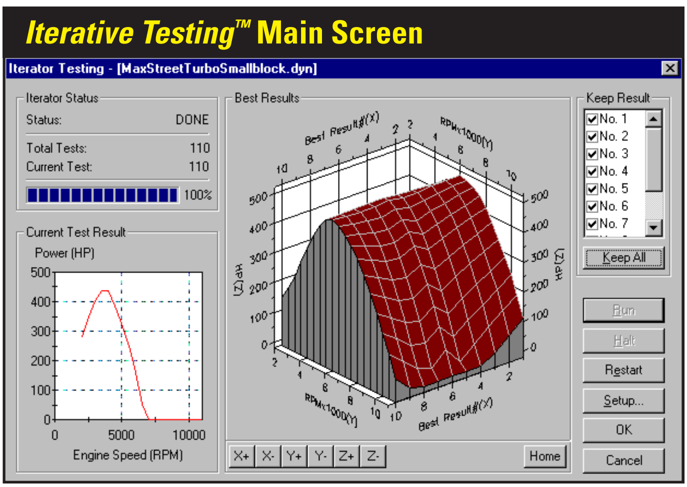
THE ITERATOR™

Before the rapid "what-if" testing possible with the Dyno2000, obtaining data about engine component combinations was an expensive, time-consuming process. Components were assembled into a complete engine, the engine was installed on the dyno, and after initial break-in runs, power testing was performed. This process could easily take several hours, if not days, for each component setup. Add up the cost of the parts, labor, and dyno time, and it obvious why even wealthy engine builders/owners could only slowly build a file of engine-test data. Sorting through the tests and analyzing results rarely required extensive cataloging and sorting; there were simply too few tests and never enough data.

While the cost of engine building and dyno testing have certainly not decreased over the past few years, the ability to fill file cabinets with *simulated* dyno tests is available to anyone. In fact, many enthusiasts become "bogged down" in an over-abundance of test data. Sorting through the results, analyzing the best power curves, and selecting promising component combinations has turned into a job nearly as difficult as the old trial-and-error dyno testing.

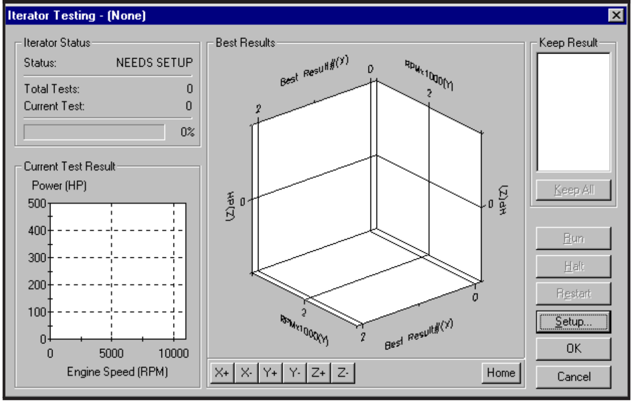
The solution to the this problem comes from the same source that made this overabundance of data possible in the first place: the computer. Sorting, comparing, cataloging, filing, retrieving, storing; these are all terms associated with computers.

Using *Iterative Testing*™, the Dyno2000 can evaluate more parts combinations than any human could sort and file. It will analyze all the results and present the best for you to review. The *Iterator* handles all testing details. In fact, you can start a test series and walk away from the computer until the task is complete.



Using *Iterative Testing™*

Empty Main Iterator Screen (Requires Setup)

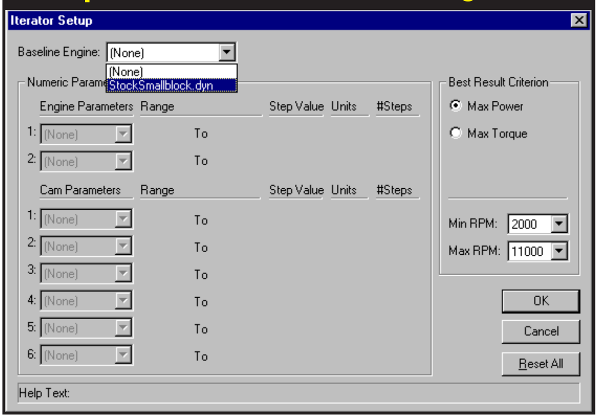


To start an iterative test, first build a baseline engine (the engine you would like to optimize). Then select *Iterator Testing* from the Tools menu or the Iterator icon in the Toolbar. The empty Main Iterator Screen is displayed (shown here). Select the Setup button to open the Iterator Setup dialog box (shown below).

In fact, computer excel at these tasks. Automating these repetitive tasks, the Dyno2000 brings not only the awesome power of engine simulation computations to the desktop, but also it offers the ability to sort, compare, and select from hundreds, thousands, even millions of dyno tests and isolate the best combinations based on your selection criterion. Using powerful, built-in *Iterative Testing™*, the Dyno2000 can test more parts combinations than any human could sort and file. It will carefully analyze all the results and present the best-of-the-best for you to review. The Dyno2000 Iterator handles all testing details. In fact, you can start a series of tests and simply walk away from the computer until the task is complete. Or you can continue to use your computer for other tasks while it performs dyno testing and analysis in the background.

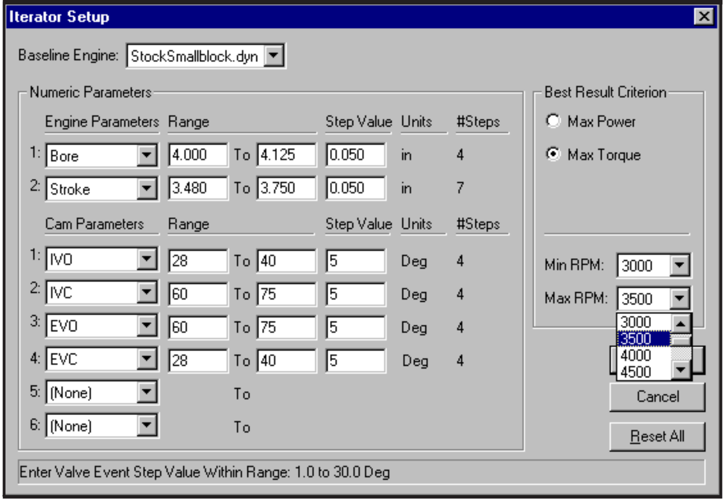
The *Iterator Setup* box allows you to choose the range of components for the testing series. Start off by selecting a Baseline engine from the *Baseline Engine* drop-down box. Every "open" engine is available for Iterative testing, providing all component categories are completed (green Status Boxes).

Setup Screen (Select Baseline Engine)



Using *Iterative Testing*™

Iterator Parameters, Step Values, Test Criterion



After selecting a baseline engine the *Numeric Parameters* menus become active. Select an *Engine Parameters* for testing. Then enter a testing *Range*, and a *Step Value*. Choose if you would like to search for peak Power or Torque in the *Best Results Criterion* box. Finally, select the Minimum and Maximum RPM.

Setting Up Iterative Testing

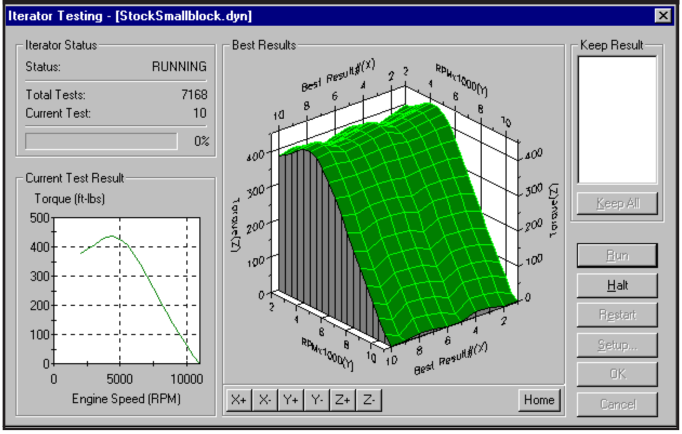
In order to perform a single dyno test all component parts must be selected (green status boxes in all component categories). An iterative test can be setup only *after* the first dyno test has been completed and the horsepower and torque are displayed in the results screen.

To start an iterative test, select the components you would like to optimize for your baseline engine, make sure all Status Boxes on each component category are green, and **Auto Calculate Valve Size** and **Valve Lift** are turned off (more on this later). Then select ***Iterator Testing*** from the **Tools** menu or select the Iterator icon in the Toolbar. The Main Iterator Screen is displayed (it will be empty the first time it's opened). Select the **Setup** button to open the **Iterator Setup** dialog choices. The Setup screen allows you select the range of components to use for Iterative testing. Start off by selecting your baseline engine from the **Baseline Engine** drop-down box. Every "open" engine that has all component categories completed is available for Iterative testing. When you have selected a baseline engine, the **Numeric Parameters** menus will become active. Select an Engine Parameter to test (bore, stroke, and compression; more parameters will be added in the next release of the Dyno2000; just send in your registration card to receive a free update!) and/or select any of the **Cam Parameter** menus.

When you select a testing parameter, range boxes are displayed and loaded with the current component value. Enter the value range for each engine parameter (always enter the smaller value in the left box and the larger value in the right box) and the **Step Value** to use throughout the test run. The smaller the Step Value, the more tests will be performed. The **Number Of Steps** (#Steps) for each parameter will be calculated and displayed to the right of each parameter field. The *total*

Using *Iterative Testing*™

Iterator Testing Begun



Begin Iterative testing by clicking the Run button. As each test is completed, the engine power or torque curve will be displayed in the small graph on the left. As testing proceeds, the ten component combinations that produce the best power or torque within the selected rpm range are "stored" in the *Best Results* 3D graph.

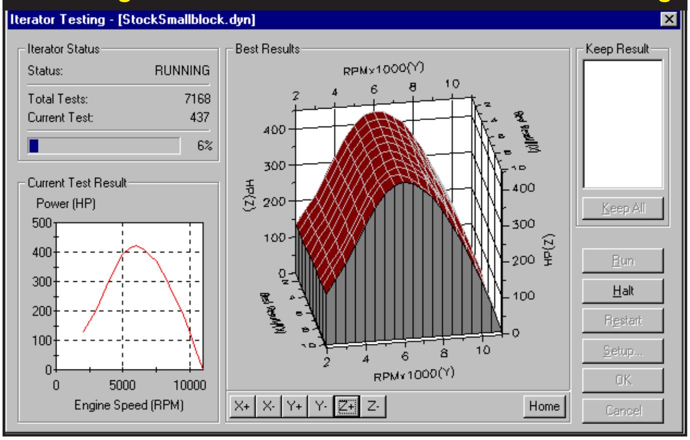
number of steps for an Iterative test can accumulate quickly, since it is determined by *multiplying together all the Number-Of-Step values*.

Cam timing values are selected from the **Cam Parameters** category. When each timing value is selected, such as IVO, the current value for that timing point is loaded into the parameter range boxes. Select the minimum and maximum values for each timing point (always enter the smaller value in the left box and the larger value in the right box), then select a step value. **#Steps** will be displayed.

When all components, ranges, and step values have been selected, choose whether you would like to search for **Max. Power** or **Max. Torque** in the **Best Results Criterion** box. Finally, select the **Minimum** and **Maximum RPM** range values that the Dyno2000 should use to search for the best horsepower or torque (rpm range must span least 500rpm). For example, you might select 2500-3000rpm in a search for maximum torque on a powerplant for heavy vehicle or towing applications. On the

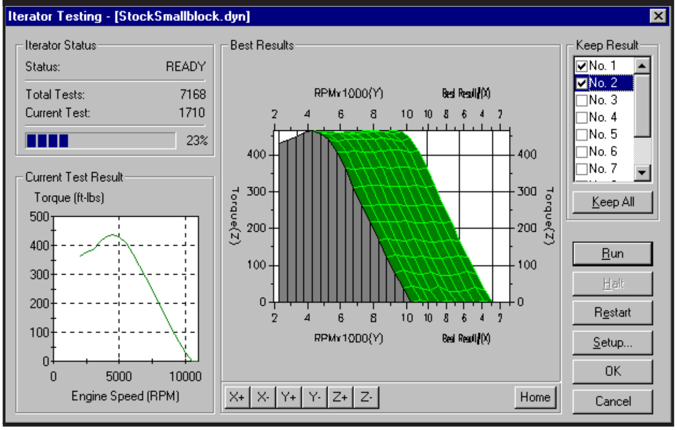
View the curves (red curves indicate Horsepower; green curves indicate Torque) from any prospective using the X+, X-, Y+, Y-, Z+, and Z- buttons (Home returns the graph to original position).

Viewing Results As The Iterator Continues Testing



Using *Iterative Testing*™

Iterator Testing Begun



At any time during the Iteration process, you can *Halt* calculation. Clicking *Run* will resume with no data loss. While the Iterator is halted, you can select up to ten curves from the *Keep Result* box. Click OK to close the Iterator. The Dyno 2000 will spawn these simulated engines.

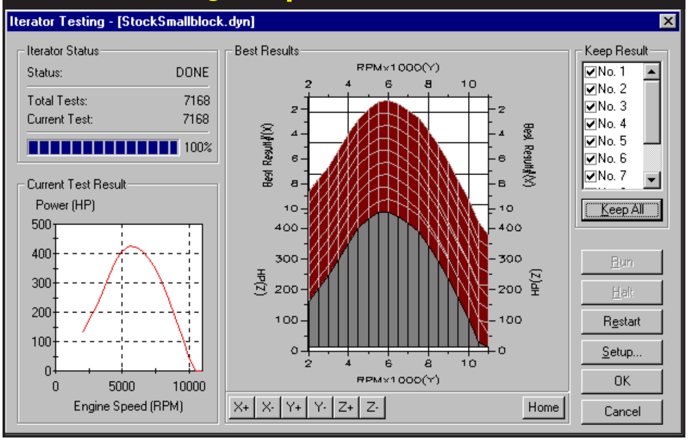
other hand, a selection of 7000-8500rpm might be used in a search for maximum horsepower on a race engine.

When you have completed all the selections on the Setup dialog, click **OK** to close the box and return to the Main Iterator Screen. Notice that the total number of tests in the current Iterator run is displayed in the **Iterator Status** box (upper left). Begin Iterative testing by clicking on the ***Run*** button. As each Iterative test is completed, the engine power or torque curve will be displayed in the small **Current Test Result** graph (red curves indicate Horsepower; green curves indicate Torque). As testing proceeds, the ten component combinations that produce the best power or torque within the selected rpm range are "stored" in the **Best Results** 3D graph.

When Iterative testing is complete (you can stop testing at any time by pressing the **Halt** button; press **Run** again to continue testing), the **Best Results** graph will contain the ten engine combinations that achieved the highest horsepower or torque

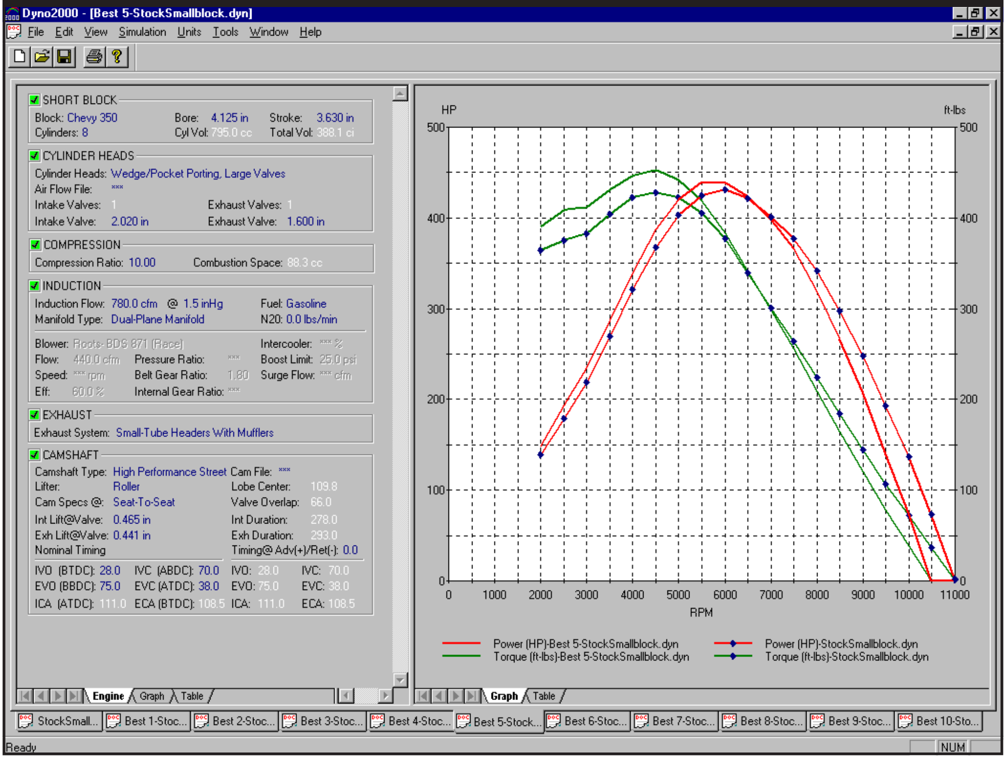
When the Iterator completes a test run (this run consisted of 7168 tests), you can keep any or all of the ten "best-results" engine configurations (click on *Keep All* to keep all ten). If you click *OK*, the Iterator will create (spawn) engine configurations that match those used to produce the best-results curves.

Iterator Testing Completed



Using *Iterative Testing*™

Spawned Engines Displayed In Engine Tabs; Iterator Power Increase



When you close the Iterator screen, new "spawned" engines will be created and displayed in the Engine Tabs at the bottom of the Main Program Screen. Each new engine can be brought into the foreground by clicking on its Selection Tab. Iterator-spawned engines can be analyzed, tested, and modified in any way, just like any other engine in the Dyno2000. The comparison test shown here between the Base-line engine and one of the ten Iterator test results illustrates the increase in power and torque that was "found" by the Iterative testing.

within the selected parameters and rpm ranges. View the curves from any prospective using the **X+**, **X-**, **Y+**, **Y-**, **Z+**, and **Z-** buttons (**Home** returns the graph to original position), then place check marks next to the curve numbers you wish to keep in the **Keep Result** box. You can keep all ten curves by clicking on **Keep All**. When all curves you wish to keep have been selected, click **OK** to close the Iterator. In a few seconds, the Dyno2000 will spawn dyno-test engines with the component combinations that produced the power or torque of the selected curves.

When the Iterator closes, the new spawned engines will be displayed in the **Engine Selection Tabs** at the bottom of the **Main Program Screen** (see page 13 for more information on Engine Tabs). Each test engine can be brought into the foreground by clicking on its Tab. Iterator-spawned engines can be analyzed, tested, and modified in any way, just like any other engine in the Dyno2000. In fact, it is possible to begin a *new* Iterator test using any of the spawned engines as a Baseline Engine to further "home in" on the desired results.

Halting And Restarting Testing

At any time during the Iteration process, you can stop calculation by click the **Halt** button. Simply clicking **Run** will resume calculation with no data loss. If, instead, you wish to reset the Iterator and start the simulation series over again, click the **Restart** button, or you can click **Setup,** change any parameters or step values you wish, then click **Run** to start the Iteration process from the beginning.

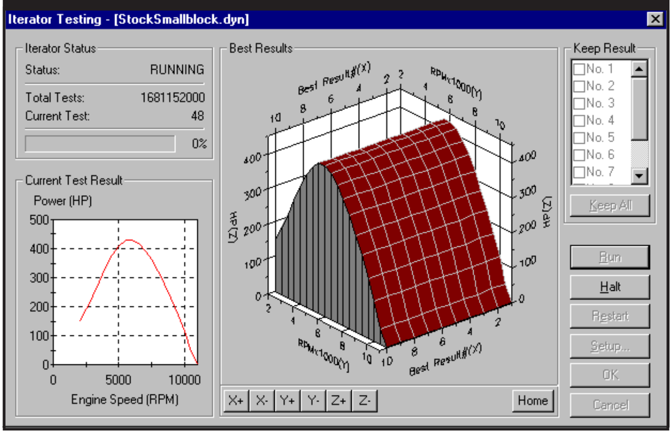
If you Halt the simulation, you can close the Iterator screen by clicking **Close**. If you reopen the Iterator screen before you close the Dyno2000, you will be able to resume calculations on the current Iterator series. However, when the Dyno2000 is closed, all Iterator calculations and results that have not been spawned to engines and saved to disk will be lost.

Tips For Running Iterative Testing

Setting up an Iterative series only takes a few seconds, however, if you include too many parameters, ranges that are too wide, or step values that are too small, you will create an Iterator series that contains too many tests. If you create a series longer than 300 million tests (even fast computer systems will require one year or more to complete 300 million tests) the Dyno2000 will request that you increase step values for selected parameters.

The best way to find optimum components, especially cam timing, is to use large step values (5 degrees or more) to "get in the ballpark" of the right values. Then run a second Iteration series on the best engine, keeping the range of values narrow (perhaps just a 5 or 10 degree range) and use smaller (perhaps 1 degree) step

Iteration Counter Limits



Setting up too many parameters, ranges that are too wide, or step values that are too small, will create a test series that contains too many runs. This experimental setup requires over 1.6 billion tests to complete (would take almost 6 years). The Dyno2000 limits a test series to no more than 300 million tests (about one year of computation time on a fast computer!).

values to precisely locate the best timing.

Narrowly-focused tests may still require several thousand test cycles to complete. A series this large may require an hour or two—or even a day or two—of calculation time depending on the speed of the computer. In these cases, you may continue to use your computer to perform other tasks. Simply use the Start menu to begin other applications and use Alt-Tab to switch between applications (see your Windows documentation for more information on program switching).

Note: If you are running Windows98, you may select the "DeskTop" icon in the task bar (usually located two or three icons to the right of the Start menu on the task bar) to "minimize" the Dyno2000 and regain your desktop.

Warning! *If you are running a simulation series and your system crashes or is turned off, all calculated test results will be lost.* One way to circumvent a loss of a long calculation series is to **Halt** the calculation from time to time, select a power curve that seems to best match your criterion, check the curve number in the **Keep Results** box, spawn the engine, then **Save** it to your hard drive. After the engine file has been saved, resume Iterative testing.

OTHER FEATURES

DYNO FILE COMPATIBILITY

DeskTop Software allows you to simulate building and dyno testing an engine, but in addition you can install any simulated engine in a simulated vehicle using the DragStrip2000, then test the combination in 1/8- or 1/4-mile drag events. You can even load simulated engines into DeskTop Pro Drag Racing and X-Car Road Racing games. It is Motion Software's goal to maintain this compatibility throughout our entire software line.

Dyno2000 engine files can be directly loaded into the DragStrip2000; no file export is required to transfer any engine into the DragStrip2000. However, the Dyno2000 has many options, like forced induction, that are not supported in older DOS-based Desktop simulations. To maintain as much backward compatibility as possible, a DOS file export feature has been incorporated into the Dyno2000 (see photo, next page). Using this export feature, you can exchange Dyno2000 engines with the DeskTop Dragstrip, the DeskTop Full Throttle Reaction Timer, and even DeskTop Pro Drag Racing and X-Car Road Racing games.

However, to maintain this compatibility, export limits must be imposed on Dyno2000 engines. The following exports limits are required:

1) Engines using **Forced Induction** cannot be exported. Switch the induction system to a naturally-aspirated manifold.

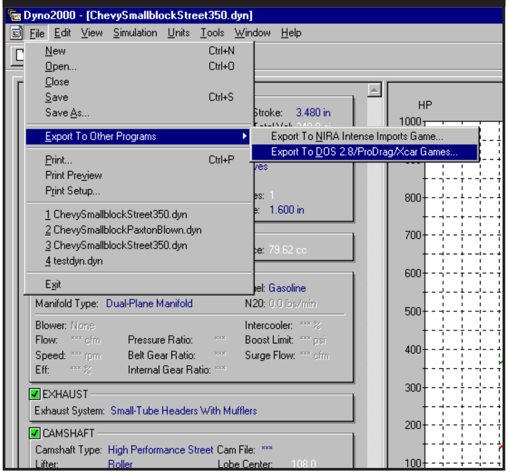
2) Engines that have **Auto Calculate Valve Size** or **Auto Calculate Valve Lift** cannot be exported until these features are turned off.

3) The **Unit** system (use the ***Units*** menu in the Dyno2000) must be set to the US measurement system before DOS file export.

Note: The Dyno2000 also provides a special export feature for users of the ***NIRA Intense Import*** drag racing game. Engines exported from the Dyno2000 using this feature have none of the above restrictions; any engine you can build in the Dyno2000 can be imported into Intense Import by using the NIRA export feature available in the File menu (see photo, next page).

Other Program Features

File Menu Export And Print Choices



The new DragStrip2000 vehicle simulation will directly read Dyno2000 engine simulation files. To support earlier software, the Dyno2000 also incorporates a *DOS File Export* feature that allows you to transfer many simulated Dyno2000 engines to the Dos-based DeskTop Dragstrip, the DeskTop Full Throttle Reaction Timer, and even Pro Drag Racing and X-Car Road Racing games. There is a special menu choice for the NIRA Intense Import racing game (no engine restrictions are required to use this export). The File menu also offers choices that will help you setup your printer and print dyno test data.

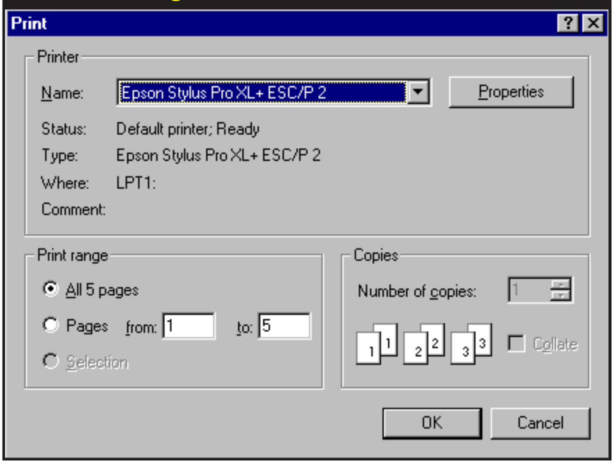
PRINTING DYNO DATA AND POWER CURVES

The Dyno2000 is capable of printing a complete list of engine components, cylinder head airflow data, exact engine test result values, and 2D graphic curves of several engine-test variables. Each of these data sets print on separate pages that comprise a complete 5-page, dyno-test report of the currently-selected engine. You can determine which pages you would like to print, preview the pages before you print, and direct the output to any installed Windows printer.

There are three choices in the File menu (located on the Main Program Screen) that will help you setup your printer and print dyno data. The choices are:

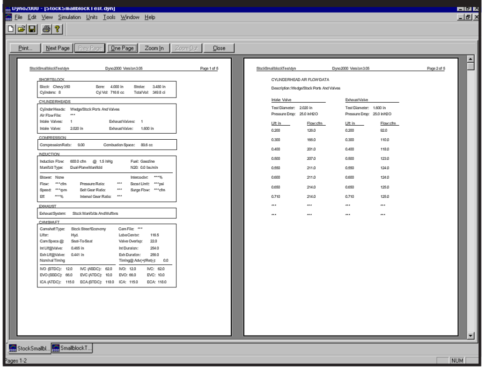
The print dialog box, accessible from the *File* menu, allows the selection of a printer, access to printer Properties, and you can enter the range of dyno-test report pages. Printing can be started from this dialog box.

Print Dialog Box

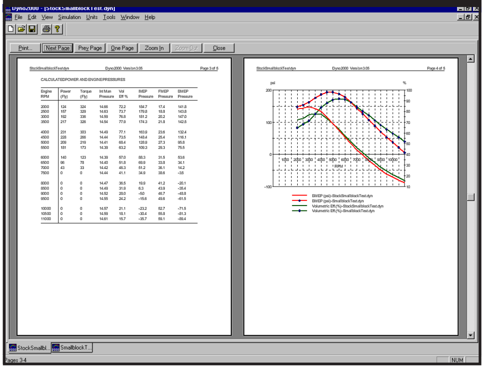


Other Program Features

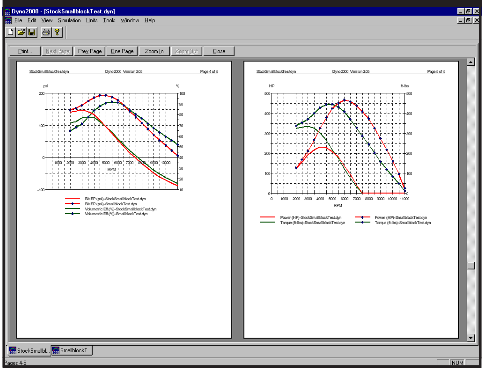
Print Preview Page 1–2



Print Preview Page 3–4



Print Preview Page 4–5



The Dyno2000 will print a complete list of engine components, cylinder head airflow data, exact engine test result values, and graphic curves of engine-test variables. Each page is shown here using the print preview function available from the *File* menu. Page one prints a complete component list. Page two displays the cylinder head airflow data. Page three shows all calculated engine power and pressures values. Page four and five reproduce both graphs (from the left and right panes of the Main Program Screen).

Print—Opens a dialog box that allows the selection of a printer, access to printer Properties, and the Print Range of dyno-test pages. Printing can be started from this dialog box.

Print Preview—Opens the Print Preview Screen that provides an on-screen rendering of what each page in the dyno-test report will look like when printed on the selected Windows printer.

Printer Setup—Similar to the Print dialog box (allows printer selection), except printing cannot be started from this box.

The dyno-test report generated by the Dyno2000 consists of 5 pages. Here is description of each page:

Page 1—This page prints the components selected for the current dyno test. The appearance of the report is similar to the Component Selection pane of the Main Program Screen.

Page 2—This page displays the cylinder head airflow data used for the test run.

Page 3—All calculated engine power and pressures are provided in chart form. A calculated value is listed for each 500rpm test point throughout the full test range (2000 to 11,000rpm).

Page 4—The first of two graphs of engine output is reproduced on this page (this is the graph that is setup on the left side of the Main Program Screen; select the **Graph Tab** at the bottom of the Component-Selection screen to display this graph). Full color printing is supported.

Page 5—The second of two graphs of engine output is reproduced on this page (this is the graph that is setup on the right side of the Main Program Screen). Full color printing is supported.

GENERAL SIMULATION ASSUMPTIONS

The Dyno2000 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 Motion Software, Inc., engine simulation software, dyno operators and engine owners readily acknowledged the possibilities of errors in horsepower measurements. Unless the dyno operator and test personnel are extremely careful to monitor and control the surrounding conditions, including calibration of the instrumentation, comparing results from one dyno cell to another (or even one test run to another) is a futile task.

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 maintained throughout the simulation process. Here are some of the assumptions within the Dyno2000 that establish a modeling baseline:

Fuel

1) The fuel is assumed to have sufficient octane to prevent detonation.
2) The air/fuel ratio is always maintained at the optimum power ratio.

Other Program Features

Environment

1) Air for induction is 68-degrees (F), dry (0% humidity), and of 29.92-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, during which the load is adjusted to “hold back” engine speed as the throttle is opened wide. The load is adjusted to allow the engine speed to rise to the first test point, 2000rpm in the case of this simulation. The engine is held at this speed 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 continues until the maximum testing speed of 11,000rpm is reached.

2) Since the testing procedure increases engine speed in 500rpm steps, and engine speed is held steady during the measurement, the measured power does not reflect losses from accelerating the rotating assembly (the effects of rotational inertia in the crank, rods, etc.). These processes affect power in most “real-world” applications, such as road racing and drag racing, where engine speed is rapidly changing throughout the race.

COMMON QUESTIONS

COMMONLY ASKED QUESTIONS

The following information may be helpful in answering questions and solving problems that you encounter when installing and using the Dyno2000. If you don't find an answer to your problem here, send in the **Mail/Fax Tech Support Form** on page 113 *(Motion Software provides Mail/Fax technical service to registered users only—mail in your registration form today).* We will review your problem and return an answer to you as soon as possible.

INSTALLATION/BASIC-OPERATION QUESTIONS

Question: Received an "Error Reading Drive D" (or another drive) message when attempting to run or install the Dyno2000. What does this mean?

Answer: This means your computer cannot read the disk in your CD-ROM drive. The disk may not be properly seated in your drive, the drive may be defective, or the disk may be damaged. If you can properly read other CD-ROM disks in your drive, but the Dyno2000 distribution disk produces error messages, try requesting a directory of a known-good disk by entering **DIR X:** or **CHKDSK X:** (where **X** is the drive letter of your CD-ROM drive) and then perform those same operations with the Dyno2000 disk. If these operations produce an error message only when using the Dyno2000 disk, the disk is defective. Return the disk to Motion Software, Inc., for a free replacement (address at bottom of Tech Support Form).

Question: Encountered "Could not locate the Dyno2000 CD-ROM disk" error message when trying to run the Dyno2000. Why?

***Answer*:** Please insert the Dyno2000 disk in your CD-ROM drive. Occasionally, the Dyno2000 will need to access the CD. Please keep the Dyno2000 disk handy.

Question: The Dyno2000 produced an *Assertion Failure* error. What should I do?

***Answer*:** Please note down all of the information presented in the message box, provide a quick synopsis of what lead up to the error, then send this information to Motion Software. Thank you for your assistance in helping us improve the Dyno2000.

Common Questions

SCREEN DISPLAY QUESTIONS

Question: I can only see a small portion of the Dyno2000 screen on my monitor, even though I have a 19-inch monitor. What can I do so that I don't have to scroll both horizontally and vertically?

***Answer*:** The screen resolution of your monitor (not its size) determines how much of the Dyno2000 you can see on screen without scrolling left and right. You can change screen resolution by RIGHT CLICKING on your desktop, then selecting PROPERTIES from the drop-down menu. Choose the SETTINGS tab and increase screen resolution by moving the **Screen Area** slider to the right. For more information about screen resolution, refer to the documentation that was supplied with your computer or your video graphics card.

BORE/STROKE/SHORTBLOCK QUESTIONS

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 Dyno2000 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 cylinder-wall 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.

COMPRESSION-RATIO QUESTIONS

Question: The Dyno2000 calculated the total Combustion Volume at 92ccs. But I know my cylinder heads have only 75ccs. What's wrong with the software?

***Answer*:** This confusion comes from assuming that the calculated **Total Combustion Volume** displayed in the component-selection screen is the same as your measured combustion-chamber volume. The Total Combustion Volume is the <u>entire volume</u> that remains in the cylinder when the piston reaches top dead center. See page 30 for more information about compression volumes.

INDUCTION/MANIFOLD/FUELS QUESTIONS

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 Dyno2000, 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, phenomena that is carburetor specific and extremely difficult to model. The Dyno2000 assumes an optimum air/fuel ratio regardless of the selected CFM rating. While the program produces positive results from larger-and-larger induction flows (by the way, the predicted power increases are close to 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: The engine I am building uses two 660-cfm Holley carburetors. How can I simulate the airflow?

***Answer*:** The Dyno2000 will simulate induction airflow from 100 to 3000cfm, rated at either standard 4-barrel pressure drop of 1.5-inches of mercury or at standard 2-barrel pressure drop of 3.0-inches of mercury (a pressure drop of 1 inch of mercury is equivalent to 13.55 inches of water). To simulate two, 660cfm, 4-barrel carburetors, simply add the airflow and enter the total 1320cfm value into the component-selection screen (for four-barrel carburetors, make sure the pressure drop shown in the INDUCTION category is 1.5-in/Hg).

Question: I am working on some custom 2-barrel carbs. My buddy has a flow bench and has tested some of my handiwork, but he used a pressure drop of 30 inches of water instead of the "standard" 3-inches of mercury. Can I convert these flow numbers to 1.5-inches of mercury so that I can test them in the Dyno2000?

***Answer*:** Yes. You can use the Airflow Math Calculator available in the Tools drop-down menu (see page 40). If you would like to understand the math behind this calculation, here are the steps you need to take: First convert the pressure drop from inches of water to inches of mercury. Use the following formula:

$$\textbf{Pressure (in/Hg)} = \frac{\textbf{Pressure (in/H}_2\textbf{O)}}{\textbf{13.55}}$$

Once you have calculated the equivalent pressure drop in inches of mercury, use the following formula to convert the flow into the standard rate for 4-barrel carburetors at 1.5-inches of mercury:

$$\textbf{New Flow (cfm @ 1.5in/Hg)} = \textbf{Old Flow (cfm)} \times \sqrt{\frac{\textbf{1.5}}{\textbf{Pressure Drop (in/Hg)}}}$$

Enter this value directly in the component-selection screen to simulate your carburetor.

Common Questions

CAMSHAFT/VALVETRAIN QUESTIONS

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 racing camshafts are used without complementary components, such as high-flow cylinder heads, high compression ratios, and exhaust system components that match the performance potential of the cam.

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 Dyno2000 uses the timing specs found on your cam card, and in cam manufacturer's catalogs, to develop a valve-motion curve (and from this curve it develops the instantaneous airflow for each port at each degree of crank rotation). 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 Dyno2000 "creates" its own seat-to-seat, valve-motion diagram for use in later calculations of power and torque. A lot can happen in induction airflow between the time the valve rests on the seat and when it reaches 0.050-inch of lifter rise. When in doubt, use seat-to-seat timing figures. They provide the Dyno2000 more information about valve motion at low lifts, and are more likely to produce accurate simulated power levels.

Question: How does the Dyno2000 allow for hydraulic, solid, and roller lifters?

***Answer*:** The Dyno2000 calculates a valve-motion diagram that is used in subsequent calculations to predict horsepower and torque (see previous answer). When the choice is made to move from hydraulic to solid, and then from solid to roller lifters, the Dyno2000 increases the valve acceleration rates to coincide with the lobe shapes that are commonly found on these cam grinds. See pages 71 for more information about lifter selection.

Question: Can I change rockerarm ratios with the Dyno2000?

***Answer*:** Yes. Simply use this formula to alter valve lift (the Dyno2000 will calculate the new valve motion throughout the lift curve):

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

When you have calculated the new valve lifts for the intake and exhaust valves, enter these numbers directly into the component-selection screen (make sure **Auto Valve Lift** is turned off).

Question: I found the published factory seat-to-seat valve timing for Pontiac engine that I am building. The IVC occurs at 112 degrees (ABDC). Something goes wrong when it enter the valve events into the Dyno2000.

***Answer*:** There are so many ways that cam specs can be described for cataloging purposes that it's confusing for anyone trying to enter timing specs into an engine simulation program. Your Pontiac is a classic example of a 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 lifter 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 Dyno2000 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 lifter-rise 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. Use the *Cam Math Calculator* built into the Dyno2000 to calculate the intake and exhaust opening and closing points. You'll need 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**.

See page 78 for more information on the *Cam Math Calculator*.

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 used in the Dyno2000 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 Dyno2000 are simply living in the "dark ages." Our suggestion is to contact another cam manufacturer and check out the Motion Software CamDisk that contains 1200+ cam files that you can instantly load and test in the Dyno2000.

Common Questions

QUESTIONS ABOUT RUNNING A SIMULATION

Question: The Dyno2000 displayed an error message "The Dyno2000 was unable to complete the simulation. A more balanced combination of components..." What went wrong?

***Answer*:** The combination of components you have selected produced a calculation error in the simulation process. This is often caused by using restrictive induction flow on large-displacement engines, using a very short stroke, or by using radical cam timing on otherwise mild engines. Try reducing the EVO timing specs, increasing the induction flow, lengthening the stroke, selecting a cam with less duration, or reducing the compression ratio. A balanced group of components should not produce this error.

Question: The Dyno2000 Iterator takes several seconds to complete one cycle of a several-thousand run test. A full series takes way too long. Is there a problem with my computer or the software?

***Answer*:** The Dyno2000 is a full 32-bit, highly optimized Windows program, however, it uses a powerful full-cycle simulation that performs millions of calculations for each point on the power curves, and this takes some time. Refer to pages 8 and 9 for more information on computation times for several computer systems.

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 Dyno2000 uses a simulation technique that iterates toward an answer—this technique is not the same as used in Iterative Testing—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.

Question: When I run a simulation, part of the horsepower and torque graph doesn't appear on my screen. What can I do to correct the display?

***Answer*:** Open the **Graph Options** menu (right-click on the graph) and select Auto Range for the **Y1** or **Y2** variable. See page 83 for more information about graph scaling and plotting variables.

Question: I specified a 30-degree range of values for IVO cam timing with 5 degree steps in the Iterator. But the number of steps displayed is zero. What's the problem?

***Answer*:** Always specify the lower (smaller) number in the left range box and the higher number in the right range box. The Iterator increments the left-box value with the step value until it reaches the right-box value.

DYNO 2000™ Advanced Engine Simulation

MINI GLOSSARY

Cam Timing @ 0.050-Lift—This method of determining camshaft valve timing is based on 0.050 inches of tappet rise 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 valve lift (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 or BDC.

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

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 improve performance. High lift rates usually require specially designed, high-strength components.

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.

Duration or **Valve Duration**—The number of crankshaft degrees (or much more rarely, camshaft degrees) of rotation through which the valve lifter or cam follower is raised above a specified height; either seat-to-seat valve duration measured at 0.006-, 0.010-inch or other valve lifts (even 0.020-inch lifter rise), or duration measured at 0.050-inch lifter rise, called 0.050-inch duration. Intake duration is a measure of all 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.

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 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 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*. The EVO timing point can be considered the second most important cam timing event from a performance standpoint.

Filling & Emptying 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.

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 Motion-PC Dyno is designed to simulate the functional processes of a four-cycle engine.

Horsepower—Torque measures how much work (an engine) *can* do; and 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 the torque arm one full revolution determines the power per revolution: Power Per Revolution = Force or 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 consideration.

Induction Airflow—The airflow rating (a measurement of restriction) of a carburetor or fuel injection system. Standard automotive 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.

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 from a performance standpoint. 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 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.

Mini Glossary

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).

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 system 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."

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.

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.

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" over which the twisting force is measured. Torque is a force *times* 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 working 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—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 (in engines equipped with rockerarms).

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—An engine measurement calculated by dividing the mass of air inducted into the cylinder between IVO and IVC 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.

Dyno Testing Notes

Dyno Testing Notes

MAIL/FAX TECH SUPPORT

Please use this form (or a copy) to obtain technical support for the Dyno2000 from Motion Software, Inc. Fill out all applicable information about your system configuration and describe your problem thoroughly. We will attempt to duplicate the problem and respond to your question as soon as possible. Mail or fax this form and any dyno-test printouts to the address below. *Note: We will only respond to problems from registered users—if you haven't already, please take a moment and fill out the registration card supplied with your Dyno2000.*

Your Phone (   ) _______-__________ Your Fax (   ) _______-___________

Your Name ____

Address ______________________________ Apt. or Building __________

City _________________________ State ______ ZipCode ____________

Email ____

Brand of computer ____________________ CPU ________ Speed ________

Size of hard drive _______ Amt of RAM _______ Video Card ______________

Running Windows95 ❑ Windows9898SE ❑ Windows2000 ❑
(If not listed here, what version ________________)

Version of Dyno2000 (see front of CDROM) __________________________

Please describe the problem you encountered with the **Dyno2000** and, if

necessary, the menu choices that caused the problem to occur ______________

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Can you duplicate the problem? __________________________________

Mail/Fax this form to:

Motion Software,Inc., 535 W. Lambert, Bldg. E, Brea, CA 92821-3911
Fax: 714-255-7956, Email: support@motionsoftware.com

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