ENGINE EXPERT

Engine Development Software

for Windows

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Calculations for Intake Port Design by Dick Phillips

Rev. 1.00

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Software Installation

Place the CD in your CD drive. If the installation program does not auto-start then run the SETUP program located on the CD.

Plug the authorization key into the appropriate port on your computer (printer port or USB port, depending upon the key). When using a parallel printer port key you can plug your printer cable into the key. It will not interfere with the operation of the printer or with other software.

If the program does not find your authorization key it will run in DEMO mode. In demo mode you can not save files and the value of some inputs is locked. The program will warn you if you try to change a locked input.

Software Updates

You can check the revision history and download the latest software update form the Audie Technology website at: www.audieteh.com/updates.htm

Introduction

Purpose and Features of Engine Expert

The Engine Expert used computer simulation to give builders of high performance engines a tool to develop and test ideas. The program helps match engine components to operating requirements for various applications and across different kinds of engines. High performance engine design and development is a jigsaw puzzle involving the weight, strength, and costs of parts; the availability of critical components; cylinder head and cam characteristics; and the parts you already have from previous projects. This program can help you identify critical items to change as you improve your engine in a deliberate, step-by-step order.

Engine Expert helps you with the following engine design decisions:

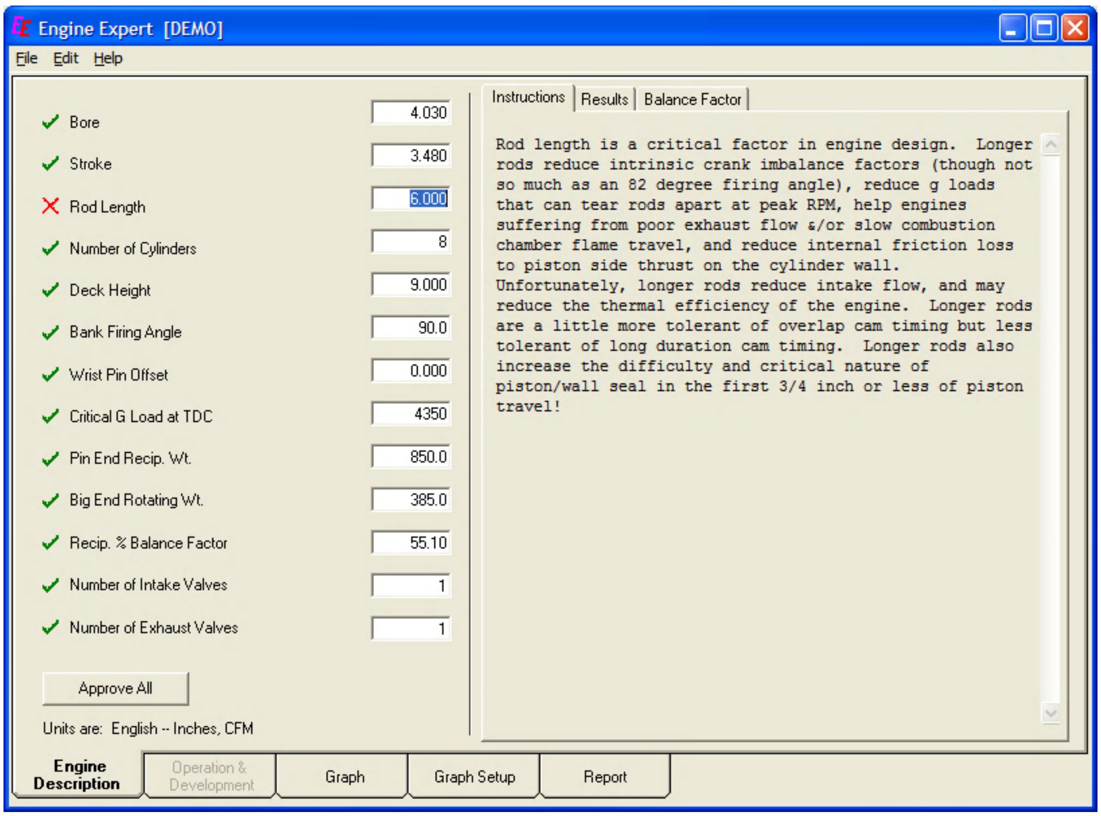
- Defining cylinder head flow requirements
- Testing trade-offs between airflow, compression ratio, engine speed, and volumetric efficiency
- Finding performance limits defined by cam and cylinder head flow
- Finding engine speed limits imposed by reciprocating or rotating parts failure
- Selecting the most cost-effective parts
- Selecting crankshaft balance factors
- Calculating machine shop specifications
- Selecting Cam, Carburetor, Intake, and Exhaust system specifications

Although the Engine Expert shows an estimated torque and estimated power, no program can substitute for a dyno, and no dyno can substitute for actual competition success. The success you achieve depends on how well your engine is built, and how well the cam, valve train, intake system, exhaust system, and ports work together in your engine. You may get 5% more power than the Engine Expert estimates, or you could get a lot less. The torque and power estimates – and the engine design files included with the program – show levels that top engine builders generally achieve. However, if you want to see what a 3 inch rod in a 6 inch stroke engine will produce with an 8 inch bore, the program will try to do it, even though you will have to build this one in the twilight zone.

You can use Engine Expert with flow bench data and camshaft data to select cams best suited for each particular engine and application. You can also use Engine Expert to define the port flow and velocity your engine design requires.

Program Organization and Operation

Five tabs across the bottom of the program define its major divisions. The first two tabs, *Engine Description* and *Operation and Development* contain the main data entry portion of the program. They are designed to be used in sequence. Thus each entry on the first tab must be approved before you can gain access to the second tab. An icon next to each item indicates its approval status. Click on the icon to toggle the approval status. For some items Engine Expert will suggest appropriate values based upon your other entries. These suggestions appear in red just below the item. Suggested items are marked with a yellow "?" for the approval icon. Regardless of approval status Engine Expert recalculates the output results anytime that you change an entry.

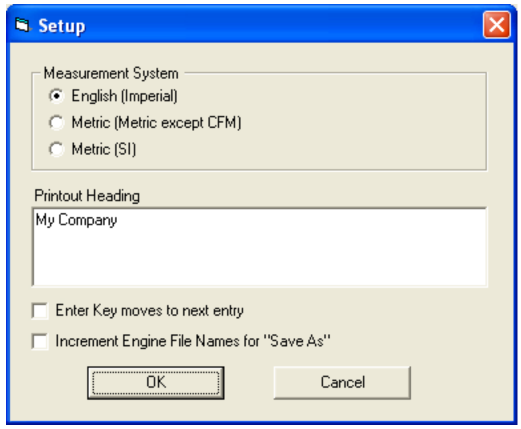


The right side of each data entry tab contains a top tab that lets you select appropriate instructions for whatever entry you are making, or output data. Output data is also available on the *Graph* tab and the *Report* tab. Use the *Graph Setup* tab to configure the graph. Use the *Report Setup* command on the *Edit* menu to configure the report.

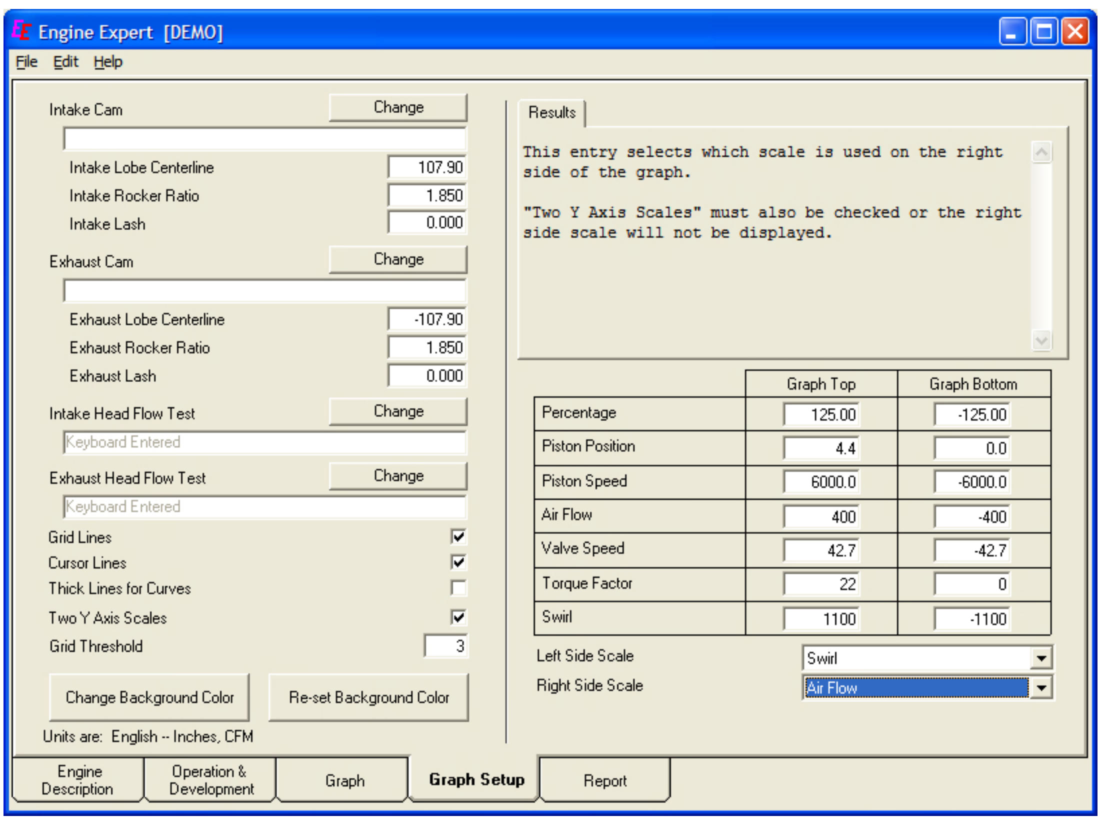
The current status of program including all input data, graph configuration, report configuration, and miscellaneous other settings can be saved in an Engine file. Use the Open, Save, and Save As commands on the File menu to work with Engine files.

The Graph and the Report can both be printed or saved to a file. The Report uses an ASCII test file format. The Graph uses a bmp file format.

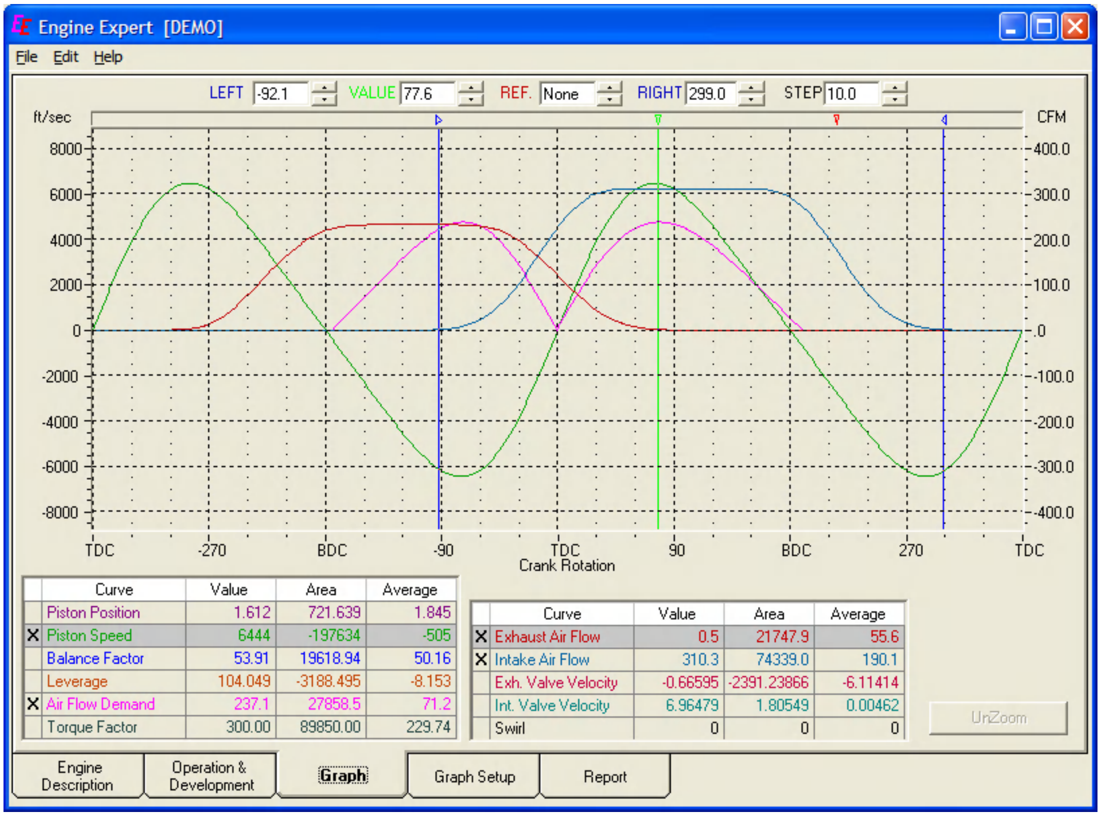
Engine Expert can work with metric or imperial units. To change the measurement system, select Setup from the Edit menu. Internally Engine Expert uses imperial units. The conversion to metric units is done each time data is entered or displayed. Thus it is possible to enter data in imperial units, change the measurement system and view the results in metric units (or visa-versa).



Air flow data from flowbench tests and camshaft profile data is used to generate the Exhaust Air Flow, Intake Air Flow, Swirl, Exhaust Valve Velocity, and Intake Valve Velocity graph curves. Use entries on the Graph Setup tab to enter this data. Camshaft data can come from Cam Pro, Cam Pro Plus, S96, Cam Doctor C1, or Doctor Doctor C1 files. Air flow data can come from Flow Pro (DOS or Windows versions) and hand entered data.



Using the Graph



Change column width in the legend

To change the width of a column in the legend below the graph, click and drag the column's right edge in the title line.

Tag/Untag

The Tag turns items on for graphing. Double click an item to tag/untag it. Pressing **SPACEBAR** will also tag/untag the highlighted item. To tag/untag all lines in the list, double click the column title for the tag mark column .

Scales

Entries on the Graph Setup tab control the graph's vertical scale. The horizontal scale is fixed at 720 degrees of crankshaft rotation to match a complete cycle of a four stroke engine. The middle of the graph is the TDC associated with overlap.

Marker Lines

The graph has 4 vertical lines which are used to select angular positions and boundaries. Two blue lines mark the boundaries for the area under the curve and average columns in the list below the graph. A green line marks the position for the value column in the list below the graph.

The red line, when activated, marks a reference so that the green line can measure relative values. To activate/de-activate the red reference line right click its marker.

Each line has a marker in the short window just above the graph. You may move a line by dragging its marker with the mouse. Each line also has a numeric entry above the graph. This entry shows its exact location and can be used to set the location.

The lines can also be moved by these keystrokes:

KEYSTROKE	***ACTION***
_ (underscore)	Move green value line to right
+	Move green value line to left
[	Move blue right boundary line to left
]	Move blue right boundary line to right
{	Move blue left boundary line to left
}	Move blue left boundary line to right
L	Move both blue boundary lines to left
R	Move both blue boundary lines to right
Up Arrow	If a numeric entry has the focus this has the same effect as clicking the up arrow next to the numeric entry.
Down Arrow	If a numeric entry has the focus this has the same effect as clicking the up arrow next to the numeric entry.

The Step entry above the graph sets the amount of movement caused by a keystroke or a click on the arrows next to the other numeric entries.

Zoom

To zoom in on part of the graph, click at one corner of the area to be enlarged. Then, while holding the mouse button down, move to the opposite corner and release the mouse button. To remove the zoom click the UnZoom button.

Your Inputs

Long Rods/Short Rods

As you increase rod length, keeping other specifications constant, the following changes occur:

- The piston reverses direction across TDC more slowly, reducing peak acceleration g loads on pistons, pins, rods, bearings, and the crankshaft.
- The piston accelerates away from TDC more slowly, over a longer distance, to a slower peak velocity–delaying ring flutter to a higher engine speed.
- Maximum intake flow demand is reduced, raising the engine speed at which Peak Power is limited by cylinder head flow capacity, and/or by valve lift. Also, maximum intake flow demand occurs farther after TDC, which reduces the intake valve opening rate required.
- Cylinder pressure is translated into crankshaft torque over a longer period of crank rotation–thus requiring a later opening exhaust to maintain the same torque as a shorter rod.
- Piston side load against the cylinder wall is reduced.
- The piston reverses direction across BDC more rapidly. (However, compression g loads across BDC remain much less than tension g loads across TDC.) Thus, pressure to push exhaust from the cylinder out the port drops faster after exhaust valve opening. Therefore, the cylinder head needs greater low-lift exhaust flow to let exhaust gas escape more easily. Because longer rod engines also prefer later exhaust opening exhaust, especially good exhaust flow is required.

Short rod behavior is noticeable for rod ratios less than 1.55, while long rod behavior is more pronounced for rod ratios greater than 1.75.

CONNECTING ROD LENGTHS (inches)				
Engine	**Length**		**Engine**	**Length**
Buick V-6	6.500		1.6L "Kent" Formula Ford	4.928
4.3L V-6/265 - 400 SB Chev	5.700		Ford V-6 SVO/Essex	6.100
427 - 454 BB Chev	6.135		221 - 289 & 302 Boss Ford	5.155
318 - 360 Mopar "LA"	6.123		302 Ford	5.090
361 - 400 Mopar "B"	6.358		351W/351 SVO Ford	5.954
413 - 440 Mopar "RB"	6.768		351C/351 Boss Ford	5.780
Mopar 426 Street Hemi	6.860		351M/400M Ford	6.580
Mopar 426 NASCAR Hemi's	7.350		352/360 FE Ford	6.540
"	7.061		390 - 428 FE Ford	6.488
"	7.174		429 - 460 Ford	6.605
Mopar 396 Hemi	6.960		429 Boss "S" Ford	6.549

Critical engine speed is higher in a long rod engine, and you can get increased Power with greater safety, albeit with less torque normally. Long rods are most suitable for cylinder heads with limited intake flow capability and high exhaust flow (75% of intake flow, or greater) such as the SB "Mouse" Chevrolet.

In contrast, for engines not limited by maximum intake flow (such as the Hi-Perf "Rat" Chevrolet) where maximum engine speed & power, cylinder wall loads, and piston velocity are NOT major design limits, shorter rods give you greater Peak Torque and Power. Overlap cam timing can be the most significant factor in cam selection for big, short rod engines. Such engines develop intake reversion or excessive exhaust scavenging problems more frequently, and prefer cams with wider lobe separation and faster opening intakes.

For more on the air flow/cam implications of long and short rods; see the section, "Optimum Tuning of Intake and Exhaust & Optimum Cam Selection"

Deck Height

Deck Height (DH) is the distance from the center of the crankshaft to the top of the cylinder block. DH limits the combination of stroke, rod length, and piston that will fit in your engine. The program uses DH to compute piston Compression Distance (the distance from the Wrist Pin Center Line (C/L) to the Block Deck). You can use the Compression Distance to find off-the-shelf pistons that fit your engine, or to specify your requirements to a custom piston manufacturer. Remember, a piston must include the ring lands, plus 1/2 the pin diameter, and rod clearance under the piston deck in the Compression Distance dimension.

If you do not know the DH for your engine, don't get too worried. The program does not require this value to run. Of course, you may get some strange Piston Pin to Deck dimensions, but those are the only values affected. Common DH dimensions are from 8 to 11 inches. Some specific values are:

ENGINE DECK HEIGHTS	
Engine	**Deck Height (inches)**
Buick V-6, stock and Stage I	9.350
Buick V-6, Stage II Race block	9.530
V-6 4.3L/265 – 400 SB Chev	9.025
427 - 454 BB Chev	9.800
366 - 454 BB Chev Truck, Raised Block	10.200
273 - 360 Mopar "LA"	9.600
361 - 400 Mopar "B"	9.980
413 - 440 Mopar "RB" & "Hemi"	10.725
1.6L "Kent" Formula Ford, approx	8.1
Ford V-6 SVO/Essex	9.234

ENGINE DECK HEIGHTS (continued)	
Engine	**Deck Height (inches)**
221 - 302 Ford	8.206
351w/351 SVO Ford	9.480 / 9.503
351c/351 Boss Ford	9.206
352, 390, 427, 428 Ford	10.170
351m - 400m Ford	10.297
429 - 460 Ford	10.300 (10.322 1972 & later)
429 SVO Alum A93	11.200

Pin End Reciprocating Weight

OBSERVED RECIPROCATING WEIGHTS	
Engine	**Recip Wt (gms)**
Buick V-6, approx	825
Ford V-6 SVO/Essex, approx	920
Drag racer with long Alum rods, 4 inch bore	660
Light stock rod, 4 inch bore flat top piston	770
Light stock rod, 4.125 bore with light pistons	870
Bracket racer, 4 inch bore	800
Short track, 4 inch bore, high CR domed piston	805
Winston Cup Nascar, 4 inch bore, over 12.5 CR	950
Stock 350 - 400 cid small block	985
Stock 4.125 inch (400 cid) pistons	1100
11 to 12 CR, 4.251 big block (454/427) pistons	1100
Mopars with 4.32+ inch bores	1300

The program uses cylinder bore diameter to compute the default Pin End Weight (Recip Weight). This estimate is for ONE rod assembly only, and is based on experience with forged, endurance pistons on steel rods. Pin End Weight includes the pin end of the rod, the piston, rings, pin, and pin retainers. Your engine balance shop can weigh the pin end of your rods for you, and can weigh the other parts as well if you wish. Or, you can weigh everything, except the rod pin end, with a triple beam or electronic scale. The actual Pin End Reciprocating Weight of your engine can be +/- 20% of the program estimate. The following factors reduce Pin End Reciprocating Weight:

- Aluminum rods (but they experience metal fatigue under long term use and abuse)
- Specialty racing pistons

- Short piston height, with the pin placed close to the piston deck
- Flat top pistons
- Thin wall, tool steel pins

Critical g Load, Ring Flutter, Critical RPM, and Con Rods

The weakest components in most, though not all, high-performance engines are the connecting rods and the parts attached to them. (Some other sources of failure are inadequate oiling systems, detonation, and valve train breakage. However, the program does not deal with these items at this time.) That is, the likelihood of your engine breaking at high RPM because of a weak part is usually directly related to the reciprocating loads on the connecting rods. In a naturally aspirated engine, the biggest of these loads occur as the piston stops and reverses direction at TDC overlap.

Our experience indicates that well built engines, with balanced and polished high-performance steel rods, upgraded rod bolts (7/16 inch aircraft steel capscrews for example), forged high performance pistons, high performance cranks (high nodular iron or steel), and 4- bolt cylinder blocks fail only when the peak acceleration across TDC approaches 4,350 g's. The Engine Expert uses this guideline to help you find critical engine speed levels in your designs. HOWEVER, if your engine has ANY weaker components than are typical in a high- performance American V-8, the 4,350 g value will be too high, and you must apply whatever experience and manufacturers' data you can obtain for your particular engine! The acceleration g Load Limit is not perfect, but is more consistent for large and small engines than pounds force (Lbf), and does not change with piston weight. Furthermore, the g Load limit across TDC compression measures the likelihood of ring flutter occurring. However, the strength of rods, bolts, etc. is measured in Lbf, so the program computes Lbf as well. These values appear in the Results tab of Operation and Development

RECIPROCATING LOADS, GUIDELINES		
Engine Type	**g Load**	**Lbf**
Short-Lived drag motor (500 cid pro stock)	5,300+	13,000+
Endurance race motor with high performance rods	4,350	8,000 – 10,500
High-performance, original equipment	3,800	less than 7,000
Street stock parts	3,200	less than 4,500
Engines known to contain weak parts	2,800	3,500-

The program assumes that the components you have, with a Pin End Weight equal to the default weight estimate, will fail at whatever Critical g Load limit you enter (4,350 g's default). If you use rods, pistons, and pins lighter than the computed default weights, and IF the lighter parts are strong enough, they will increase your engine speed limit. Conversely, if your reciprocating parts weigh more than predicted (such as pistons with solid domes, etc.) the critical engine speed will be less than the program shows! When you enter a Pin End Weight heavier or lighter than the default, the program will lower or raise the default engine speed limits accordingly. If you need more information, see the section on pin end weight.

The program uses the engine specifications you entered, including the Critical g Load limit and the Pin End Weight, to suggest the engine speeds at Peak Torque and Peak Power. The suggested engine speed at Peak Power is 80% of the critical engine speed, minus 500 RPM; and the suggested engine speed at Peak Torque is 76% lower than at Peak Power. If you entered a light pin end weight, the g loads shown at peak power RPM may exceed the Critical g Load limit. This is OK, because the loads (Lbf) created by the lighter components are no greater than the heavier parts would have caused.

Acceleration g's across TDC at compression/ignition try to throw the compression rings off the bottom ring lands, in spite of compression pressure on top of them. Ring flutter bypasses combustion pressure behind the rings, and unloads gas pressure on top of them in the most critical phase of the power stroke !!! When designing very short stroke, ultra high speed engines such as the Lamborghini Formula 1, piston ring flutter can be the limit to engine speed, rather than the Lbf strength of components. The lighter (thinner) the ring(s) and the greater the Compression Ratio (pushing the ring(s) against the bottom ring land) the higher g's can be tolerated.

Piston ring flutter is delayed using narrow, light weight motorcycle engine type compression ring(s), with high compression, and/or gas ports in the pistons. Nevertheless, at some Critical g Load limit – which varies with ring thickness, Compression Ratio, and ring type – flutter will occur! You can enter a value for Critical g Load at which your type of rings will flutter, and then make sure that your engine speed stays under this limit, or else you must change your piston/ring design.

How to Estimate the Critical Load for your engine

The Critical g Load is based on the engine speed at which components fail, and you should want to get under that point. How far under is up to you. An endurance engine runs long and continuously, and so must be designed to run at lesser loads than a drag race engine.

The quickest way to find your safe load limit is to run the program with the RPM and Recip Weight at which a similar engine broke. Engine Expert will report the pounds load at which the engine broke, and you can redesign your engine to run at 80%–or less–of that value. This method will require some trial and error as you run each redesign alternative to find the safe load point.

The elegant way is to calculate the Critical g Load so that the program can automatically set RPM limits for alternative designs. To do this, set the Peak Power RPM at the break point, and find the Tension g Load at TDC for that engine speed. Next, divide the actual reciprocating weight for the engine that broke by the default reciprocating weight (calculated by the program for your cylinder bore size). Multiply the result by the Tension g Load you got from the program, and you have the new Critical g Load for your engine designs.

Example:

```
Actual Reciprocating Weight in broken engine = 894 gms
Reciprocating Weight default from program    = 945 gms
Engine Death Rattle at 8350 RPM    =           5012 g's
                                               (Tension from program)
Your Critical g Load would be:  (894 / 945) x 5012 = 4741.5 g's
```

When you enter 4741.5 g's, instead of the default Critical g Load, you will see the effect of lighter parts, longer rods, shorter stroke, etc. on the critical RPM for your particular engine. You will also be able to set your peak power RPM as near or as far from the critical RPM as you dare/care. Over 4,000 g at Peak Horsepower RPM, or in endurance racing engines, you should plan on using a 4-bolt or other high-strength cylinder block.

The program helps you upgrade your engine design with stronger rods. All you need to do is enter a higher Critical g Load or adjust the RPM until the pounds load is within the safe strength range of the stronger rods. You may be able to get strength information directly from the rod manufacturer, or you could measure the strength of your rods directly. You need to know the weight on the rod which will either begin to stretch it permanently or will stretch (egg-shape) the big end, while loaded, 0.0003 to 0.0005. To measure this, you have to mount a rod so you can suspend weight from it and find measuring pads on it. You should measure big end distortion under load and total length before and after loading. A big block rod will require 8,000 to over 10,000 pounds of tension, so an hydraulic press rigged to pull, instead of push, helps a lot. This solution is sort of extreme, but less so than breaking an engine after YOU build it. Let someone else blow up, then find out where his tattle-tale tach is stuck! However you get the information, reduce the value obtained by 10% to allow for variability between parts, and divide it (pounds by pounds, or grams by grams) by the default pin end weight estimated in the program. The result will be the Critical g Load to use for rods. (Of course, if some other component gets weak before the rods do, or detonation occurs, you will still blow up!) If the rod manufacturer can tell you the critical or "safe" engine speed for the rods with a given pin end weight, rod length, and stroke, use the program to find the Critical g Load as described above.

The program computes the following RPM limits for component failure:

Stroke	Rod Length	Recip Wt	4,350 g's RPM	4,120 g's RPM
3.75	5.565	1100 gms	7,815	
3.75	5.700	1100 (Default)	7,839	
3.75	5.700	1000	8,221	8,000
3.48	5.700	800	9,628	9,111
3.25	5.700	800	10,041	9,502
3.25	5.700	1040 (Default)	8,333	

When you don't have much information on the strength of rotating & reciprocating parts, use 3,500 to 4,000 g for the Critical Load limit–or 2,800 g if the parts are old, skinny, and cast or aluminum. Over 4,000 g at Peak Horsepower RPM, and in endurance racing engines, you should plan on using a 4-bolt or other high-strength cylinder block.

Balance Factors

The program tells you the average balance factor, and reports ideal balance factors at each degree of crankshaft rotation. These balance factors are shown as the percent (%) of total pin end weight which the balance shop would add to the total big end weight, when calculating the bob weight to place on each crank throw. The big end weight is a constant factor, because it simply rotates with the crankshaft. However the counterbalance for the pin end weight varies continuously with crankshaft rotation.

To see an example of this, run the program using the default 350 Chevy motor, but make it an in-line (Bank Firing Angle = 0) four cylinder, and view the balance factor results. Then try the Chevy as a V-8, first with the 90 degree bank firing angle, then at 82 degrees. For a surprise, try the 108 degree bank firing angle of the 229 cid V-6! At the cost of increasing the vertical rocking couple across the main bearings, GM uses a balance factor that reduces the external vertical bouncing of the complete engine. The vertical bounce is replaced with a larger, more easily damped, less noticeable, horizontal shake–and the torque impulses at low engine speed are more even. However, for high performance, Chevy finds less vibration and better durability with a 90 degree firing angle and 46% to 50% balance factor!

For an in-line engine, the pin end weight is the total weight of reciprocating masses at the TOP (pin) end of one connecting rod. That is, the weight of the upper end of the rod, plus the piston, pin, rings, and pin retainers. This weight would require no counterbalance at TDC nor BDC. However, at 82 degrees before and after TDC, about 102% of the pin end weight is required for perfect balance. Since we can't install a counter-balance smart enough to change its mass every degree of crank rotation (except in the Twilight Zone), we just use an average pin end counter-balance, which is computed by the program, and normally varies between 49% and 51% of the pin end weight. Of course, the counterbalance required for the rod big end (including rod bearing, rod bolts & nuts, and oil in the crank) is constant. So, the balance shop uses bob weights equal to the total rod big end weight, plus the 49% to 51% pin end average counter-balance.

In V type engines, where 2 rods and pistons connect to each journal, the combined weight of BOTH rod assemblies is treated as one. This program works only with engines in which rods share a single crankpin. Thus, the % balance factor computed by the program represents the % of the COMBINED pin end weight for both pistons, etc. added together. The balance shop will use bob weights equal to the average % counterbalance for the combined pin end weights, plus the weight of both big ends.

Remember, however, that the program expects you to specify the pin end weight of ONE rod assembly only. Do not try to enter or override Pin End Weight values with more or less than the weight of one rod upper end assembly.

In any piston engine, there is a great deal of variability in pin end weight required for perfect counterbalance as the crankshaft turns. This variability shows up as crank flex across the main bearings, and can be reduced by longer rods and lighter pistons. Firing angles of 80 to 90 degrees between paired rods also will reduce crank flexure across main bearings. If you are designing a fully counterweighted crankshaft, you can use the balance factors for each degree of crank rotation to find counterweights which minimize flexure across the main bearings. An engine counterbalanced this way will probably vibrate more than one balanced around 50%, but it will be easier on its crank and main bearings!

Peak Intake Velocity

Peak Intake Velocity is the velocity at the entrance to the intake tract at peak torque RPM. It is used to compute the intake area required at peak torque engine speed. 365 ft/sec is suitable for a high performance engine, and up to 450 ft/sec is OK for a double overhead cam engine. At less than 250 ft/sec, air doesn't have enough kinetic energy of motion for good volumetric efficiency, and so torque drops rapidly.

Peak Intake Velocities over 55% of the speed of sound cause pumping losses, which reduce power. That is, when an engine moves intake air at more than 55% of the speed of sound, the power used to pump the air exceeds the power released when the intake charge burns. The Engine Expert computes the 55% critical velocity and uses that velocity at Max Power engine speed to calculate the minimum Critical Intake Area. Since air temperature changes the speed of sound, the Intake Peak Velocity at Critical Area and the cross sectional size of the critical area are affected by changes you make in the Intake Air Temperature at Critical .

Volumetric Efficiency

The Engine Expert uses Volumetric Efficiency in a unique way. The value you should enter depends on whether you run a blower / supercharger and/or your success in tuning the engine. This program does NOT use Volumetric Efficiency as defined in classic engineering textbooks which don't deal with internal volumetric efficiency, nor as measured by a flowmeter on a dyno (which includes intake air bypassed directly out the exhaust during TDC overlap).

Don't change VE as a design variable, except to simulate alternative blowers. Instead hold VE constant for the type of engine you are designing, while you change other specifications and look for improvements in the peak torque and/or peak power. You should enter a value for VE that measures how well you reduce pumping losses and increase ram tuning in engines you build. When in doubt, pick an appropriate value from the following table.

You can think of VE as the ratio of the volume of air taken into each cylinder, divided by the cylinder volume, divided by Internal Volumetric Efficiency. For a description of Internal Volumetric Efficiency see the section "PROGRAM OUTPUT – Flow Demand". In a high performance engine, VE will be greater than 100% because intake air pushes into the cylinder after the peak demand point and continues into the cylinder after BDC.

Mathematically, VE is the ratio of air flow volume (CF) fed into the engine by the intake system (which may be acoustically tuned or blown / supercharged); minus intake air lost out the exhaust across TDC overlap (CFLOS); to the displacement volume of the engine (CID) divided by Internal Volumetric Efficiency (IVE). Fortunately, intake loss across TDC overlap is small when the cam timing is correct.

The formula for this is:

```
VE = (((CF - CFLOS) x 3456) / CID) / IVE) x 100
```

However, you don't need this equation to use the program! A typical street engine will show about 85%, a good race engine will show about 110%, and an exceptional race engine will achieve 120%.

Table of Suggested VE's

Typical street stock engine	85%
500 + inch Pro-Stocks where flow demand exceeds practical port/valve size limits	90 - 100%
High performance street engine with mufflers	95 - 105%
Single carburetor and/or high rise manifold	105 - 110%
NASCAR or Super Stock single carburetor engine	110%
Fuel injection/Webers/Tunnel Ram/Tuned intakes	115 - 120%
Pure race (Cosworth, Motorcycle, etc.)	120 - 125%

Every racing engine we simulated that computes a recommended intake manifold area larger than the actual valve diameter has a power band narrower than the default engine speed ratio and a combined VE of less than 125%. We have observed that the ratio of actual valve area to recommended port area, times 125% calculates the actual combined VE quite well, for otherwise unrestricted engines.

Example: A 330 cid Pro Stock engine with a 2.050 intake valve

```
               Area of 2.050 valve = 3.300 sq. in.
  Recommended intake manifold area = 3.580 sq. in.
                     3.300 / 3.580 = 0.9218
                      0.9128 x 125 = 115.2% combined VE
```

However, do not be misled into using ports much larger than recommended.

Compression Ratio, Altitude, and Blowers

To calculate the ratios used in the following discussions, you should understand the difference between 2 ways of measuring pressures—gauge pressure (psig) and absolute pressure (psia). Gauge pressure is the actual number you read from a measuring device such as a tire pressure gauge, and to be nit-picking precise you should refer to it as "pounds per square inch, gauge" or psig. However, we live at the bottom of an ocean of air. The weight of the air in the "ocean" above us is the atmospheric pressure around us. At sea level, the standard atmospheric pressure is 14.7 pounds per square inch, absolute (psia). Thus, at sea level a tire measuring 30 psig pressure contains 30 pounds per square inch pressure MORE than the 14.7 psia atmospheric background–or 30 + 14.7 = 44.7 psia. In Denver, there is less air pressing on us, and the average atmospheric pressure is 12 psia. Thus the same 30 psig tire in Denver contains air at 42 psia.

Compression Ratio and Altitude

The thermal efficiency of an engine is largely determined by Compression Ratio (CR). Higher CR makes more torque from the fuel burned. Ultimately, limitations of fuel quality, combustion chamber design, and internal heat/friction buildup cause detonation when the engine's temperature limit is exceeded. The temperature limit is different for each engine, combustion chamber, and fuel. Notice that the limit is not on mechanical CR; which the engine builder can control; but on temperature; which depends on altitude, VE, cam timing, and to a lesser extent intake system restriction. The Engine Expert uses a default CR of 12.8 to 1 simply because this value is common in racing engines using racing gas and long duration cams. Use the best experience you can find, to get the highest safe Compression Ratio for your altitude, fuel, cam, and cylinder head combination.

Torque and power estimates computed by the Engine Expert are largely based on dynamometer experience, and are corrected to sea level standards (29.92 inches Hg, 60 degrees F, and 0 % humidity).

You can adjust CR's you use with the Engine Expert to sea level standard pressure or to the altitude at which you operate, if you have a calculator which will raise a number to a power (not just do squares or square roots). Without a calculator, you can still correct to and from sea level standard pressure–14.7 psia as follows:

```
        Sea level CR = Engine Expert CR/1.132
    Engine Expert CR = Sea level CR x 1.132
```

Examples for Engine Expert CR of 13.85 and sea level CR of 11.89:

```
        Sea level CR = 13.85/1.132 = 12.23
    Engine Expert CR = 11.89 x 1.132 = 13.47
```

Other conversions require you to know the average atmospheric pressure for the lowest altitude at which you will operate your engine. In the following examples, the A^B means A raised to the power of B.

```
    Sea Level CR Correction Factor = (14.7/Your atmospheric pressure)^0.7143
Engine Expert CR Correction Factor = (Your atmospheric pressure/12.35)^0.7143
```

Example, where your atmospheric pressure is 13.2 psia, and your engine has 12.8 CR:

```
    Sea Level CR Correction Factor = (14.7/13.20)^0.7143
                                   = (.8980)^0.7143
                                   = 1.080
Engine Expert CR Correction factor = (13.2/12.35)^0.7143
                                   = (.8401)^0.7143
                                   = 1.047
      Your equivalent sea level CR = 12.8/1.080
                                   = 11.85
```

```
Engine Expert equivalent CR = 12.8 x 1.047
                            = 13.41
```

At over 14 to 1, up to somewhere around 17.5 to 1, sea level equivalent compression ratio (CR), extremely high pressure and temperature combustion gases raise engine friction enormously. Increased internal friction at such high CR's (including ring drag, which is dependent on piston speed and peak cylinder pressure times ring thickness) cancel torque increases which would otherwise occur with increased compression. So, don't get carried away with lots of compression. Above 15 to 1, even on alcohol, you probably won't find much more power on the dyno, but you will find steadily increasing temperature over 11 to 1 CR!

Reduced VE from intake restriction explains, and provides a guideline to the calculation of, the extreme compression ratios found in many Pro Stock engines. The Effective Compression Ratio (see the section including Effective CR elsewhere in this Manual) can be as high as 16.9:1 in recent endurance race engines, with Buick heads. Reduced VE reduces Effective CR, so mechanical Compression Ratio (CR) can be raised as high as 17:1 as long as Effective CR is less than 17:1. You MUST use the highest quality racing gasoline, or alcohol, and hold your breath when flirting with CR's this high! In this range, a small error in jetting, spark advance, &/or combustion chamber design will lead to detonation and break every part you own!

You will find that the Nominal Torque Sum peaks (very gently) at higher engine speed with increasing CR. You should use enough CR to put the Max Torque engine speed within the Nominal Torque Sum peak. This peak is only a few tenths high, and extends for more than a thousand RPM, so don't spend a lot of time trying to find the absolute "best" CR for your power band. Just make sure you have enough CR to operate at your intended engine speed.

Superchargers and Positive Displacement Blowers

At this time, the Engine Expert does not simulate turbocharged engines very well. If there is enough demand, we will develop the models required. Send us a letter if you need a turbo model, and tell us your engine specification and/or design problems.

Basically, blowers trade compression ratio for increased intake charge mass. Blown (or supercharged) engines require little change to the intake system (a cubic foot of intake remains a cubic foot, even if it is a whole lot denser); but, they have to get rid of much more exhaust at higher temperature than naturally aspirated engines. You can model blown engines with this program by using an equivalent Compression Ratio and a Volumetric Efficiency which accounts for the increased intake charge. Positive displacement (Roots and "Jimmy" style) blowers are the easiest to design. For example, a 350 cid engine breathes 175 cubic inches per revolution. If you use a blower drive ratio that pumps 350 cubic inches per engine revolution, then you can enter 200 for Volumetric Efficiency to the Engine Expert and enter a CR twice the actual mechanical CR of your engine. Thus, if your mechanical CR is 6, you would enter 12 to the Engine Expert.

If you only have intake pressure information you can still estimate performance as follows. However, please note that you need both temperature and pressure data and somewhat more complex math than shown here to obtain precise answers. Example (please review the previous section, *Compression Ratio and Altitude*):

The Engine has 9.0 to 1 static compression ratio and 8.5 psig blower pressure. Atmospheric pressure gets up to 13.2 psia where you operate the engine. Standard sea level pressure is 14.7 psig.

```
Sea Level air density factor  = (13.2/14.7) = 0.898
Air density ratio from blower = [(8.5 + 13.2)/13.2]^0.7143
                              = 1.6439 ^ 0.7143
                              = 1.4263

Sea Level equiv air to engine = 0.898 x 1.4263 = 1.2808 = 128.1%
Sea Level equiv CR            = [(8.5 + 13.2)/14.7]^0.7143 x 9.0
                              = 1.3207 x 9.0 = 11.89
```

So, you should use a Volumetric Efficiency (VE) of 128.1%, and Compression Ratio of 11.89 sea level equivalent, when entering data for this engine into the program. The program will recommend an exhaust system, and estimate the Torque and Power produced. For the fun of it, why don't you run this example on the default 350 Chev engine in the program. Enter 5,000 RPM for the engine speed at Peak Torque. You will see why people can get excited about these packages!

Notice that the program does not subtract the power absorbed to drive the blower. Although blowers/superchargers absorb a great deal of power, some of that horsepower is recovered during the intake stroke, when positive intake pressure PUSHES the piston down. This positive pressure also reduces critical tension loads on the connecting rods. In any case, you can be sure that between 2 blown engines designed with this program, the one that reports more power will make more power in the real world. How much power depends on how much power the blower/supercharger absorbs.

As with naturally aspirated engines, don't get carried away with super high equivalent compression ratios!

Intake Airflow at 28"

Intake Airflow at 28" is the flow–measured at 28 inches of water–that the head, valve train, and cam TOGETHER make available to your engine at (or before) peak cylinder demand. To convert airflow data from 10 inches water to 28 inches, multiply by 1.673; or, to convert from 25 inches water, multiply by 1.058.

Peak cylinder demand occurs at 72 to 82 degrees ATDC, depending on the Stroke, Rod Length, and Engine Speed at Peak Torque. Engine Expert tells you exactly where the peak cylinder demand occurs at the end of each simulation; so, you must assure that the valve train and heads really meet your requirements. If not, the engine won't achieve the power and power band you intend. You must also check the Critical Intake Port Dimensions reported by Engine Expert. We have not (yet?) found an engine that will achieve Peak Power at an engine speed greater than allowed by the minimum intake port area of the head. For example, SB Chev heads (461 "Fuelie" and 292 "Turbo") are limited to just over 2.3 square inches in two places–first by the push rod tunnels and second by the cylinder head bolt and intake valve spring pocket where the floor rises before rolling down to the valve seat. Engines using these heads reach Peak Power at

2.30 to 2.36 square inches of Critical Port Area and about 255 cfm, even when tested at 270 cfm on the flow bench.

Roller tappet, overhead cam, and 4-valve engines open the ports to maximum flow earlier than flat tappet, pushrod engines–and so achieve Peak Power at higher engine speed than flat tappet pushrod engines. Thus, the Engine Expert asks you to enter your valve train type, and adjusts the power band accordingly.

The default cylinder head flow is computed from the cylinder bore diameter. The default value is a good estimate across a range of engines from British 4 cylinders, through the Indianapolis Ilmor-Chev, to American V-8's, and includes the Allison 4-valve aircraft V-12 with 5.500 inch Bore! In general, the default value is achievable with some porting work. Starting with a contemporary cylinder head design, a professional porting shop–using flow bench experience–can achieve up to 20% greater flow than the default.

Torque & Power Engine Speeds, Flows, Etc

Typically, engine speed at Peak Torque is about 76% of engine speed at Peak Power, so this is the default in the program. When you enter the RPM at Peak Torque, the program suggests an RPM at Peak Power which is 32% higher. Likewise when you enter RPM at Peak Power the program suggests the corresponding RPM at Peak Torque, the program suggests an which is 32% higher. You can enter the suggested values or other values that you might have. For example, engines with the right balance of intake & exhaust tuning, camshaft, and breathing will show 2000 to 2500 RPM from Peak Torque to Peak Power. If you are designing such an engine (often a road racer), use the more appropriate values instead of the suggested values.

Engine speed at Peak Torque should be about 500 RPM higher than the point at which the engine must accelerate hard at WOT. (See also, the discussion of Peak Intake Velocity and low velocity intake port flow.) Typically, this value will be from 3500 to 5800 RPM for oval track and bracket racers and 5800 to 9000 RPM for drag racers and small bore road racers. The RPM at Peak Power must be within g load and piston velocity limits your parts can tolerate, and also within the limits imposed by your heads, cam, valves, and intake/exhaust systems.

The program designs the intake dimensions for Peak Torque RPM, and the Exhaust at Peak Power RPM (although exhaust system recommendations show sizes for both).

Flow Demand CFM (at 28 inches of water depression), Carburetor sizes, Piston Velocity, Critical Intake Port dimensions, and Exhaust Flow versus crank rotation are shown for Peak Power RPM. Flow Demand is computed at Peak Power RPM, because the intake system must not be restricted. Carburetor size is computed for Peak Flow Demand at maximum piston speed (72 to 82 degrees ATDC), rather than for average engine flow.

When you need to get values for a particular engine speed, you must first determine whether the values you need are dependent on RPM at Peak Torque or at Peak Power. Then, you can enter the proper engine speeds to compute the values you need.

Total Ignition Advance (TIA)

The Total Ignition Advance (TIA) requirement for your engine, in degrees before Top Dead Center, depends on Compression Ratio, fuel type, combustion chamber design, and engine speed. Small bore, high-compression engines with good turbulence require less than 24 up to 36 degrees TIA. Low-compression, low turbulence &/or large piston dome engines require 48 to 55 or more degrees total advance. If different from the default, you should enter the TIA the engine prefers from dyno tests, track tests, or other experience.

Even though Engine Expert shows changes in power as you change TIA, you should not interpret this as a guide for tuning the engine. You should enter the TIA that the engine prefers. The LESS Total Ignition Advance an engine prefers, the sooner it builds combustion pressure to turn the crank and make torque! The MORE advance an engine prefers, the LESS Torque and Power it makes relative to a similar engine that prefers less!

The Engine Expert computes the suggested Total Ignition Advance (TIA) using effective Compression Ratio, Bore Diameter, and engine speed. The default TIA estimate will be close enough for most engines. However, the Engine Expert assumes that the piston dome and cylinder head do not significantly block flame travel, and that there is not much turbulence. If a large piston dome is required to get the compression you want to simulate, you will have to increase the TIA 30% or more. Conversely, TIA can be reduced to 75% of the default with a flat-top piston style combustion chamber. Similarly, if you have a high-turbulence / fast-burn combustion chamber, you can reduce the TIA to as little as 45% of the default. By the way, always remember that the combustion chamber includes the top of the piston!

If you can obtain the upper RPM portion of the ignition advance curve for a combustion chamber similar to yours from dyno testing or written specifications, and you know the engine speed at Peak Torque, then you can compute TIA as follows:

Example:

```
32 degrees advance at 3000 RPM
37 degrees at 5000 RPM
Peak torque at 4300 RPM

TIA = ((37 - 32) X 4300 X 4) / (5000 - 3000)
    = (    5     X 4300 X 4) /     2000
    = 43 degrees
```

If you find when testing that your spark advance requirement stays constant, or declines as engine speed increases, you have either come up against detonation, or combustion turbulence increases nicely with RPM. High turbulence combustion chambers can prefer constant or even slightly declining degrees of ignition advance as RPM increases. In such a case, use the TIA that works best in your tests.

Program Output

Flow Demand Lead / Lag in Intake & Exhaust, and Intake Reversion

There are other time delays in a piston engine besides Combustion Pressure Buildup. These delays occur because pressure changes occur at opposite ends of the cylinder (valves/ports at the top and the piston at the bottom). Pressure changes caused by the piston changing velocity and by valves opening and closing travel no faster than the speed of sound in the cylinder! For example, the intake port doesn't know that the piston has changed direction across BDC until pressure from the upward moving piston reaches the port, after traveling up the cylinder.

The Engine Expert computes the lag time for intake flow demand to reach the intake port, for each degree of crank rotation at Peak Torque RPM. Similarly at Peak Power RPM, the Engine Expert computes the lag time for rising piston pressure to reach the exhaust port during the exhaust stroke. These times affect Flow Demand on the graph and on the Air Detail report. If there were no lag time Flow Demand would reach zero at BDC. Lag time may increase the intake cycle up to 10 degrees after BDC, and may reduce the exhaust cycle up to 7 degrees.

Intake Reversion, as output on the Air Detail report, is the tendency of the piston to push the intake charge past the valve, back into the intake manifold, when the intake valve is open after BDC. When the engine is operating in its power band, intake flow inertia overwhelms the reverse pumping tendency–if the valve isn't open too long. However, at engine speeds less than the Torque peak, intake reversion with a long duration cam in a long rod engine can be significant. The report shows the percent of cylinder volume, with degrees adjusted for lag at Peak Torque RPM, trying to escape back to the intake port after intake flow demand ceases.

Another delay occurs during the power stroke, from the time the exhaust valve opens until the piston experiences reduced pressure. Although the time delay is constant, the crank degrees before pressure is relieved increases with increasing engine speed. Thus the power stroke increases slightly with increased engine speed, while the exhaust stroke, in crank degrees as well as time, shrinks with increased engine speed. This lag effect is not shown on the output reports, but is included in the Nominal Torque Sum the Engine Expert shows while you are designing an engine.

All these leads and lags may be confusing at first. Just remember, the Engine Expert will deal with the Intake at Peak Torque engine speed, and the Exhaust at Peak Power engine speed. Degrees of crank rotation are adjusted to show you what is happening AT THE PORT so you can make a reasonable cam selection.

Combined Volumetric Efficiency

Combined Volumetric Efficiency is the product of Internal Volumetric Efficiency multiplied by regular VE (entered when the program begins to run). Combined Volumetric Efficiency is used to compute exhaust primary tube sizes. Combined Volumetric Efficiency is equivalent to Volumetric Efficiency as defined in engineering textbooks.

Internal Volumetric Efficiency and Effective Compression Ratio

Internal Volumetric Efficiency, as we use the term, is the ratio of cylinder flow demand at the intake valve, divided by the cylinder volume.

Because cylinder flow demand continues for more than 180 degrees, as described above, Internal VE is in the range of 102% to 105%. The program computes Internal VE, and multiplies it with your input VE to calculate Torque Factors, Peak Torque and Peak Power Estimates, combustion pressure buildup time, and recommended exhaust tube sizes. Internal VE rises with increasing Stroke and engine speed.

What Are Torque Factor, and Nominal Torque Sum?

The effects of engine speed, Compression Ratio, Rod Ratio, Total Ignition Advance, and Power Stroke Pressure Lag are ALL combined into one set of values–the Torque Factors–computed for each degree in the power stroke. Essentially, Torque Factors show how different engine designs develop torque. You will find that the maximum Torque Factor occurs 22 to 27 degrees ATDC in any engine design. This fact emphasizes the importance of minimizing Total Ignition Advance and paying close attention to ring seal during the first 1/2 inch of piston travel. Also, this shows the importance of ring drag on power.

Gas pressure behind the rings is enormous in an all-out race engine. Engine builders who achieve high levels of sealing with a single, thin compression ring experience significant friction reduction! A single, thin ring also reduces blow-by from ring flutter.

Torque Factors are independent of displacement. The larger the NOMINAL TORQUE SUM (more on this follows), the more torque, and generally power, the engine will make per cubic inch.

The Nominal Torque Sum is the standard of comparison for alternative engine designs. As you develop your engine design interactively, watch for improvements or losses in the Nominal Torque Sum with each change. The Nominal Torque Sum is defined as the Torque Sum you get relative to a cam and exhaust port that relieves pressure at 100 degrees ATDC in a 350 Chev at 5000 RPM (which requires valve opening around 97 degrees ATDC). This standard is somewhat arbitrary – but we find it to be a stable and reliable standard from which to compare high performance engines! The Nominal Torque Sum is also used to compute Peak Torque and Power.

As you may have guessed, there is a lag time from when the exhaust valve opens until pressure drops at the top of the piston. Engine Expert calculates this Power Stroke Pressure Lag and uses it to compute the Nominal Torque Sum. Power stroke lag is one of the few factors that work in your favor when you go for higher RPM's. As the rev's climb you can open the exhaust valve 1 to 3 degrees earlier (dumping exhaust gases during a longer crank period) without decreasing the Torque Sum.

What About Peak Torque and Power Estimates?

After Peak Torque, engines begin to run out of airflow capacity, so VE and torque drop from their peak values. An engine with a narrow power band may have only 800 RPM from peak Torque to peak HP, while a carefully developed road racer may have more than 2500 RPM in its power band. The power band your engine achieves depends on your requirements and success in tuning. In any case, the Engine Expert tells you the Power you may achieve at the Peak Power RPM you select, assuming Torque has dropped 7% from its peak value.

The Engine Expert shows you the airflow required in CFM to achieve Peak Power RPM, so you can select heads and cam suitable for your design. (See also the section on Optimum Tuning and Cam Selection.) You must also be sure that your heads have at least the Critical Intake Port area recommended by the Engine Expert. Of course, you may find that there are no adequate heads for your design in the engine speed range you want to use. We think you will prefer finding this out with the Engine Expert instead of on the dyno or track. You may be able to get around the problem by using a shorter stroke, larger bore, and/or longer rod to reduce the peak flow demand. Conversely, if your heads and an acceptable cam allow more flow than you require, you may decide to lengthen the stroke, shorten the rod, or increase the bore to take better advantage of the parts you have, in the engine speed range you want to use.

If the g loads of your engine design are higher than parts can tolerate, you should shorten the stroke, lengthen the rods, or find some other way to avoid a blow-up!

The actual Torque and HP your engine produces will probably be somewhat different from what the program predicts. It will always be easy to make less! If you take one engine design from this program, because it estimates more power than you think your competition has, you can get yourself into endless difficulties. The Engine Expert estimates performance; we do not guarantee your results. However, you CAN be sure that between 2 engines simulated with the Engine Expert, the one with the higher estimate has a higher power potential in the real world. Always start with a baseline engine design that you have run, or you know to be competitive. Then develop a new design with the Engine Expert until you find a combination that shows greater performance, durability, and/or lower cost than the baseline!

How Intake & Exhaust Sizes are Computed —

Why a Smaller Intake Area is Better

The Engine Expert recommends intake port size for Max Torque RPM and exhaust port size for Max Power RPM which flow the specified velocities at the maximum piston speeds (rather than at average cylinder flow demand). This method is a whole lot more useful than sizing based on average flow demand. Peak flow demand varies up to 10% with rod ratio.

When you compare 2 engines with similar displacements and power estimates, and one has a smaller intake and exhaust port size than the other, it tells you that the smaller port is working longer and easier. The smaller port engine needs less valve lift, less valve size and weight, softer valve springs, and may have a wider power band. That is, smaller port engines move gases at high velocity (but under 55% of sonic velocity) for as long as possible in the cycle. Such ports turn cylinder flow demand into kinetic energy in the intake tract as efficiently as possible.

Intake port size is computed without any volumetric efficiency factor, because the flow which alters VE occurs before and after peak flow demand at WOT. Exhaust flow, however, is multiplied by the Combined VE (the product of the Internal Volumetric Efficiency and the VE specified in the input data). That is, whatever quantity of gases are contained in the cylinder are expelled throughout the entire exhaust cycle.

You can use the Engine Expert to estimate the lower end of the power band. Torque drops rapidly when the intake velocity is too slow for good cylinder filling after 70 - 80 degrees ATDC. When you have an engine design you decide to test further, reset the Peak Intake Velocity (PIV) to 200 Ft/Sec. Then, enter low values for RPM at PEAK TORQUE and check the computed intake manifold size. As long as the computed size is LARGER than you will actually use, the engine will operate at that level. However, when the computed intake size is SMALLER than you will use, the engine is unlikely to have useful torque at that engine speed. This procedure is based on typical American V-8's. Engines with fewer cylinders, intake cam duration long after BDC, and/or restricted exhaust systems will have less low end power band than indicated here. Again, you should develop your own experience, or borrow the best you can find, to set PIV at proper levels for the engines you build. As a guideline, remember that kinetic energy for cylinder filling increases as the square of Peak Intake Velocity.

Using a Large Exhaust System with an Earlier Peaking Intake

Exhaust systems operate best when the exhaust velocity is at the optimum speed. The program computes exhaust tube diameters for the specified Peak Exhaust Velocity at both Peak Torque and at Peak Power engine speeds. We recommend relatively large primary tube headers with longer tube lengths, close to the torque length. Here's why: First, too small an exhaust system kills more power from back pressure than one too large loses from low inertia scavenging. Second, you normally have more space to work with the exhaust than you do with the intake. By using long primary tubes and a collector, you can extend the operating range of the exhaust downward, without restricting top end flow. You won't have much success extending the operating range of an exhaust system upward though. Third, exhaust system scavenging across TDC overlap will increase early intake flow before peak intake velocity can be demanded by the piston. Thus a late peaking exhaust can broaden the operating range of the intake upward. We have observed a BB Chev drag motor at peak power on the dyno, exhausting a thin haze of soot caused by over-scavenging from a 106 degree lobe center cam and a great set of headers. Conversely, there is no intake energy that extends the operating range of the exhaust up or down.

Finally, high velocity intake inertia, supercharging, as well as TDC scavenging help extend the operating range of an intake system upward. However, relatively little can be done to extend the operating range of an over-sized intake system downward.

Some of the top engine builders use stepped headers. These headers use 2 or more sizes of primary tube, becoming progressively larger away from the cylinder head. As you use this program, you will find extreme performance and/or high engine speeds call for very large primary tubes. Stepped headers achieve larger sizes without the shock loss of a big "reversion" step at the header flange, and without the expense of conical, megaphone tubes.

Optimum Tuning of Intake and Exhaust, & Optimum Cam Selection

See also the Technical Topic "Estimating Cam Timing Requirements"

The flows and crank degrees (adjusted for lead and lag) shown on the reports, can be used to identify cam requirements for different engines, using experience about what has worked in the past.

Flow bench information is required to match flow rates and valve lift to engine demand! The Engine Expert shows CFM for flow bench values at 28 inches of water vacuum. To convert to 10 inches, divide by 1.673; or, to convert to 25 inches divide by 1.058.

Two values of flow demand are shown: a MINIMUM value sufficient with a racing roller cam or racing overhead cam engine, and a MAXIMUM value required with the slower valve acceleration of a flat tappet/pushrod cam. A mushroom lifter cam (or any large tappet style cam) requires flow between these two values, to achieve the estimated horsepower at the RPM shown.

A Critical flow event occurs at TDC overlap, where the exhaust valve should be closing into the poor port flow region, and the intake valve should be opening into the good flow region of its port.

Short-rod engines have higher flow demands across TDC overlap, and so tend to have reversion or over-scavenging troubles. Thus, short-rod engines prefer less valve lift across overlap than long-rod engines; yet, they also demand more flow closer to TDC overlap and more intake duration after BDC.

Long-rod engines (particularly long stroke, high RPM ones) delay their peak airflow demands farther on each side of TDC overlap than short-rod engines, and have lower peak intake airflow demand. Combined with their greater tolerance for overlap, such engines require less radical intake opening and exhaust closing rates than short-rod engines. Thus, they prefer narrower lobe separation cams, with slower intake opening rates and thus lower valve spring rates. They also prefer less intake duration, resulting from earlier closing. On the exhaust side, long-rod engines need more low-lift port flow, &/or a faster opening exhaust valve, and delayed exhaust valve opening. Frequently, a long-rod engine's requirements can be met with 1 to 3 degrees cam advance, albeit with a few percent loss of Torque.

Short-rod characteristics are noticeable for rod ratios (Rod Length/Stroke) less than 1.55, while long-rod behavior is pronounced for rod ratios greater than 1.75.

Another important flow requirement occurs at peak intake flow demand. The intake valve must be open into the maximum flow zone of its port at this time. To see what this means, use the Engine Expert to design a 496 BB Chev. Enter 4.25 stroke, 4.311 bore, 6.135 rods, and 10.45 CR at 4500 RPM peak Torque. You will see why roller cams work! They can open the intake faster than a flat tappet cam in the short period from TDC till peak flow demand. Roller cams also reduce pumping losses in general by holding the valve open in the high flow region of the port for a longer period of time. Very short rod engines, such as the 496, prefer roller cam timing! Conversely, short rod engines produce torque earlier in the power stroke, and relieve pressure on the exhaust gas more slowly across BDC than long rod engines. Thus they tolerate early opening exhaust timing, and work well in engines biased toward too much intake and not enough exhaust flow capacity (like the Hi-Perf Big Block Chev's and most Ford's)

Use the Intake Air Flow curve on the graph to see if the valve is sufficiently open. This graph curve is produced by combining cam profile and head flow test data from the files you have selected in Graph Setup.

The intake closing point is CRITICAL. In simplest terms, the intake valve must close at just the right time to capture the maximum amount of intake charge. The greater your Volumetric Efficiency, engine speed, and Stroke, the later the intake must close. Engineering analysis for intake closing requires simulation of the intake system, flow characteristics of the cylinder head, and exhaust scavenging across TDC overlap.

Using valve opening and closing rates currently available, the intake valve will generally be equally open at TDC overlap and at the 4.5% reversion point. This information allows us to estimate the lobe separation angle for engines with average (around 75%) exhaust to intake flow ratios. For example, if your simulated engine shows 4.5% reversion occuring 37 degrees ABDC, you add 180 + 37 = 217 degrees, and divide by 2 = 108.5 degrees. So you would look for a cam with 108 to 109 degrees lobe separation.

The longer the stroke, the poorer the exhaust to intake flow ratio, and/or the higher the RPM, the more lobe separation an engine prefers.

Next, you need sufficient lift to meet the "Required Intake Flow at 28 inches" and the "Single Carburetor Flow (at 1.5 inches Hg)" as reported in the Air Flow Results. If your head and cam combination achieve better flow than required at the timing point reported, your engine speed range will be higher than computed. Conversely if your head and cam cannot achieve the indicated flow and timing specification, then the power and power band, will be less than computed.

NOTE: Intake valve opening greater than required for maximum port flow at the degrees ATDC reported will raise the engine speed at Peak Power, and may narrow the power band !!!

The exhaust opening point is generally least critical. You can advance or retard this point +/-5 degrees with only a few percent effect on torque. In any case, the exhaust has a much easier task than the intake, unless it is terribly poor. The exhaust cycle allows 60 to 96 degrees during which more than 60% of the exhaust gas escapes out the port under its own pressure, BEFORE THE PISTON HAS TO PUSH, AT ALL !! Cam lobe centers can be 1 to 3 degrees closer together when exhaust flow capability is more than 75% of intake flow. The Buick V-6 with aluminum racing heads works best with 105 degrees lobe separation. Conversely, when exhaust flow is less than 70% of intake flow, the exhaust timing (and lift) should be greater than for the intake, and/or the lobe centers may be widened a few degrees or the cam may be installed advanced, allowing more time for the exhaust to escape.

Recommended and Critical Port Sizes, Valves and Valve Lift

The Engine Expert can help with intake valve size and lift; and, you can determine whether intake size may restrict your engine speed. The "Port Entrance Diameter if Round" as reported in the Air Flow Results is also the minimum diameter intake valve the engine wants. Theoretically, a port will reach maximum flow when valve lift reaches 25% of the valve diameter. In fact, flow-bench prepared racing ports will demonstrate increased flow at lift up to 33% of the valve diameter. Thus, the valve/port diameter recommended by the Engine Expert also defines the range of valve lift required at Peak Cylinder Flow Demand (72 to 82 degrees ATDC)–if you can use a valve equal to the recommended port diameter! If your valves are larger or smaller, then the required lift will be proportionately smaller or larger.

The Engine Expert can check for valves that may be too large (hurting low-range power and low/mid-lift flow). Intake dimensions are computed for peak Torque RPM. So, to find an upper size limit, enter the engine speed at Peak Power it the entry blank for Peak Torque RPM, and write down the intake port/valve size recommended! If you can fit valves (without cylinder wall shrouding) in this size range, then the intake valve shouldn't present a problem. If your intake valve is larger then it is too large. Conversely, if the best flowing intake valve & port combination you can use is SMALLER than the program recommends at peak torque, you are operating at, or beyond, the flow limit. In this case, you trade specific Torque for high RPM Power with diminishing returns. *Review the section "YOUR INPUT: Volumetric Efficiency" to see the effect of small valves.*

Intake Port Bowl Area & the Location of the Critical Area

In a properly designed port the air is accelerated from the port entry to the critical area. Air speed must be maintained until it leaves the valve. If the bowl area is too large it limits the air speed. If the air speed falls below 600 feet/sec the air can not sufficiently fill the cylinder. This will reduce volumetric efficiency, peak power, and the rpm at peak power. The bowl should be sized to keep the air speed at the bowl between 600 and 650 feet/sec at peak horsepower RPM.

"Critical area" is the point of highest air velocity in the intake. Its location in the intake influences volumetric efficiency. There must be sufficient volume below the critical area to fill the port. If the critical area is too close to the valve, air will not be available when it is needed. This will reduce volumetric efficiency, peak power, and the rpm at peak power.

Engine Expert uses port bowl area to calculate the bowl peak velocity, bowl diameter (for a round bowl area), and bowl dimensions (for a square bowl area).

Volumetric efficiency, port bowl area, critical area, and peak power RPM are used to calculate the volume below the critical area and the location of the critical area. The calculations assume a port bowl diameter to lift ratio of 0.400.

Technical Topics

Estimating Cam Timing Requirements

The camshaft and valve train simply enable an engine to reach its airflow potential. Valve action should be no more than necessary to get the job done, because excess valve lift and opening & closing rates:

- Raise the cost and maintenance expense of your engine, while lowering its reliability.
- Waste power through greater friction, which appears as heat in the oil.
- Narrow the power band, or move it where you DON'T want it.

When the cylinder head airflow, port sizes, intake and exhaust tuning, bore, stroke, rod length, and strength of components are compatible, then proper valve timing delivers the power you need to win. However, identification of "proper" valve timing has been a black, mysterious art since the first poppet valve engine clattered into life. Some aspects of valve timing will remain a black art for a while longer. Nevertheless, the ENGINE EXPERT can help you avoid some of the mystery, and better identify critical valve timing events.

Estimating Cam Timing Requirements: Intake

As the piston retreats from TDC and the intake valve opens, air (and fuel) are accelerated along the intake runner by increasing cylinder demand. Cylinder airflow demand peaks at 72 to 84 degrees after TDC, and then declines to zero by 5 to 10 degrees after BDC. However, air flowing in the intake runner does not decelerate as quickly as cylinder demand decreases. The faster the air flows (without friction or turbulence) in the intake runner, and the larger its volume, the more air will stuff itself into the cylinder — raising volumetric efficiency over 100%. Proper intake timing:

1) Allows cylinder demand to accelerate intake airflow as quickly as possible.
2) Provides sufficient airflow to the cylinder at peak cylinder demand to meet performance objectives.
3) Closes the intake port when the pressure in the cylinder (from the rising piston) equals pressure in the intake runner (from air still flowing toward the valve).

Engine performance will be different than intended when intake valve timing does not satisfy all 3 criteria.

To satisfy criteria 1 and 2, the intake valve(s) must begin to open before TDC overlap; and, if the exhaust system works properly, a small vacuum in the cylinder and exhaust pipe will start intake air flowing into the cylinder. In any case, your intake valve timing MUST provide as much airflow to the cylinder at peak cylinder flow demand as the ENGINE EXPERT requires for the type of cylinder head/cam you use.

For example, if your engine design requires "250 to 276 CFM Int Flow at 28 inches water by 79 degrees ATDC," then your head, cam, and valve train must provide airflow by 79 degrees ATDC as follows:

- For a flat tappet high performance cam, 276 CFM
- For a drag race roller cam or 4 - 5 valve cylinder head, 250 CFM
- For an endurance racing roller or overhead cam head, 263 CFM (midway between the high and low CFM values)

You can calculate intake valve lift requirements based on cylinder head airflow data. Using the example above, if you have cylinder heads that flow 263 CFM at 0.550 valve lift, you could use an endurance (street & strip) roller cam to meet the design objective. Current cam profiles, when installed with the lobe centerline at the proper position, will reach a little more than 95% of net valve lift at cylinder peak demand. Thus, for a cam with 0.026 clearance lash you need:

```
(0.550/0.95) + 0.026  =  0.605 inch Gross Valve Lift
```

This number allows you to calculate the rocker arm ratio needed with various cam lobe lifts. Be sure that you achieve the advertised rocker ratio when the rocker arms are installed in your engine. Installed rocker arms frequently measure 0.100 less ratio than advertised!

If you can't achieve the required flow capability — with the minimum port area as big as the "Critical" area computed by the ENGINE EXPERT — the engine speed at peak power will be less than you want and peak power will be low as well. If you achieve much more flow than required, peak power will occur at a higher engine speed than planned, which may exceed the strength of your rods, etc. (And I don't much enjoy cleaning up oil, water, and broken parts, do you?)

The "Critical" port area computed by the ENGINE EXPERT is just as important as the cylinder head airflow capacity. Your engine will not achieve peak power above the engine speed dictated by the critical port area — even though flowbench measurements indicate otherwise. For example, small block Chevrolet intake ports are limited to 2.35 square inches in two places — first, by the intake pushrod separation, and second by the cylinder head bolt and spring pocket where the port turns down to the valve head. As the result, these engines consistently achieve maximum performance at 255 CFM airflow — which corresponds to 2.35 square inches of critical port area. Similarly, we have seen pro stock type engines limited by 3.625 square inches area to 390 CFM airflow, even though the flow bench measured over 425 CFM.

The "Critical" port area is a good estimate for the size of the intake throat where the port turns to the head of the valve. Reduced area at this point allows airflow to change direction more efficiently.

A port throat larger than the critical area rarely shows increased airflow in the usable range of valve lift, and the reduced throat velocity from larger area doesn't improve an engine's operating characteristics.

You may find that your engine design requires so much flow early in the intake cycle that valve lift across TDC overlap is excessive. Valve/Piston interference, exhaust reversion into the intake runner, or over-scavenging of intake air & fuel directly out the exhaust can occur. However, you can relieve these problems through A) revised porting and/or B) engine design modifications.

A) You can revise the intake ports to:
   - Reduce low-lift airflow (particularly reverse airflow). This also makes the engine more tolerant of long intake duration.
   - Increase mid- to high-lift flow, possibly with a larger intake valve, so that less lift is required

B) You can modify your engine design to delay peak flow demand 1 to 3 degrees by:
   - Increasing the Stroke, AND increasing the Rod/Stroke ratio.

- Increasing the engine speed at peak power (which may require a smaller Bore to lessen reciprocating weight and reduce displacement — IF airflow in the smaller Bore is adequate)

B) When your engine design requires peak airflow at less than 78 degrees ATDC, you usually have to use a cam &/or cylinder head type that allows air flow to increase rapidly, such as:

- Changing up to a mushroom tappet or roller tappet cam and valve train
- Changing to an overhead cam or 4 or 5 valve cylinder head

The 3rd criteria, intake valve closing, depends on several factors and is the most difficult to determine. The optimum closing point is best determined by coordinating intake manifold design with dyno testing.

Fortunately, a computer program for manifold design will be coming on the market soon, and several companies have instrumentation to measure intake port pressure, cylinder pressure, and crank position during dyno testing. The proper intake valve closing point depends largely on the strength and timing of resonant pulses in the intake manifold. These pulses can yield volumetric efficiency greater than 125%, OR cause fierce "reversion" if the pulses snatch fuel-air out of the cylinder after BDC. This "reversion" problem may be reduced by lessening intake timing after BDC, making reversion steps or blocks in the intake runner, or by re-designing the intake manifold. Unless intake timing is seriously wrong, the best solution is to change intake manifold design.

In addition to intake manifold design, the correct intake closing point depends on engine speed, intake restrictions, Rod/Stroke ratio, and Stroke length.

- The higher the engine speed, the later the intake valve closing point should be.
- The more restrictive the intake port, runner, and/or valve size, the lower the pressure in the cylinder will be, and the later the closing should be (such as pro-stock engines).
- The more restrictive the air inlet/carburetor, the lower the pressure in the runners will be, and the earlier the closing should be. (Such as NASCAR 390 CFM carburetors and restrictor plate engines.)
- The larger the Rod/Stroke ratio, the sooner intake closing should be; and, the more sensitive an engine will be to intake closing. If you examine Page 2 reports from the ENGINE EXPERT, you will find that a short rod engine (less than 1.65 ratio) requires MORE than 3 degrees of crank rotation per 1% of reverse pumping, while a long rod engine (greater than 1.90 ratio) requires LESS than 2 degrees of crank rotation per 1% reverse pumping around 35 degrees after BDC.
- The longer the Stroke, the later the intake valve closing point should be.

The ENGINE EXPERT allows you to translate the proper crank position for intake closing from one engine design to another. For example, suppose you simulate an engine that has shown superior performance with intake closing at 73 degrees after BDC. When you examine the Air Details Report for this engine you find that the Intake Reversion Percentage at 73 degrees after BDC is 25%. To translate this knowledge to a new engine, find the degrees after BDC corresponding to the same Intake Reversion Percentage for the new engine. If the new engine design peaks at a higher engine speed and uses a longer Stroke and/or smaller Rod/Stroke ratio, you may find the new design reaches 25% Intake Reversion at 78 degrees after BDC. Thus, if the new engine has similar low-lift intake airflow characteristics and achieves similar Combined Volumetric Efficiency, the intake should close at 78 degrees after BDC in the new engine.

When you don't have experience with a similar engine design, you can make a reasonable estimate based on whether the engine is a street/strip, wide power band race, or pro-stock type engine:

In a street/strip flat tappet engine it is difficult to get the valve lift, intake lobe centerline (more on this later), and duration (220 - 230 degrees at 0.050 tappet lift) you would like. A rule-of-thumb is to select the best compromise cam you can find, and install it with equal lift at TDC and at the 4.5% Intake Reversion point after BDC.

In a wide power band race engine (that is, where the valves and ports are adequate to the airflow requirements), the 0.050 tappet lift intake closing point can be placed at, or a few degrees beyond, the 15% Intake Reversion point.

In pro-stock type engines the intake valve and/or maximum intake port cross-sectional area is smaller than required for optimum airflow — and intake manifolds are carefully developed for maximum VE. Thus, manifold pressure remains greater than cylinder pressure long after BDC. These engines prefer the 0.050 intake tappet closing point around the 25% Intake Reversion point.

Detailed testing is still required to determine the best intake closing point for each engine type. However, preferred timing normally lies within plus or minus 3% of the estimates described above, and you should start by testing performance with an early closing point.

The intake lobe centerline should be installed based on the peak cylinder demand point reported by the ENGINE EXPERT. Experience shows that peak cylinder demand at 80 degrees ATDC corresponds to an intake lobe center at 103.5 degrees ATDC relative to 0.050 tappet lift. For example, if your engine design requires "250 to 276 CFM Int Flow at 28 inches water by 79 deg ATDC," then the intake lobe centerline should be at 102.5 degrees ATDC. If this type of engine runs best at 12% Intake Flow Reversion occurring at 55 Deg After BDC, then intake duration will be

```
((180 + 55) - 102.5) X 2 = 265 degrees at 0.050 lift
```

For a long stroke, high RPM engine, peak flow demand may occur as late as 84 degrees ATDC; which places the intake centerline at 107.5 degrees ATDC.

Thus, the ENGINE EXPERT helps you find:

- Cam or cylinder head type required
- Gross intake valve lift required
- Rocker arm ratio required
- Intake lobe centerline relative to 0.050 lobe lift
- The intake duration (or the range of duration) required around the intake centerline to close the valve at the proper time.
- Critical minimum intake port cross-section area at the throat, and cross-section area for best torque

Estimating Cam Timing Requirements: Exhaust

Exhaust gases exit a high performance engine much differently than intake enters the engine — and quite differently from the way many people think. The upward exhaust stroke of the piston does not push exhaust out of the cylinder. Instead, when the exhaust valve opens, about 60% of the gases in the cylinder flow out the exhaust port — escaping from the hot gas pressure in the cylinder — while the piston is still moving down to BDC. This phase of the exhaust cycle is called "blow-down." After BDC, the slug of exhaust gas now moving down the pipe pulls a slight vacuum behind it. In turn, this vacuum helps scavenge remaining exhaust from the cylinder, gently draws the piston upward, and ultimately pulls the first part of the intake charge into the cylinder across TDC overlap.

Thus, you can improve engine performance with exhaust ports that increase airflow from valve opening until BDC. Good low-lift exhaust flow is the breath of life to an engine. Exhaust airflow achieved at valve lift after BDC doesn't help the critical first 60% of exhaust gas escape from the cylinder!

The velocity of exhaust gas flowing through the port is largely set by the (high) speed of sound in the hot gas. The amount of exhaust that leaves the cylinder depends on the airflow capability of the exhaust port and the amount of time from exhaust valve opening until just after BDC. The total amount of exhaust depends, of course, on the cylinder displacement and volumetric efficiency. The time lag until the exhaust port knows that the piston has reached BDC is a part of the time for exhaust gas to escape under its own pressure (*see "Flow Demand Lead / Lag in Intake & Exhaust, and Intake Reversion"*). As a rule of thumb, the exhaust port airflow you wish to access across BDC will be reached at 78% of gross valve lift. For example, suppose you want to access 180 CFM exhaust flow, which your exhaust ports achieve at 0.475 valve lift, and you will use 0.026 valve clearance lash:

```
(0.475/0.78) + 0.026 = 0.635 inch Gross Valve Lift
```

This value allows you to calculate the rocker arm ratio required with alternative lobe lifts. Next, the 0.050 tappet lift exhaust opening point can be estimated by the following equation:

$$\text{Degrees Before BDC} = 0.0003954 \times \frac{\text{TRPM} \times \text{Combined VE} \times \text{cid per cylinder}}{\text{Exhaust Flow}} + \text{Exhaust signal lag from BDC}$$

This equation estimates the time duration for cylinder blow-down before BDC, and assumes you can reach adequate valve lift at the end of the computed duration. For example, in an engine with torque peak at 4500 RPM, Combined VE of 110%, 56.5 cubic inches per cylinder, and 180 CFM exhaust flow; you need to find the exhaust signal lag.

If you examine the Air Detail report the first line with a non-zero value in the Flow Demand column will show you when the exhaust port begins to receive gas pumped out by the piston. (You can also see this by examining the Flow Demand curve on the graph.) For instance assume that the entry in the Crank Angle column on that line is –174. This means that the exhaust port begins to receive gas pumped out by the piston 174 degrees Before TDC, a 6 degree lag after BDC.

Thus:

```
(0.0003954 x (4500 x 110 x 56.5)/180) + 6 =
68 Degrees Before BDC at the 0.050 tappet lift exhaust opening point.
```

There are two more events to consider in the exhaust cycle, exhaust valve closing and peak exhaust pumping. Peak exhaust pumping occurs when the pressure signal from Maximum piston speed reaches the exhaust port. Exhaust airflow must not be restricted at this point. We prefer about 10% more duration after the peak exhaust pumping point, than there is from intake opening to peak intake flow demand, adjusted for the difference between intake and exhaust valve lift requirements. This allows the valve to close more slowly than it opens (which is normal except for some overhead cam engines). ASSUMING WE FOUND AN INTAKE LOBE AND VALVE TRAIN THAT MEETS THE PREVIOUSLY CALCULATED INTAKE REQUIREMENTS, the preferred duration to 0.050 tappet closing lift after peak exhaust pumping is:

```
110% of (.475/.550) x (79 degrees ATDC + 30 Degrees BTDC opening)
=  0.95 X 109 = 103.55 Degrees
```

Next, we need to find the peak exhaust pumping event on the Flow Demand column of the Air Detail report (or by examining the Flow Demand curve on the graph). Suppose that the entry in the Crank Angle column on that line before TDC that has the largest Flow Demand is –73. This means that the Maximum piston speed signal reaches the exhaust port at 73 degrees Before TDC. Thus, adequate exhaust airflow should be available if the 0.050 tappet closing point occurs no sooner than 103.55 - 73 = 31 degrees After TDC.

Overhead cam engines with large diameter bucket tappets allow extremely fast closing rates (as do roller tappet cams). Thus, you can ask your cam grinder to put the exhaust closing point where you want it, in most cases. We like to have significant exhaust flow across TDC overlap in unrestricted engines, allowing the exhaust to start intake flow before the piston actually starts downward. YOU MUST HAVE FLOW DATA FOR YOUR HEADS AND VALVES TO SELECT THE RANGE OF EXHAUST VALVE LIFT NECESSARY. From 15% to 30% of total port airflow, at lifts of 5% to 12 % of the exhaust valve diameter, often work in unrestricted engines. Some experimentation or closely related experience with similar engines will be necessary.

The type of engine — street stock, NASCAR restricted, or pure racing — also affects the proper exhaust closing point. At TDC overlap NASCAR restricted, and street, engines (with air cleaner, mufflers, etc.) can register higher pressure in the exhaust port than in the intake. This, of course, can reverse exhaust flow into the intake port, unless the exhaust is closed soon after TDC. In pure racing engines, however, exhaust port pressure at TDC is normally less than intake port pressure, so the exhaust starts the intake charge flowing into the cylinder. If the exhaust valve remains open too long after TDC, intake charge will short circuit across the cylinder head and escape out the exhaust. These considerations usually lead to exhaust valve closing within + or - 5 degrees of the same crank angle after TDC as the intake opens before TDC. For example, using the previous intake timing events where the 0.050 intake opening point occurs at 30 degrees before TDC, the typical exhaust closing point at 0.050 tappet lift would normally occur between 25 and 35 degrees after TDC. In this example, though, we already determined that 31 degrees after TDC would be an acceptable closing point.

Thus, the exhaust timing for this engine should be in the following range:

- 273 degrees duration at 0.050 tappet lift, with the lobe centerline at 111.5 degrees BTDC
- 279 degrees duration at 0.050, with the lobe centerline at 108.5 degrees BTDC

Within this range, we prefer a cam with lesser duration and greater lobe centerline because it opens the valve faster.

Exhaust lobes for street stock engines are designed according to different criteria. In this case, exhaust valve opening is delayed as late as possible — to around 45 degrees before BDC — to improve fuel economy and low-end torque. Street stock designs yield about 10% more torque, at a lower engine speed, and a lower exhaust gas temperature than the ENGINE EXPERT computes for a performance engine. However, at peak power, stock engines must work harder to pump the exhaust gas out under pressure.

Lastly, just like the intake port there is a critical, minimum, exhaust port area. It occurs at 1,126 ft/sec for 1,600 degrees F exhaust gas temperature in the port. You can calculate the critical minimum exhaust port area by multiplying the Critical Intake Port Area by 0.609. You can also calculate the critical exhaust port diameter by multiplying the Critical Intake Circle by 0.781.

Thus, the ENGINE EXPERT allows you to identify:

- Gross exhaust valve lift required
- Rocker arm ratio required
- Exhaust valve opening point
- The range of acceptable exhaust timing duration and associated exhaust lobe centerlines
- Critical minimum exhaust port cross-section area

Turbo-charged Engine Design Tips

Non-intercooled engines are easiest to design, because you have only turbo boost and CR to worry about. Inlet charge temperature is going to be raised by the boost, so you have to hold effective CR to the same limit as if the engine were naturally aspirated or supercharged. First, you will need to calculate the density ratio of the pressurized air, allowing for the increase in pressure from adiabatic heating:

```
Density Ratio = [(Boost + Atmospheric Pressure) / 14.7] ^ 0.7143
```

For example, at 14 psig boost at sea level where atmospheric pressure is 14.7 psia:

```
(14 + 14.7) / 14.7 ] ^ 0.7143  =  1.613
```

However, there will be a tuned VE effect for the manifolding, mufflers, etc in the system. The higher the tuning VE the lower the boost pressure. Or stated another way, the higher the VE the higher the power for the same boost pressure. If you have a street engine, with manifolding and mufflers designed for convenience, then you will have a lower VE than you would have with a tuned manifold and no mufflers. A VE of 85% is reasonable for a street engine.

For example:

```
      1.613 X 0.85  =  1.37
convert to percent:
      1.37 X 100    =  137 %
```

So, we would design the engine targeted on 137% combined VE, which would require you to input approximately 134% to the program. In this case, if you have a static CR of 8.5, the effective ratio will be 11.275--a bit high for street fuels, but tolerable with GOOD premium unleaded gasoline and maybe an electronic knock sensor.

Intercooled, or alcohol-fueled, engines are more challenging. Detonation is caused by temperatures exceeding the flash point of the fuel. As long as you can keep the inlet air temperature down to 160 to 175 deg F with intercooling or lots of alcohol, you can use as much CR as you would use in a naturally aspirated setup. The Ilmor Indy engines use 11:1 CR and 45 in Hg (7.4 psig) boost. The effective CR is 16 to 1. This is a pure racing engine, where the exhaust back-pressure is less than the inlet manifold pressure. Thus, the inlet pressure that pushes the piston down when the inlet valve opens, appears as crank power that is not lost in pumping out exhaust or driving a supercharger. The resulting power is 60% to 90% greater than the simple increase in VE created by the turbo. Of course, the combustion of alcohol adds 11% power and torque over gasoline, so the numbers get interesting quickly.

The tractor puller crazies use 80 to 120 psi boost, alcohol, and less than 7.5 CR to achieve phenomenal power levels. A 310 cid inline 6 cylinder engines achieve 2500 Hp at 5500 rpm and live, for a while. Of course, when anything fails, the engines disappear in a puff of smoke, while the cylinder head goes into orbit. Effective CR's greatly exceed 29, but detonation can't occur unless (until?) the mixture leans down.

Swirl

The swirl graph is produced by combining cam profile and swirl data from the files you have selected in Graph Setup. While the use of swirl data in head and engine design is not as common as some other aspects, some facts have become generally accepted:

- Heads with unstable swirl will hurt engine performance.
- Swirl that changes direction as the valve opens is undesirable.
- Inconsistent swirl from cylinder to cylinder is also undesirable.
- The shrouding of the intake valve by the cylinder wall affects swirl.
- Small details in the shape of the bowl can affect swirl.
- Four valve heads with symmetrical valves that open simultaneously do not swirl, because the swirl induced by one valve is canceled by the other valve. However, four valve heads will tumble.
- Generally speaking, full-bore race engines make better power with low swirl.
- There is still more to be learned.