1.1 - Using the correct components.

As with any project, one of the most important steps is properly identifying the correct components that need to be incorporated. This process starts by carefully outlining the goals to be achieved. What the intended use of the engine is will guide you as to what may need to be done.

If a motor is for a street car, with a moderate increase in power, then you may choose to use some standard components. If you are building a motor for a "historically" correct restoration of a car, such as an Abarth 750, then you will have a different focus. On the other hand, if your intent is to build a motor for competition use then you will want to use a great many specialty parts, to insure both good power and reliability. This means buying the best possible parts available for the purpose that your budget will allow. As Scuderia Topolino deals principally in competition parts and services, most of what I discuss will have to do with developing competition engines.

It goes without saying that, if you are building an "all-out" competition motor, you have already spent some time developing the rest of the car. This means chassis preparation, gear ratio analysis, and brake development. All too many people start a race car project by building the motor first. I do not recommend this. My suggestion would be to build the car first and get it to handle correctly. This does not require a high horsepower motor and perhaps a standard motor could be used to do this development. This does not require a great deal of horsepower until the car is fairly well developed. Only then do you need a powerful motor to test the ultimate limits of vehicle development.

Once you have identified the components that you need, they should be assembled and laid out to make sure that everything that you need is there. Then each component must be double checked to make sure that it meets the dimensional and quality standards you have set. Do not simply assume that the part is "right", just because you bought it from a company that has a good reputation. Everyone makes mistakes, so make sure that your parts are made to specification before using them. Making the proper measurements means having your own measuring tools or at least having access to someone who can make the measurements for you.

1.2 - Preparation

Once you have assembled all of the parts and determined what you project is going to be, make a list of all of the operations that have to be performed in preparing all of the components that have to be assembled. For purposes of illustration, I will go through the steps involved in building an Autobianchi 1050 motor, assuming that everything has to be checked, measured, and renewed. Many of the steps will applicable to other engine combinations, including Fiat and Abarth 750, 817, 843, 847, 903, 965, and 982cc motors. For specialist Abarth "Bialbero" motors, those with double overhead camshaft installations, there are some other special considerations.

The list of operations below is simply an outline, and not particularly in any order. Some items marked with an "*", may be done during assembly. As I work through each particular area of the motor, in following chapters, I will expand on each of the subjects in the outline. It goes without saying, that you are going to start out with a block that does not have any obvious internal, or external, damage. The fact that it is well used, or rusty and dirty, is really inconsequential.

A. Hot tank block and head. Not a chemical hot tank, as you do not wish to destroy installed cam bearings.
B. Crack-test and pressure-check the engine block that you wish to use.
C. Check block "squareness".
D. Check block for main bearing and cam bore alignment and bore diameter size.*
E. Check crankshaft for cracks and bearing size.
F. Regrind main or rod bearing surfaces if required.
G. Clean all crankshaft passages.
H. Check camshaft for straightness.
I. Modify front cam bearing for oil pump drive oiling.
J. Check lifters for proper lifter surface curvature.
K. Check lifter bores for damage and clearance.
L. Check push rods for straightness.
M. Check connecting rods for straightness.
N. Check connecting rods for bore and pin size*
O. Check pistons for proper bore and pin dimensions.
P. Check piston/rod combination for proper deck height.*

Q. Check rockers and rocker shaft for proper alignment and clearance.
R. Check valve guide/valve stem clearance.
S. Install new valve guides if required.
T. Check valve/seat for proper seal
U. Calibrate valve springs
V. Check valve installed height.
W. Check valve spring for coil bind.
X. Check valve spring seat and "over the nose" pressure.
Y. Check rocker arm geometry.
Z. Check for correct push rod length.
AA. Check center cam bearing alignment and oil feed orifice to cylinder head.
AB. Debur block inside and outside.
AC. Remove any excess block material not required.
AD. Bore/Hone cylinders for new pistons
AE. Align hone main bearing bores.
AF. Debur, port and surface finish all head surfaces.
AG. Surface block and cylinder head.
AH. Install new cam bearings if required, line bore and check clearance.
AI. Check camshaft for clearance with connecting rods.
AJ. Check main and connecting rod bearing shell thickness.
AK. Debur any sharp edges on piston crown, check valve pocket size, depth and radial clearance.
AL. Check piston ring to ring groove clearance.
AM. Check piston ring end gap in bore.
AN. Check piston to valve clearance at 20-30 deg. BTDC and ATDC.
AO. Double check main and rod bearing clearance.
AP. Check crankshaft thrust clearance and replace bearing if required.
AQ. Install pilot shaft bushing/bearing in crankshaft.
AR. Balance crankshaft, flywheel, pressure plate and front pulley/nut.
AS. Balance pistons/rods.
AT. Determine the cubic capacity of each of the cylinder head chambers and equalize.
AU. Determine the cubic capacity of each piston crown or depression and equalize
AV. Check clutch assembly clearances
AW. Make sure that all gaskets, seals and other items are available

1.3 - Motor Assembly and Environment.

After you have all of the components individually prepared, cleaned and inspected, you are ready to begin assembling the motor. If there is anything that is not ready, then you will not be able to finish the job. That is not the end of the world, but it will simply mean that you will have to stop at some point, and then continue later.

The room that you use for engine assembly should be clean. By this I mean that if you have access to a dust free environment, use it. It does not have to be a "cleanroom environment", but this area should absolutely not contain any machinery or abrasive materials. If there is reason to modify any component, then this should be done outside of the assembly room, and the part thoroughly cleaned before it is returned to the assembly room.

Assembling a motor is like a surgeon doing a major operation. There is a certain sequence and order to be maintained.

Make a running record of all specifics about the motor as it is being assembled. This includes numbering and dating the block and head used. This will assist you, later on, when the engine needs to be rebuilt again. As you then inspect the motor it will give you a better understanding of the things you find when you disassemble the motor. Your record keeping should include all clearances maintained and torque, and/or stretch, settings for all fasteners in the motor.

Remember that we are assembling a competition motor, which will be subject to severe stresses. These records will be vital to make sure the components still comply with the original component specification.

> As an example, you will want to make a record of the overall length of each connecting rod bolt, and the location in the motor. As these bolts are designed to stretch up to 0.005-0.006 inch (0.1-0.12mm) when properly installed, you will want to make sure that they are within 0.0005 inch (0.0127mm) of their original length when removed. If they are not, then it is an indication that they have been stressed beyond their elastic limit and should be replaced.

1.4 - Post Assembly Test Procedure

Testing a motor is all about "comparison". Sure it would be nice to say that a motor develops 115 horsepower. The problem is, by what standard? Actually the finite number of horsepower is not nearly so important, as whether improvement is being made. Even "improvement" can have different meanings.

Therefore, what is important is to make sure that the same test procedure is used in all cases. Only then can you be sure that an improvement was made or not.

By preference, here is a list of performance testing options.

A. Engine dynamometer with a temperature/humidity controlled environment.
B. Chassis dynamometer
C. A stop watch.

By far the best test procedure would be to test each engine build on the same engine dynamometer. Here you have control over as many testing parameters as possible. You will be able to asses a multitude of performance parameters, not the least being torqued and horsepower.

The problem is that 95% of amateur racers will not have such a device. However, I am sure that they would have access to a chassis dynamometer. The chassis dynamometer has the additional advantage of being able to test the entire driveline package. So if you are also looking at reducing parasitic losses in the power transmission system, it can be of help there as well.

Finally, find a track where you can test. An accurate stop watch should be able to tell you if you are turning better lap times or not. You may not fully understand the reason for the better lap times.

As you can see each successive testing option introduces more variable that have to be dealt with. On the engine dynamometer "driver" variables have no impact, as compared to track testing, but each will tell you different things.

Next you must develop a repeatable testing procedure. Here are some of the steps that I include in my procedure.

1. Engine Run In
2. Carburetion settings
3. Power Runs

Engine Run In - To begin with each motor should be "run in" for a period of 20-25 minutes, in an RPM range from 2500-4000 RPM under light to moderate load with no more than 26 degrees total ignition advance. I recommend that this run-in procedure be conducted using a non-synthetic, petroleum based oil, with sufficient levels of zinc and phosphorus. This is vitally important to ensure that items such as camshafts and lifters are properly broken in. DO NOT run the engine at idle for extended periods of time. It goes without saying that you would monitor oil pressure, oil temperature and water temperature during this run-in period. See Section 3.2.2 Oils and Additives for more information on oils with the proper levels of zinc and phosphorus.

Even though I have carburetion settings as the next item, you should at least make sure that in the run-in RPM range the engine is running at an air-fuel ratio of around 12:1-13:1. If either too rich (less than 11:1) or too lean (more than 14.7:1) then this must be addressed BEFORE continuing with the run-in procedure. To rich a mixture may cause a decrease in cylinder lubrication and cause piston damage, and too lean may bring on pre-ignition or detonation.

Once this run in procedure is completed then the cylinder head should be retorqued, and the valve clearances re-set to the camshaft manufacturer’s specification.

Carburetion Settings - The most common carburetion will be either a single two barrel downdraft carburetor or, as used with the PBS 8P and TCR heads, a set of dual side draft carburetors. These could be manufactured by Solex, Del’Orto or Weber.

In the range of 2500-4000 RPM the engine will be running on either the "cross over" orifices or the main metering system. For the run-in procedure you need only concern yourself that each cylinder is operating in the correct air-fuel mixture range. In the next section I will go into more detail the carburetion tuning options for maximizing performance. For a complete discussion of carburetion settings, please refer to section 3.4.1 Carburetion and Fuel Injection.

Power Runs - Here is where we find out what the optimum settings are for horsepower (and of course torque). This will be a combination of a multitude of settings, including ignition, fuel, cam timing, valve settings and other variables. What is important is that you develop a methodology that can be repeated so that improvements can be noted.

You must make sure that certain basic parameters are properly adjusted before any power runs are attempted. These are:

> Oil Change - If you wish to change to synthetic oil, then this would be a good time. Again, make sure it is oil with elevated levels of zinc and phosphorus (1200-2000 PPM).
>
> Camshaft timing - Usually set during engine assembly (for chain driven camshafts this is usually 4 degrees advanced).
>
> Ignition Timing - Use a total advance of 28 degrees as a starting point.
>
> Fuel Pressure - Maximum pressure of 3-4 PSI, with adequate flow.
>
> 1. Carburetion Testing - Optimizing carburetion settings for Idle, Cross-Over, Full Throttle and Acceleration.
> 2. Venturi air flow optimization.
> 3. Ignition timing variations.

Power runs may combine changes in any of the above elements to finalize what the best combination of settings may be. It is important to maintain good air-fuel ratio measurements during all of these power runs, as changes in ignition timing will affect carburetion settings and vice-versa.

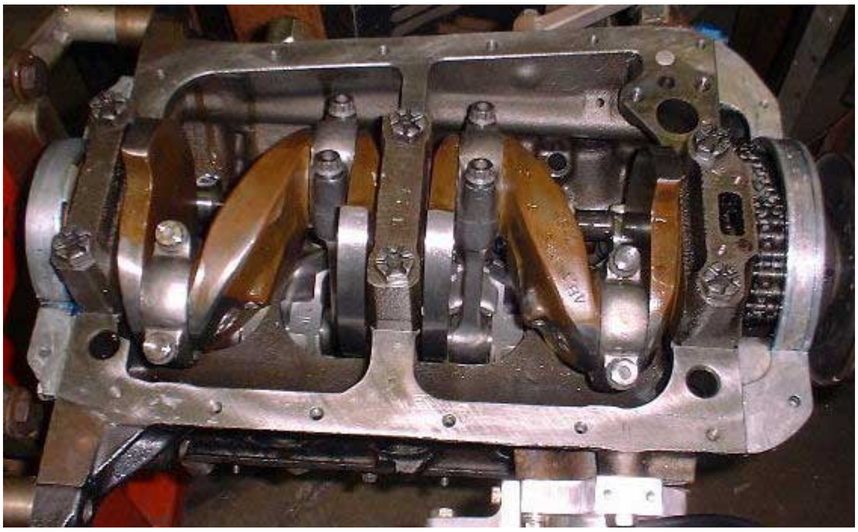
For more detailed information on each of the elements involved in power runs, please consult applicable subject categories.

As before, recordkeeping is a vital element of the testing methodology. Without it you will not be able to cross-reference your findings and get the best benefit of your testing.

2.1 - Cylinder Block Preparation

Caution: Racing is like any other hobby, if you want to be good at it you must spend time at it. The same goes for engine building. You cannot read too much, you cannot listen to much, but eventually you will have to get down to brass tacks and put that motor together. A good friend of mine who builds a lot of small block Chevrolets put it quite well when he said, **"The Cheapest way to Build a Really Good Motor is to do it Right the First Time".** The recommendations that I give below are just that, recommendations. There is no express or implied guarantee of performance.

Now it has to be said, that it would be really neat to just wave a magic wand and end up with that 1000 cc killer motor you always wanted, but that is simply not possible. Even more to the point, while it is possible to build such a motor, unless you are doing all-out racing it may be better to try for something more conservative. Now that I have gotten my reservations out of the way, I am hopeful of working through a complete engine buildup for an all-out racing motor. Along the way I will suggest alternatives for something more conservative as well.

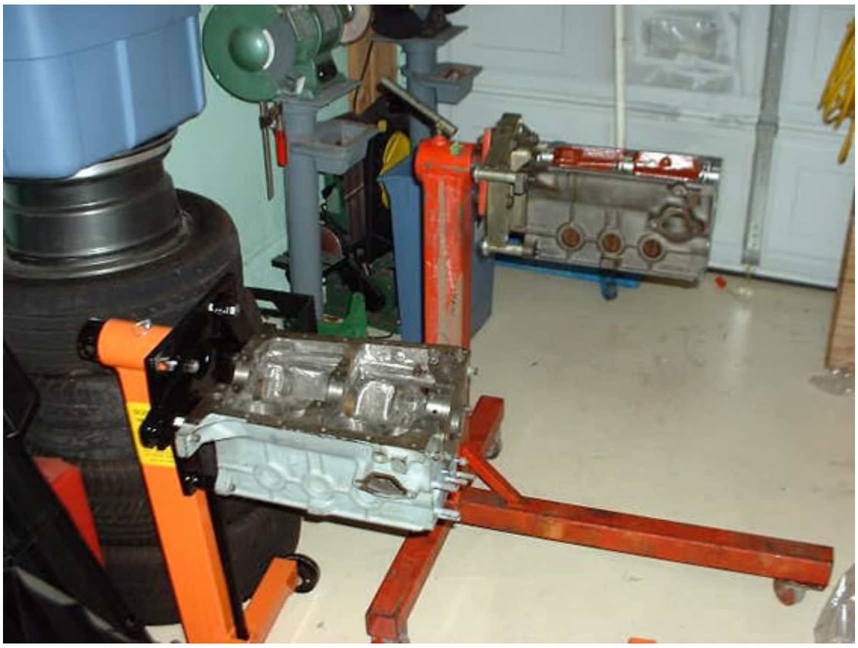


Bottom end of motor that went to Europe in 2001

Note: You should recognize some of the components. This is the bottom end of the motor that is currently in the car in Europe. As you read on you will see that we have changed our mind as to what the preferred way of doing this is. This motor incorporated prepared 850 connecting rods. The current motors have Scuderia Topolino rods, with special ARP2000 connecting rod bolts . We are continuing to use Abarth billet crankshafts, as long as they last and/or can be found.

2.2 - Block Preparation

I cannot stress strongly enough how important this first step is. The largest single item is the block. Unless you want to build an engine that is historically correct for an Abarth 750 GT, or the like, I would stay away from using a 600 block. For 1000cc motors these blocks require a great deal of work. There are two worthwhile alternatives, either an 850/OT1000 block or one of the 903/A112-1050 blocks. (For those wanting a real shortcut for a street motor, just buy a 70 hp A112 1050 motor and put it in your 600 and it will transform your car. This is still not a straight swap, but the work is quite easily done. The clutch end of the cranshaft will have to be drilled for a bronze input shaft bushing. This is not a hand drill operation and will require the removal of the crankshaft from the block. You can also start off this way and then if you want more performance, upgrade for more horsepower later.)



Two new blocks being prepared

The 817/843/OT1000 family of blocks are what I refer to as the "short deck" variety of block, whereas the 903 and 1050 are "long deck" blocks. The difference being that the long deck blocks are about 4-5 millimeters taller. Other than this the blocks a dimensionally similar, with the exception of the main bearing bore. The 817/843/903 use a smaller diameter main bearing bore, as compared to the 982 and 1050 blocks.

Before proceeding any further, clean the block in a parts washer and then have the block thoroughly magnafluxed for cracks. 1050 blocks are particularly prone to cracks in the front surface of the block, between the front main bearing and front cam bearing bores, and all blocks can suffer block cracks around the center main bearing web. Doing this now will save a great deal of wasted work. If the block is at all doubtful, find another one.

Check how far the block has already been bored . I consider the maximum safe cylinder diameter is 66.4mm or so for all but the A112a2000 (A112 70HP) block, which can be bored to 68.00mm. If the block is already to 1mm over standard, consider a different block. There are many blocks out there that are still standard, and these provide a good bases for development.

Note: US machine shops will probably measure blocks in thousands of an inch, so a good formula is that 1mm equals very close to .040 inch. Therefore, a .25mm larger bore is the same as .010 thousands of an inch.

Next determine if the deck of the block has been decked and is flat. If the top of the block has been previously machined, you may find that a standard connecting rod/piston combination will stick out above the block slightly. This is not the end of the world, but just means more work. If it is not flat, but at close to standard height, do not worry about it for now. The standard height for a 817/843/1000OT block is approx. 172mm, whereas a 903/965/A112A1 or A112A200 block will be around 177.8mm in height.

Now determine if the deck is parallel to the crankshaft bearing bores, the cylinders are at 90 degrees to the top of the block, and the front and rear block surfaces are precisely at 90 degree to the crankshaft bore. It is important that all of these dimensions are exact if we are to extract the maximum amount of horsepower out of the engine. If any of these dimensions are out, bring them back into specification. If the ends of the block are not square, and are out by more than .005 thousands of an inch, find another block.

Note: Do not assume that the factory manufactured the block correctly. Remember that this was a production motor and the tolerances for a race prepared motor are much tighter.

Next, knock out all of the freeze plugs (sometimes known as welsh plugs) from the water jackets and the one at the end of the camshaft gallery. These may look like they are OK from the outside, but as they are in contact with the cooling water on the inside, you may find considerable erosion. Put in new ones. You will also find small plugs at the end of the oil galleries and these should also be removed. It is likely that you will want the block hot-dipped, unless you do not plan on replacing the camshaft bearings. These will just disappear in the hot tank process. The oil galley plugs must be removed for cleaning no matter what. Further, remove all bolts, studs etc. so that the machine shop gets a totally naked block, as this means you will get it back with all of the threaded holes thoroughly cleaned. This will save the machine shop time and you money.

Have the machine shop check the block for main bearing saddle alignment and dimension, otherwise you may never get a good bearing fit. The 1000OT and A112 blocks register the main caps with either dowel pins or hollow dowels. I think doweling is a really good idea.

Do not automatically assume that a block needs to be line-honed, but if it appears that the block does, then this is a warning flag that it has had a rough life. Have another good look at this block before you go further. Once this is done, have the machine shop double check the ends of the block for squareness.

The next step is to examine and clean every thread in the block. This means running a tap, or better yet a “thread chaser” down each threaded hole and then blowing it out so that all of the debris is removed. Only then will you be absolutely sure that the fastener will properly torque to the required specification. I have found that many older blocks require some threads to be repaired with helicoils. I no longer take any chances with the head bold threads, if they look at all suspect, they get a helicoil inserted. Inserting helicoils is a precision job. DO NOT attempt to do this with a hand drill, as the threads will not be straight. Later on in this article I will talk about head bolts versus head studs, and you will understand exactly why this is so important.

Two different doweling methods.

Your machine shop should also be able to pressure test the block for you. You will have to make sure that all of the freeze plugs and oil galley plugs have been replaced. If the block passes the water and oil pressure tests, then you can proceed.

Next, the machine shop will need the pistons that you plan to use. Follow the piston manufacturer's instructions for piston fit, but a good rule of thumb is .001-.0015 (0.025-0.038mm) for each inch (25mm) of piston diameter. Thus a 65mm piston would have about .0025-.003 (0.063-0.076mm) of piston to cylinder wall clearance. (Max. recommended clearance .0035" [0.090mm]) Any honing of the block should be done with a deck plate bolted in place and torqued to the required head bolt torque. Cylinders will distort slightly and using such a plate insures that when the head is torqued down the bores will be round. Generally, bores should not have more than .0005 taper to them. To get accurate, straight cylinder bores requires the use of a Sunnen honing machine, or similar equipment. I do not recommend hand honing, except as a last resort and then only if you are well experienced at doing so. All too often, with hand honing, the hone spends more time in the center of the stroke and you end up with a "barrel" shaped cylinder. This makes ring seating very difficult and is very hard on piston ring lands, as the rings move horizontally during each piston stroke, very quickly wearing out the piston ring land and generating unwanted friction and heat.

Note: The dimension printed on the piston box usually denote the bore size required in the block for the enclosed pistons to meet specification. I say USUALLY. Do not leave this to chance. Whenever I have a block bored/honed, I ALWAYS have the pistons available for the machinist, so that he can personally check the size and fit.

Next, you will need to temporarily assemble certain components into the block to double check critical dimension. After making sure that the block is clean and dry install the crankshaft with the three main bearings and thrust bearings in the block. Install these dry, as at this point we are not assembling the motor yet, only checking dimensions. Lay the crankshaft in the block. Then place a dial indicator against one of the counterweights of the crankshaft and test for thrust bearing end play. I like an end play reading around 0.003 0.005 inch (0.076-0.127mm). Even clearance up to 0.010 inch (0.245mm) is probably OK, but you will have to double check other clearances more carefully.

2.3 - Lubrication System

The standard configuration for Fiat blocks is an oil pump within a wet sump, attached to the bottom of the block. This oil pump feeds oil to the various components of the block (and head which will be discussed later).

One problem associated with 817/843 and some early 903 blocks is the implementation of a partial oil filtration system, using a centrifugal filter in the front pulley. This implementation also did not have a direct oil supply to the center main bearing. Abarth realized this problem and in all of the Abarth derived blocks this was changed. I also recommend the following:

First, ANY orifices that have sharp edges on them should be radiused. Normally after oil is picked up by the pump, it goes to the pressure relief valve in the block. This area needs some serious cleaning up and blending to make sure that we get good oil flow. (1972 or later 903/A112 engines have the revised oiling system already and also have a filter mounted to the block.) For a race motor I recommend that you have a new crankshaft pulley machined from steel or aluminum. The pressed metal ones that are silver soldered have been known to come apart.

Follow the instructions below for reverse engine oil flow and pressurized center main bearing on earlier 843/OT1000 blocks:

1. Drill a 0.187 inch (4.5mm) diameter hole in the center main bearing saddle (use a center main bearing with the requisite hole to locate the position) on an angle to intersect with the bottom of the left main cap bolt hole. You do not have to drill very far, perhaps just a quarter of an inch or so.

This is the 3/16th hole going to the bottom of the main cap hole from the center main.

2. Remove and discard the oil galley plug from the outside of the block. (You should have already done so by now anyway for cleaning purposes) and use a long, straight 0.187 inch (4.5mm) drill bit, to drill an oil passage to intersect with the previously drilled hole. (I recommend that you do this on a vertical mill) If you allow the drill to go off-center, then you will get to start over with another block.(In my engines, as I use the rear-most boss on the oil galley for return pressurized oil supply, I tap this center entry in the oil galley for a 1/8th pipe fitting for an electric oil pressure sender.)

3. Carefully enlarge the outer hole at the oil gallery and tap for a fitting for the oil pressure feed line for the block. Remember this is cast iron and it does crack. A "-10" Aeroquip fitting with a pipe fitting on the other end is fine. Make sure that you install this fitting with a sealer or Teflon tape. This is where oil goes in to pressurize the entire engine.

AN-12 Fitting installed in bypass hole
Be careful when installing. Use Teflon tape and do not over tighten.

4. Next turn the block upside down and find the hole where the oil pump outputs oil. Enlarge this hole for its full depth to 10mm. (The same goes for the gasket).

5. Next to this hole is another hole (bypass return hole) that must be permanently blocked.

6. Fabricate a main cap blanking plate for the #1 main cap and install with gasket.

7. If the crankshaft you plan to use came from a motor with a centrifugal oil filter, plug the hole in the snout of the crankshaft. Again, I recommend drilling and tapping for a Allen head plug, installed with loctite.

8. You will have already removed the pressure bypass valve in the side of the block. Inspect the oil passage and relieve/open up as required for better oil flow. Remember that this is going to be the output orifice for the oil pump.

9. Make fitting to go in place of the pressure bypass valve. Again, this will have an AN -10 or AN -12 Aeroquip fitting on one side and and a metric (20x1.5mm) thread with a flat face for an annealed copper or aluminum washer on the other side. Scuderia Topolino has available a kit to do this modification.

The oil now is picked up by the pump and exits the block via the special fitting in the bypass orifice in the block.

2.4 - Oils and Additives

Recent changes in oil formulations have proven to be troublesome for older vehicle is general, and for cars with flat tappet engines in particular. Up until a year or so ago, almost all engine oils had small amounts of zinc and phosphorous as part of their oil chemistry. The typical amount would have been 1200 ppm (parts per million). Manufacturers have asked that the level of phosphorous be reduced, as it has a negative effect on the longevity of catalytic converters. Oil companies have responded by cutting these additives by 75%, as they are also expensive ingredients in oil-formulation chemistry.

Both phosphorous and zinc are specifically indicated friction modifiers, particularly applicable to "sliding interfaces". These would include the following interface junctions:

- Cam lobe/lifter,
- push rod/lifter,
- push rod/adjuster,
- rocker arm/rocker shaft,
- rocker tip/valve tip.

Most engine oils have had the level of these vital additives reduced to 400 ppm. The exceptions to this rule are as follows:

Manufacturer	Oil Type	Synthetic/Organic	Weight	Phos. PPM	Zink PPM	
Castrol	Syntec	Synthetic	5W-40	1000ppm		Vehicle
Castrol	Syntec Classic	Synthetic	20W-50	1200ppm		Vehicle
Castrol	TWS Motorsport	Synthetic	10W-60	1000ppm		Vehicle
Castrol	BMW Long-Life	Synthetic	5W-30	995ppm		Vehicle
Castrol	Power RS GPS	Synthetic	10W-30 10W-40 20W-50	1000ppm		Motorcycle
Castrol	Power RS R4	Synthetic	5W-40 10W-50	1200ppm		Motorcycle
Brad Penn Oils	Formerly Kendall	Synthetic	Various	860ppm		Vehicle
Swepco	306					

Royal Purple	Max Cycle	Synthetic	20W-50	1200ppm		Motorcycle
Amsoil	Harley V-Twin	Synthetic	20W-50	1200ppm		Motorcycle
Cosworth	Racing Oil	Synthetic		1150ppm	1250ppm	Vehicle
Shell	Rotella T CI4	Organic		1300ppm	1400ppm	Diesel
Pennzoil	Racing Oil	Synthetic	20W-50	1800ppm	1950ppm	Vehicle
Quaker State	Q Racing	Synthetic		1800ppm	2000ppm	Vehicle
Valvoline	VR1	Synthetic	20W-50	1200pp,	1300ppm	Vehicle
Valvoline	Racing Oil	Synthetic		1200ppm	1200ppm	Vehicle
Royal Purple	Racing Oil 21	Synthetic	5W-30	1130ppm	1961ppm	Vehicle
Royal Purple	Racing Oil 41	Synthetic	10W-40	1171ppm	1901ppm	Vehicle
Redline Oils		Synthetic	10W-40	1371ppm	1350ppm	Vehicle
Redline Oils		Synthetic	10W-30	1340ppm	1407ppm	Vehicle
Redline Oils		Synthetic	5W-30	1419ppm	1421ppm	Vehicle
Mobile 1		Synthetic		1223ppm	1376ppm	Vehicle
Joe Gibbs*		Synthetic	Various	6000ppm	6000ppm	Vehicle

Joe Gibbs Racing oil also has increased levels of Sulphur.

This was the information as of the middle of Jan 2008. Obviously almost all of the racing oils had increased levels of zinc and phosphorous, vital to the proper running of an Abarth motor. You should check on the brand that you are using, to make sure that it has not been reformulated. As a general rule, somewhere between 1000 -1200 PPM would be a good minimum number for both elements.

The follow-on question is then what weight of oil should I be using. The key to oil numbers is the second number. So, in 10W-40 weight oil, the "40" part indicates that this oil has a viscosity rating of 40 weight oil at 212 deg .F (100 deg. C) however it has a consistency of 10W oil. Now the question is, just what rating do I need? Do I automatically go for 10W-60? Well, maybe !!

The key is temperature and oil pressure. At whatever you oil temperature happens to be, you should be able to maintain 75 PSI (5 Bar) of oil pressure at 6500 RPM (this happens to be Ferrari's formula for high performance vehicles). If you can do this with 10W-30 then good. If not, then move up to 10W-40, or 10W-50 etc. In other words, depending on the state of the motor, you can tailor your oil grade to maintain a pressure level that is indicated. Using a grade of oil with a higher viscosity rating, beyond this, will only cost you horsepower. Now in the top leagues of auto racing they may use 0W oil for qualifying, but this usually means a motor that was broken in on the dyno on a higher grade, and then they only care if it last 3 laps.

For those who want to know more about oils and how they perform, please read this report. Draw your own conclusions. Yes, the report was sponsored by Amsoil, but the results really do speak for themselves.

http://www.syntheticoilnlubes.com/pdf/g2156.pdf

At Scuderia Topolino we break engines in on Shell Rotella T non-synthetic oil. This is for two reasons. First it has a good zink/phosphorous additive mix to protect the camshaft during break-in. Second, it is not "super slippery", and therefore rings will seat quickly in the cylinder bores. After this we change to Redline Synthetic Racing Oil or Amsoil 10-50 Motorcycle oil. Yes, motorcycle oil, as it has the higher level of zinc and phosphorous. Alternatively, Joe Gibbs oil is probably very good, but the price my put some people off.

Amsoil has also earned a very good reputation with their engine oils in motorcycle gearboxes with a rating equivalent to 80/90W gear oil. As it has no Extra Pressure (EP) friction modifiers, it may be a good solution for Fiat transaxles as well. This oil may work well with the standard synchro rings and the bevel gear pinion used in this transaxle. As it has superior gear wear characteristics, Scuderia Topolino hopes to test the oil on our dyno, and in the transaxle during the 2008 season, in preparation for a major effort in 2009.

Reciprocating Components and Engine Balancing

3.1 Connecting Rods

There is a fair amount of mystique associated with connecting rods, but one thing is certain, they are one of the most important components of an internal combustion engine. There are many opinions about what makes a good connecting rod, and I for one am not sufficiently versed in metallurgy to go about designing one. What I can do however is look to those companies that have a good track record and try and see what they do well.

Companies like Saenz, Pankl, Carillo, Crower and many others have been making connecting rods for many years, with great success. So what is so different about a specialty rod, as compared to a standard production rod. Three words describe the difference.

Fasteners - Fit - Finish.

Fasteners - In my view the most important aspect of any connecting rod is the bolts. For connecting rods that are to be exposed to the stresses of competition, only the absolute best will do. Whether you purchase the bolts you use with the rods, or buy them from after-market suppliers like SPS or ARP, the money spend on better connecting rod bolts will pay off every time.

Standard Fiat bolts are fine for normal road applications. They rate at about 90,000 to 100,000 PSI. For an engine that is going to see 8000 RPM or more this just will not do. After-market bolts start at about 160,000 PSI and go up from there. The ultimate strength of the bolt will be determined by both the metallurgy and the size of the bolt.

Due to size restrictions, the standard bolt for Fiat rods is a fine pitch threaded 8mm unit. If you are using standard connecting rods, use the 850/903/1050 units. (Remember - The 903 rods are 2mm longer) These can be cleaned up quite and then install a set of replacement rod bolts rated at 160,000 PSI. If all the components are lightened and balanced, this will give a very robust installation.

Stretchgauge

If you want even better then use connecting rods from Scuderia Topolino, Carillo, Crower or Pankl. All offer uprated bolts for their connecting rods ranging from 180,000 to 285,000 PSI. In all cases you will note that the shank of the bolt is cut down, so that it is slightly less than the original diameter of the bolt. There is a reason for this. All bolts must be able to stretch - but only a predetermined amount. Without stretch the bolt will not properly clamp the cap to the body of the rod. If the bolt stretches too much, or fails to return to its original length when loosened, DISCARD IT. It is also important "where" the bolt stretches. The reason for the shank reduction is this is the area where the stretch is supposed to take place. The diameter of the bolt in this area will be slightly less than the root, or bottom, of the threads. If this shank reduction were not there, then the bolt would have a tendency to stretch in the thread area. This would not be a good idea. In general an 8mm bolt should stretch between .005-.006 thousands of an inch at or below indicated torque. Each manufacturer will have his own specifications, and it is important that their recommendation be followed.

One thing that all of the premier companies agree on is how connecting rod bolts should be torqued and to do it correctly this takes a "rod stretch gauge". When I build a racing motor, I record the free length of each bolt and note it in the build records of the motor. If not, how would you ever be able to check whether the bolt failed to return to its free length when slackened off. The bolt is then fitted, finger tight, and torqued to 80% the specified amount. Now you check for stretch. If the bolt has stretched less than .005 increase the torque on the bolt until the required stretch is achieved, but do not exceed the maximum torque recommended by the rod manufacturer.

If you are fitting after-market rod bolts to standard connecting rods, or to rods for which they were not specifically manufactured, pay close attention the fillet radius of the underside of the bolt head. In many cases with standard connecting rods you will have to radius the edge of the hole in the rod. If you do not do this then the bolt will make contact with a sharp edge in the radius area and it is almost certain that an early and catastrophic failure will occur.

A WORD OF CAUTION - You must fully lubricate the connecting rod bolt before torqueing. This includes the threads, shank and the underside of the head. If the bolts come with special assembly lubricant, USE IT. You would be amazed at the different friction coefficients of regular motor oil, synthetic motor oil and special assembly lubricants. Also the torque recommendations will be different for all three.

One reason why I personally did do not use Carillo rods, is that the H-Beam design mandated the use of a 1/4 (approx. 6mm) bolt, where as in the Scuderia Topolino design a 5/16th (approx. 8mm) bolt is utilized. Carillo defends their use of the 1/4 inch bolt on the basis that the rod/piston combination's dynamic forces are will within the design limits of a 1/4 inch bolt. Call me a skeptic, but in the connecting rod bolt department "bigger is better" in my view.

Fit – Each aftermarket connecting rod manufacturer has their own technical solution. Carillo's H-beam design has been the standard for racing engines for many years (Even the Chinese are now making a knockoff of this design). The Scuderia

Topolino connecting rod also includes an H-beam component, but this is combined with a certain aspects of the I-Beam design as well. The result is a connecting rod that has the best features of both designs and which is lighter overall..

Using a custom connecting rod allows the engine builder to vary the length as well. This freedom, along with innovative piston designs, may make impressive horsepower gains, but more important are the gains to be made in engine acceleration due to lowered rotating mass and better rod angularity. One extra consideration is that all Scuderia Topolino rods have doweled caps. This rod caps will have two hollow dowel pins to locate the cap.

Many times I am asked why I go the trouble of buying custom connecting rods. Many competitors have done quite well with the standard 850 or A112 rods, suitably reworked. I guess it depends on how much time you have and what the ultimate objective is. If you consider that you may spend as much preparing a set of standard rods, presuming you account for your own time, then buying a set may not seem so outlandish, even to the amateur competitor. By the time you do the following:

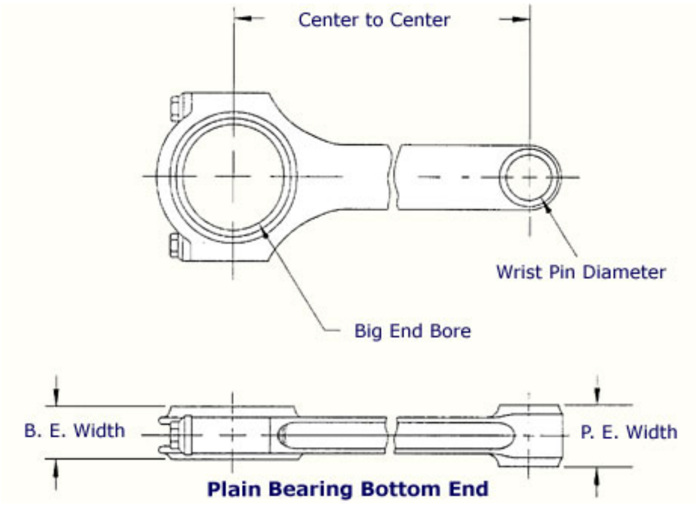
- Grind off the excess material (large blocks of metal at the big and small ends of the rod),
- Checked for axial twist and straightened if required,
- Sized or honed for bearing fit,
- Sized or bushed and honed for piston pin fit,
- Polish the rod and then have it shot peened (stress relieving after grinding); and
- Finally have the set balanced

then you may have spent more on your time, materials and outside machine shop labor than a custom set of connecting rods. A set of custom rods from Scuderia Topolino will have all of this done to specification.

Finish - In modern connecting rod technology there are two "finish" areas. One is the actual machine work quality, including proper alignment of oiling holes and making sure that the bolt holes have proper a proper edge radius. Finally, there is the overall finish of the connecting rod. While polishing really looks neat (at least to the person putting the engine together), no one else is going to see all that work, unless of course the engine disintegrates and there are lots of nice, shiny polished parts that are lying around on the track. I prefer a shot blasted finish which is the finish resulting from the stress relieving process.

The second half of the "finish" equation is surface coating. Companies like Calico Coatings now have function specific coatings, including for connecting rods they have special oil-shedding coatings that assist in getting the oil off the rods and back into the pan.

If you decide that custom rods are the way to go, then the following diagram will come in handy. All of the measurements are required in order to manufacture a custom connecting rod. One measurement not shown is whether the PE is offset from the BE. I personally do not use any offset, but if you are ordering special connecting rods, you can include this specification. In addition to these measurements, if you are building a 1050 or 982cc motor, then the width of the shank of the connecting rod will also be critical. Even if the width is no greater than a standard rod, the camshaft may still have to be relieved to prevent the rod from striking the camshaft core and the sides of lobes 1,4,5 and 8.



So why would we want to lengthen the connecting rod? Well, the short answer is of course the ever present holy grail of increased horsepower. But, just what is it that we are going to accomplish with a longer rod. Here is a short list.

1. Decrease in piston skirt side force - reducing friction and reducing parasitic losses

2. Increase in Dwell Time at TDC and BDC. This allows for better combustion control, particularly at high RPM, but from a negative perspective it may affect exhaust scavenging if the head used has a poor exhaust port.

3. More equalized piston acceleration with reference to TDC and BDC - not piston speed.

For high performance engines, where the builder has already explored better porting, valve angles, exhaust systems, camshafts etc., going to a longer rod will have more plus benefits than negative ones. Certainly longer dwell time at TDC will allow for additional time for flame front travel with high RON fuels. This will allow a small increase in static ignition advance, while still maintaining the maximum pressure crank angle on the combustion stroke. Even if everything is just right, the maximum gain that one should expect from longer rods is just around 1%. In small engines this might be 1 to 1.5 horsepower.

Connecting rod angles

Many engine builders will tell you that an optimum rod angle is 1.75. Thus, the stroke/rod length combination for the 843cc motor is pretty close to ideal. Even the 903 is pretty good, but the 982cc combination is getting pretty marginal. However the 843cc engine does have a downside. By moving the piston pin lower more side load will be produced on the piston skirt. There would be a way to optimize the situation to give the 903 and 843 engines the same high piston pin location, namely using a longer connecting rod. This give the following combinations:

- 74mm stroke - 0.995 inch (25.25mm) piston pin location - 982cc - 110mm rod length - Rod angle = 1.486
- 69mm stroke - 0.995 inch (25.25mm) piston pin location - 903cc - 112.5mm rod length - Rod angle = 1.63
- 63.5mm stroke - 0.995 inch (25.25mm) piston pin location - 843cc - 115.25mm rod length - Rod angle = 1.81

If we now look at what effect the same changes will have with the "tall deck" blocks, you will see that there is even some advantage for the 903/1050cc motor. These blocks are approx. 5.6mm taller than the short deck blocks. This means that the piston pin will be approx. that same amount lower again in the piston. Note: The 843cc motor combination would not be advisable !!!

- 74mm stroke - 1.219 inch (30.93mm) piston pin location - 982/1050cc - Rod angle = 1.486
- 69mm stroke- 1.318 inch (33.45mm) piston pin location - 903cc - Rod Angle = 1.594
- 63.5mm stroke - 1.426 inch (36.19mm) piston pin location - 843cc - Rod Angle = 1.732

Now, if we again modify the rod length to maintain a high pin position, the following results are obtained

- 74mm stroke - 0.995 inch (25.25mm) piston pin location - 982cc - 115.7mm rod length - Rod angle = 1.5636
- 69mm stroke - 0.995 inch (25.25mm) piston pin location - 903cc - 118.2mm rod length - Rod angle = 1.713
- 63.5mm stroke - 0.995 inch (25.25mm) piston pin location - 843cc - 120.9mm rod length - Rod angle = 1.90

In all cases the rod angularity is markedly reduced, which should result in some reduction of parasitic friction losses.

Quite obviously, in order to make these changes, custom connecting rods have to be used. These must of be the best possible materials and must use the highest grade connecting rod bolts, if the engines are going to survive 8000+ RPM limits. If a good quality 69mm crankshaft were available, then an interesting combination would be a 67.6mm bore with a 69mm stroke using a 118.2mm rod. This would produce a displacement of 990cc.

In an effort to "standardize" rod length configurations and to provide some economies of scale as far as manufacturing is concerned, Scuderia Topolino will make pistons for two rod lengths available, standard (110mm) and overlength (117mm). The Scuderia Topolino General Catalog lists all of the piston/rod combinations available as standard items. Of course if you need a connecting rod/piston combination that is different, please do not hesitate to ask.

> CAUTION - **Some of the rods will get quite long and only the absolute best of materials should be used**. In additional the bottom of the cylinder will likely require notching on most of these combinations and certainly on ones using a short deck block (600/850/1000OT) with a long rod.

All rods supplied by Scuderia Topolino are equipped with ARP2000 or ARPL19 connecting rod bolts. The A112 standard rod is the heaviest at 443 gram. By comparison, an Abarth 1000SP connecting rod (also used in the TCR motors) is lighter at 365 gram. Even after some grinding and polishing, it is still the heaviest. For its size, the 1000SP rod is surprizingly light. Its design is I-beam, only much stouter. The H-beam rods from Carillo weigh in at 384 gram. The Scuderia Topolino connecting rod, with a combination of H and I beam characteristics weighs 350 grams in standard form and less than 330 grams in the "narrow" version.* Finally, you could decide for a titanium connecting rod and this would weight approx. 30% less than the steel equivalent or 220 grams.

3.2 Pistons

Piston metallurgy. There are basically four types of forged pistons on the market today. Three of these are made of an Aluminum/Silicon Alloy, whereas the forth is made of 2618 Aluminium.

- 2618 Aluminum - No Silicon
- Hypoeutectitic - Less than 12% Silicon (typically 9 %)
- Eutectic - 12% Silicon
- Hypereutectic - More than 12% Silicon

Basically, silicon adds "wearability" to the alloy and in street applications would be a great plus. One drawback is that it makes the aluminium somewhat brittle. For low RPM motors this is not a problem.

For high performance applications, almost all pistons, including those supplied by Scuderia Topolino, are made from 2618 Aluminium or similar, but without silicon. This means that they will expand somewhat more than an aluminium/silicon alloy piston, but they will be significantly stronger.

While I have not specifically addressed cast pistons, the same considerations would apply, plus the cast piston will have a slightly less dense structure and therefore may not be as strong. Although some would claim that the expansion characteristics of the cast aluminum material are also less than a similar sized forged piston, for ultimate strength a forged piston is recommended

Ring Configurations - All Scuderia Topolino pistons are manufactured with the following ring pack. We find that it provides a good combination of wall tension, seating, wear resistance and cost.

- Top Ring - 1.2 mm Chrome
- 2nd ring - 1.2mm Nodular Iron
- Oil control ring - 2.8mm multiple segment oil control ring.

Engine Displ.	Bore mm	Stroke mm	Rod length mm	Block Height mm	Main Brg Bore Diam mm	Piston Pin Diam mm	Comp. Height mm	Comp Height Inch	3-Ring Pin -Oil groove distance mm	3 Ring Pin -Oil groove distance inch	Rod/ Stroke Ratio	Vehicle/Engine
1049.3	67.20	74	110	177.850	57.53	18	30.85	1.215	8.14	0.321	1.49	A112 - 1074, 1077, 1077B, 1077C, 1091
981.7	65.00	74	110	172.350	57.53	18	25.35	0.998	2.64	0.104	1.49	1000OT - 1074, 1077, 1077B, 1077C, 1091
964.2	67.20	68	112	177.850	54.42	20	31.85	1.254	8.14	0.321	1.65	A112 965cc - 1074, 1077, 1077B, 1077C, 1091
902.1	65.00	68	112	177.850	54.42	20	31.85	1.254	8.14	0.321	1.65	Fiat 903cc - 1077, 1077B, 1077C
842.4	65.00	63.5	110	172.350	54.42	18	30.60	1.205	7.89	0.311	1.73	Fiat 843cc - 1077, 1077B, 1077C
846.3	62.50	69	110	173.000	54.42	18	28.50	1.122	5.79	0.228	1.59	Abarth 850 TC - 1076

STD 3 RING PACKAGE		
Top land	4.00	0.158
Top ring groove	1.22	0.048
Second land	2.72	0.107
Second ring	1.22	0.048
Oil land	2.03	0.080
Oil Ring	2.52	0.099
Total Space	**13.710**	**0.540**

2 RING PACKAGE		
Top Land	4.00	0.158
Top ring groove	1.22	0.048
Oil land	3.00	0.118
Oil ring groove	2.52	0.099
	10.740	**0.423**

Note: All pistons produced after Feb 2007 will also come standard with "peripheral gas porting" to aid on top ring sealing.

It is possible to use a 2-ring configuration. This would be purely a "competition only" setup. The overall size of the ring pack would be reduced by about 3mm, allowing the piston pin to be moved up an equal amount. This would be particularly helpful for 74mm stroke motors. The following results would be obtained.

- 843 block motor - 74mm stroke 113mm long rod for a rod angle of 1.527
- 903/1050 block motor - 74mm stroke 121mm long rod for a rod angle of 1.64

See explanation of rod angle issues under the connecting rod section.

Piston Layout Combinations – Scuderia Topolino pistons are now in their 3rd design generation. The current piston has a reduced skirt length, thinner skirt sections, tapered forged piston pin, and the top ring groove is now gas-ported for improved top ring sealing. Weights of all of these new pistons have been reduced by an average of 15%. As an example, the 3rd generation piston with pin now weights 218 gram (compared to 279 gram for the 2nd generation piston).

Scuderia Topolino now stock three different piston dome configurations. There are:

1. Flat Top. These pistons have approx. 10.5:1 compression with a standard Fiat/A112 cylinder head. These is perhaps 0.5 point higher compression than the standard cast A112 piston, which is slightly dished.

2. Small Dome with Valve Reliefs. These pistons have approx. 12:1 compression with a standard Fiat/A112 cylinder head.

3. Large Dome with NO Valve Reliefs. These pistons have more than 13:1 compression with a standard Fiat/A112 cylinder head. Caution: Depending on the type of camshaft used, these

pistons may produce more higher dynamic compression than desired.

I am sure that someone will ask why we did not make it a "full slipper style" piston. The overall goal was to reduce the weight of the pistons with pins to 250 grams or less. We have more than achieved this without resorting to a full slipper design. These were the considerations.

1. Because of the small diameter of the piston, in order to achieve a full slipper design, the engineers felt that the pin bosses would become too small to support the RPM levels expected from the pistons. At 8500 RPM the piston speed is around 68 ft/sec (21 m/sec), with 3900 G’s of force at TDC at the same RPM.

2. It was determined that not much additional weight advantage could be gained from a full slipper design, as the amount of aluminum material saved by repositioning the pin webs would me minimal.

3. The piston would retain better dimensional stability with a semi-slipper design with more predictable expansion characteristics. This would promote better ring stability and cylinder sealing.

Here are the weight comparisons between 2nd and 3rd generation pistons.

Abarth/Fiat 10.5:1, 3.5mm dome with valve pockets - 2nd Gen. 279 gram - ***3rd Gen. 218 gram***

Abarth/Fiat 13:1, 6mm dome, no valve pockets - 2nd Gen. 275 gram - ***3rd Gen. 226 gram***

Abarth TCR 10.5:1, 2nd Gen. 264 gram - ***3rd Gen. 239 gram***

Abarth TCR 12.5:1, 2nd Gen. 274 gram - ***3rd Gen. 249 gram***

Anti-Friction Coatings - Developments in anti-friction coatings have come a long way in just a few years. The coatings available today certainly provide a measurable advantage when used in a race motor. At the moment I specify coating applied by Calico Coatings. This plasma sprayed coating can be used on piston skirts, main and rod bearings, rocker shafts, lifters etc. They also have oil shedding coatings for use on reciprocating parts such as crankshafts and connecting rods.

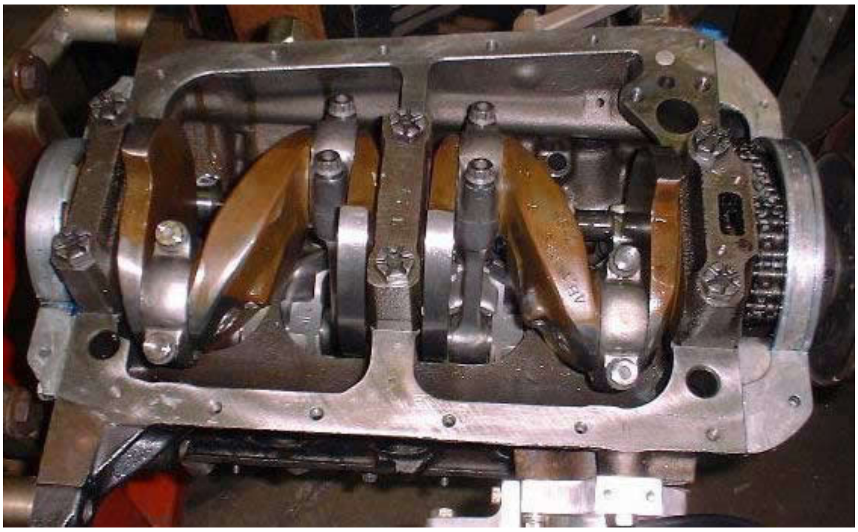
In January 2007 we started testing with DLC type coating by Sorevi (Bakaert). This company has developed several proprietary coatings by the trade name CAVIDUR. These can be applied to various engine "friction interfaces" such as rocker shafts, camshafts, lifters, finger followers and quite interestingly pistons skirts. I have also done a complete gearbox to see how the material stands up high levels of both friction and shock loading. Certainly on rotating shafts, collars and plain bearings, the material would appear to have distinct anti-friction advantages. We examined a set of lifters, camshaft, pistons and piston pins, after a full season of races and found virtually nil wear to these items.

Flame Propagation – In order to get the maximum combustion pressure from the engine, measures must be taken to insure that there is complete burning of fuel. Just because an engine has 13.5:1 computed compression does note mean that if will produce the most power, unless it can burn the fuel efficiently. A good general rule of thumb is that the higher the dome on the piston, the more difficult it is going to be to burn all of the fuel in each combustion event. In essence, the fuel charge will be distributed on either side of the dome and the spark plug is on one side of the dome. Therefore an engine running with 12.5:1 compression, and completely burning all of the fuel in the cylinder in each combustion event, may in fact produce better horsepower. About the only way to tell is to run two engines, back to back, on an accurate engine dynamometer.

3.3 Balancing Considerations

Rotational Assembly Imbalance Tolerance - and its affect on engine reliability

No one involved in the preparation and building of race engines would argue the importance of balancing. Improper, or better yet inaccurate, balancing may have far reaching implications on both the performance and the useful life expectancy of the engine in question. Balance, or the lack of it, is more than simply a matter of matching individual components. Perhaps more importantly, it is the balance tolerance limit of the entire assembly that is of vital importance. As I will explain later, a small static weight difference can have a profound negative effect at higher RPMs.



In dealing with historic engines, like the Fiat 600 engine and it many derivatives, one has to come to grips with the fact that this engine was designed well over 50 years ago, and probably to a much different design and performance criteria than used in historic motor sport competition today. After all, it is a fairly plain, garden variety 3 main bearing, OHV motor. For this very reason balance tolerance limits may be much more critical here than in say a modern, five main bearing engine such as the Duratec 2.0 litre 4-cylinder. Yet amazingly, at high levels of tune, it is not impossible to get power outputs of 100 HP/Litre from these early Fiat block designs. Over the years I have seen Fiat blocks fail in two general ways that could be attributed to "excessive balance tolerance".

- Center Main Bearing Support Failure - Here the center main bearing portion of the cast iron block literally breaks away from the block. It is sometimes difficult to determine cause/effect with this failure if it is determined that the block is broken after the engine has had a major mechanical failure (thrown connecting rod). One might argue that the rod broke first, and then the block was damaged when the block was struck by the rod. I believe the cause/effect sequence may be different. I believe that due to excessive imbalance tolerance the center main was pulled from the block and then due to crankshaft flexing further damage is inevitable.
- Cracked Front Block Surface - The failure I have seen on several A112 blocks where there is a fracture between the front main bearing bore and the front cam bearing bore. I believe this to be entirely an excessive balance tolerance problem

So, what is involved in achieving a minimum balance tolerance. First we have to understand the nature of imbalance, and how the affects of this imbalance are manifested, and then work backwards from there. Imbalance is generally caused by some type of "uneven distribution of weight". In the case of a race engine, imbalance has to deal with rotational as well as reciprocating elements. This uneven distribution of weight can be cause by several factors including

- Improper manufacturing and installation tolerances
- Metal inconsistencies (forgings or castings)
- Fasteners
- Trapped oil
- Overall component and assembly weight
- Damping (or lack thereof)

First it is important to understand the balancing procedure. Almost all dynamic balancing is done at rotational speeds between 200 and 1200 RPM. This is far from the 7000-9000 RPM that these engines are likely to see in competition. Therefore, any small imbalance at 1200 RPM will be much more pronounced at 9000 RPM.

Second, many "assume" that the crankshaft, the major engine rotational component, is stiff enough to resist bending caused by compression loads imposed on it, given that it is adequately supported in a crankcase of sufficient "beam stiffness". These are very large assumptions when it comes the 3-main bearing Fiat blocks, so it may be prudent to keep the balance tolerance as low as possible.

Let’s start by defining the types of mechanical issues that can affect balance. Most engine builders would concern themselves with the rotational balancing of the crankshaft. However, this process is more complicated that first meets the eye. The crankshaft is made either from cast iron, nodular iron, steel casting, forging or billet, in corresponding order of resistance to bending. Given the irregular shape of the crankshaft, as well as the other assembled components that make up a complete crankshaft assembly, the job of balancing is made all the more difficult. Certainly fully machined forgings or billets are the most preferable, as they should have equal dimensions for the various webs and counterweights associated with the crankshaft itself. Cast crankshafts, other than being more flexible than their forged or billet steel counterparts, may also have casting inconsistencies, causing differing dimensional characteristics and associated imbalance. Certainly it is helpful if these rotational dimensions are standardized, before any balancing is done.

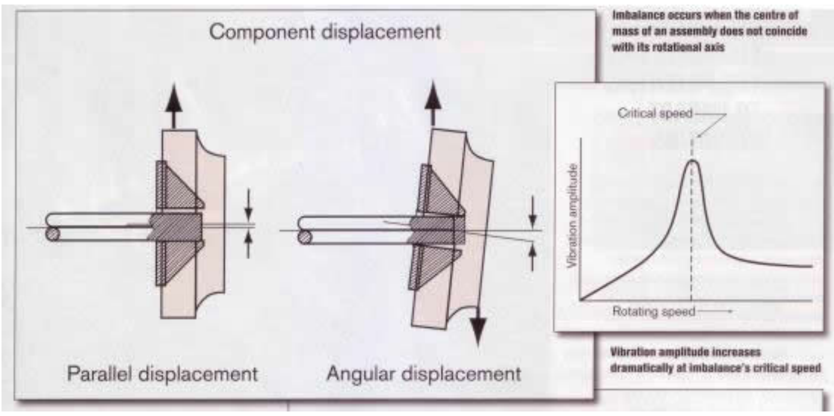
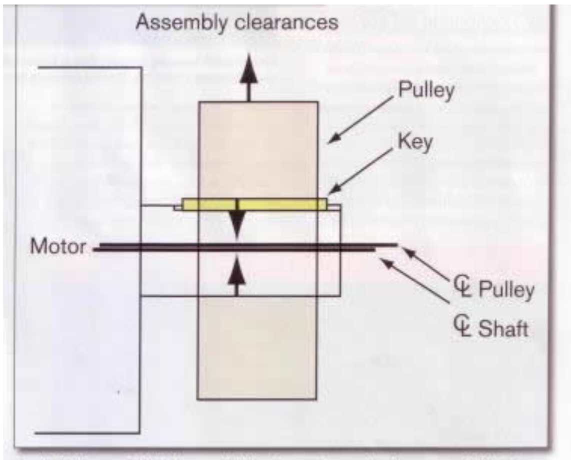


Illustration attributed to Steve Smith from Racecar Engineering article "Vibration Free"

It is also important to inspect all of the ancillary attachments to the crankshaft to make sure that they are concentric. The tin front pulley of the Fiat 600, if bent or out of round should be discarded, as it may have both parallel and/or angular displacement, contributing to overall crankshaft assembly imbalance. Better yet, it should be replaced with either a light alloy or steel machined pulley where the outside diameter is concentric with the central bore, and the front surface of the pulley at 90 degree to the bore centerline. With the pulley centerline the same as the crankshaft snout centerline, this will insure the minimum of parallel and angular displacement. The keyway bore should not have any excessive wear or clearance, as this will allow the pulley to rotate in relation to the keyway. The same argument would also hold true for the flywheel and pressure plate assembly.

There is an "order" of balancing that I recommend, but first we need to look at what type of imbalance tolerance we are willing to accept. It is simply not enough to tell the machine shop to "balance the assembly".



Acceptable tolerances of fit or balance on indvidual components can add up to an unacceptable imbalance on the complete assembly if all biases are orientated in the same direction

We can illustrate this by first balancing the four associated connecting rods (make sure the bearings are installed and rod bolts seated). This requires a scale, accurate to 0.1 gram or better, and a connecting rod weighing fixture capable of supporting the small end of the connecting rod so that the rod centerline is level and square to the top of the scale platform. The large end of the rod should be placed in the center of the scale platform, and as the rod is a fixed length, the small end will be the same distance away. In this way we can match the connecting rods so that the big end weights are within 0.1 gram of each other. How important is this measurement? According to Steve Smith, of Vibration Free" in England, "If one rod was 0.1 gram heavier than the other three, this would produce a force of 5.5 lbs (2.5 kg) at 6000 RPM for an engine with a 70mm stroke". As Abarth motors of 982cc displacement have an even large stroke (74mm), and will achieve engine revolutions of up to 9000 RPM, for a 0.1 gram imbalance the amount of force would be more than 13 lbs (5.9 kg). An assembly that was out by 5 grams would generate a force of 659 lbs (299 kg) at 9000 RPM.

Engine designers calculate the engine bearing surface to be able to support a certain load force at a given oil pressure. If this "loading limit" is exceeded, then meta-to-metal contact occurs and a catastrophic failure is inevitable!!

Obviously this is just one component out of many that may affect the balance of a complete crankshaft assembly. Once the big ends of the rods have been balanced (probably to 0.1gram or less) then the overall weight of the rods must be measured and recorded. It is helpful to temporarily number the four rods with a marking pen. Next weigh each piston assembly (complete with rings pins and clips) at match them to the respective rods so that the combinations are as close to the same as possible. Only them can you remove weight from the small end of the rods that are heavy, as compared to the lightest combination of rod/piston. Once you have them all the same, then you can permanently mark the rod/piston combinations for the cylinder where they will be installed.

Every component that rotates or oscillates has an inbuilt resonant, or critical frequency. Crankshafts may in fact operate at higher critical speed than their natural frequency and will need to be balanced in several planes along their length. As any imbalance is amplified at/near the critical speed of any component, balancing at multiple planes is essential. It is this amplified imbalance at different locations along the crankshaft that causes bending and flex, resulting increased parasitic losses and if too great engine failure.

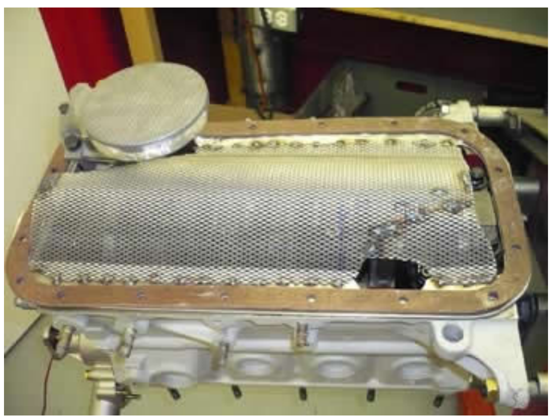
The order of things should be as follows:

- Crankshaft only
- Add front pulley and nut
- Add flywheel
- Add pressure plate

Now you can replace any one of the ancilliary items attached to the crankshaft without having to rebalance the crankshaft itself.

There is one more factor that can be lessened, in terms of imbalance tolerance, and that is the effect of any oil clinging to the rotating and reciprocating assemblies. Engines running at high RPMs create what an "internal whirlwind" of air and oil, around the crankshaft. This is referred as "oil roping". This may add to any other imbalance that exists. This can be minimized by the following methods.

1. Installing a windage tray - This is a full separation between the rotating crankshaft and the oil in the pan. This will keep the oil from contacting the crankshaft during cornering, acceleration and deceleration.

   <u>Are oil pan windage trays important in preventing oil aeration and why?</u> The answer is both yes and no. All engine oils have a level of dissipated air, as part of their makeup. At one atmosphere (14.7PSI) it is generally accepted that there is 9% by volume of dissolved air in mineral oil (Bunsen's Coefficient). Therehave been several papers written about the behavior of oil within the sump of a wet-sump lubricated engine (dry sump engines have a different set of circumstances). From the various studies, on the effect of windage tray design on surface aeration of oil, it would appear that the even at 50 PSI (3.4 times higher than atmospheric) the percentage of allowable air entrainment in oil may be on the order of 50% for rotating assemblies such as

crankshafts, camshafts and counterbalance shafts. That is not to say that 50% should be the design goal. As the percentage of entrained air in oil tracks both RPM and oil temperature, it is important to control free air in oil to a workable amount, as concentration higher than 50% will cause main failures and concentrations of 30% is considered by many to be the upper limit for connecting rod bearings due to interrupted nature of the oil feed.

From empirical testing at Ford Motor Co. the effects of oil droplets, flung from the rotating assemblies, on the free air percentage of oil in the sump (prior to entering the pump) is inconclusive. Yes, at higher RPM these high speed droplets did cause the oil to foam on the surface of the oil in the sump, but this appeared to have little effect on the entrained air percentage of oil entering the pump. Adding a windage tray to a wet sump appeared, according to cited evidence, to have little positive effect. Once oil enters the pump no additional air is added to the oil as it traverses the system. With changes in pressure there may be a conversion of some free air to dissolved air and vice-versa. A windage tray, or some other means of controlling the movement of the mass of oil in the pan, does have considerable positive value in terms of ensuring that the oil pump pickup does not become uncovered during high G force situation (Braking, cornering and acceleration - in order of magnitude), or preventing the crankshaft from causing additional aeration from physical contact with the oil in the sump.

So it would appear that the principle benefit of a windage tray is to control the "gross" movement of oil in the sump. However, while we are examining this situation, lets take it one step further.

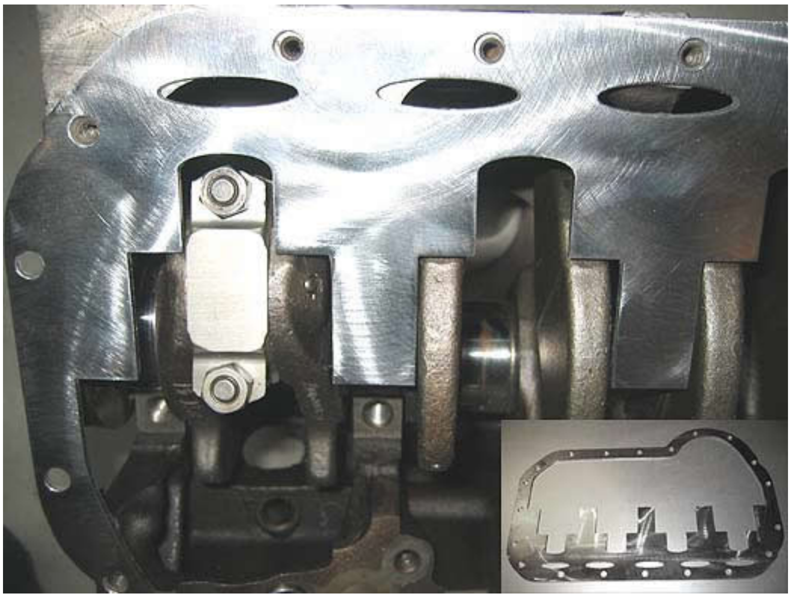
Because there is both dissolved and free air in the oil, the two must be considered together. Most test specifications dealing with oil aeration assume an 18% air-to-oil volume for dissolved and free air combined. This number also appears to be a good average to shoot for in an oil system design. As I understand it, using Bunsen's coefficient for engine oil, "for every one Bar (14.4 PSI) increase of oil pressure, the oil can take up an extra volume of air equal to 9% of the oil volume". As such at 3.5 Bar (50 PSI) the volume of air-to-oil could be as high as 31.5%. Using the old "hot-rod" formula of 10 PSI oil pressure per 1000 RPM the allowable percentage of air-to oil would equate to 49.6%.

So is this high percentage of air-to-oil what we want? **The answer is patently NO.** Quite the opposite is what is required, as there is a mechanism within the oil delivery circuit that will negatively impact the percentage of free air in oil to certain parts of the engine, particularly the upper end. Generally oil routed to the cylinder head will travel via a "sharp edged restrictor". The effect of this type of restrictor is to cause an oil pressure drop in the oil traveling to the upper end. Typically, the pressure drop will be in the order of 40%. Thus if the pressure at the main bearings were 3.5 bar, then the pressure at the rocker shaft (in the case of the Abarth motors that I work with) would be on the order of 2.1 Bar. Whereas at the oil pump, due to the increase in pressure over atmospheric, the percentage of allowable entrained "could" be more due to the higher pressure, this might place the upper end lubrication at risk. The lowering of pressure at the orifice feeding the upper end means that dissolved air may reverse from dissolved to free air. Combine the additional free air with reduced oil flow (a function of pressure and orifice size), then it is quite feasible for rocker/shaft boundary interface in an OHV engine design to be marginalized. In competition engines with more aggressive camshafts and higher rate valve springs this would give serious cause for further analysis.

One could argue that it would be beneficial to reduce the size of the restriction, thus increasing the pressure/flow to the upper end of the motor. This would have a beneficial effect on reducing the pressure drop, thus lowering the possibility of reconstituting free air from dissolved air in the oil. In addition, the additional oil to the upper end would enhance cooling due to the increased flow. However, this may decrease overall engine oil pressure and yet other measures may have to be taken to return upper end oil to the sump in an efficient manner.

So there is a fine balance to be struck if one were to contemplate running lower oil pressures to reduce parasitic losses. Certainly the use of a windage tray to control gross oil movement in the oil sump and keep the oil from coming in contact with the rotating assembly is a good idea, as the lower the entrained air percentage per volume of oil, the better. Further, the use of an effective "oil scraper" to strip oil from the rotating assembly and return it to the sump would also be recommend.

2. Installing a crankshaft scraper - A crankshaft scraper is a device attached to the main caps of the engine that is contoured to very close tolerance of the counterweights and cap areas of the connecting rods. The scraper assembly may be as close as 0.015 inch (0..4mm) of these surfaces and is intended to scrape any remaining oil from the crankshaft counterweights and rods.



The combination of an effective windage tray and crankshaft scraper will greatly reduce the impact of oil on the balance of the crankshaft assembly. Perhaps even more importantly, by reducfing the amount of oil clinging to the crankshaft and connecting rods dyno tests have shown an increase in horsepower of up to 5%.

This is of course for wet sump situations. If a dry sump is used a windage tray is not necessary, but a good crankshaft scraper my still reduce parasitic roping losses.

The instances of block failures are, in all likelihood, due to larger than allowable imbalance tolerances. A large out-of-balance condition may cause the beam stiffness, available from the standard Fiat block, to be exceeded. This will cause the engine to "pound the bearings", causing eventual main bearing and/ or block failure. Plus, the effect of maintaining strict balance tolerances will pay off in added reliability and performance due to the reduction of parasitic losses.

My thanks to Steve Smith of Vibration Free in Oxon UK for some of the material used in this discussion.

3.4 - Compression Ratio Fundamentals

There is a basic difference between “Calculated Compression” and “Dynamic Compression”. Both numbers are important, but the one that will make difference in the reliability of a race motor will be the dynamic compression.

Calculated compression is basically the ratio between the volume of the cylinder and combustions chamber, divided by the combustion chamber volume. In the case of a A112 motor with a standard cylinder bore, this could be as follows:

Cylinder volume (per cylinder) 259.75cc

Plus, Combustion chamber volume 28.00cc

Equals 287.75cc

Divided by, Combustion chamber vol. 28.00cc

Equals 10.28

So this motor would have a Calculated Compression of 10.28:1 and would run quite happily on high octane pump fuel.

For most competition engines we will want to raise the compression ratio to around 12.25:1, and in some cases we may even go as high as 13.5:1.

This calculated compression ratio is the beginning of a much more complicated calculation to determine the “knock resistance” of the engine. This will involve both the camshaft intake valve closing point, the octane rating of the fuel to be used and, the Dynamic Compression ratio.

First, we have to compute Dynamic Compression ratio. I am not going to list the actual formula for computing it here, but this information can be found in the internet for those who are really interested.

Here is a comparison of different camshafts in a 1046cc motor with 110mm rods, 12.25:1 static (computed) compression ratio.

Camshaft	Duration/Lift	L/C	Overlap In.	Int. Closing Deg	Dynamic Displ (DD)/DCR
SLR300S	290/300/12.4mm	107	81 deg.	69 deg.	804cc/9.49:1

		deg		ABDC	
SLR300	300/12.4mm	108 deg	84 deg	74 deg. ABDC	765cc/9.07:1
Kent FT 6	304/10.8mm	106 deg	92 deg	78 deg. ABDC	732cc/8.73:1
PBS A8	305/10.6mm	108 deg	89 deg	76.5 deg. ABDC	745cc/8.86:1
CatCams	305/11.45mm	108 deg	89 deg	80.5 deg. ABDC	711cc/8.50:1
CatCams	310/11.45mm	108 deg	94 deg	83 deg. ABDC	689cc/8.27:1
Abarth 316	316/10.4mm	105 deg	96 deg	88 deg. ABDC	644cc/7.80:1
Laur 319	319/10.5mm	108 deg	103 deg	87.5 deg. ABDC	649cc/7.84:1
Abarth 336	336/11.7mm	105 deg	126 deg	93 deg. ABDC	634cc/7.70:1

From the above, it is obvious that if someone asks you what compression ratio you are running, the answer will be meaningless, without telling the person asking the question, additional details.

The formula for computing horsepower is: **Horsepower = rpm x torque / 5252**

From this you can deduce that if all of these camshaft combinations were to produce the same "peak" horsepower, then those camshafts with smaller valve overlaps, and early intake valve closings, will produce higher torque/horsepower at lower RPM and taper off before 8000+ RPM, whereas the cams with large valve overlaps, and later intake valve closings, will produce lower torque numbers and will rely on higher RPM levels (perhaps as high as 9000+) to make the same "peak power".

Peak horsepower is great for bragging rights, but we don't actually spend a great percentage of our time running at 8000 RPM or more. Much more time is spent between 5500 and 7500 RPM, so this is where we should aim for the best engine efficiency. The better indication is to plot the torque/horsepower numbers 5000-7500 RPM and to find the highest "average" horsepower and torque in that range. Then choose a camshaft that will best deliver this. This will, in most cases, provide the best overall performance in a road racing vehicle.

There is a direct relationship between the "Dynamic Displacement (DD)" of an engine and the closing degree of the intake valve. This, then in turn, determines the DCR of a motor. This is one of the critical design criteria in building any race motor. The earlier that you can close the valve, without creating negative pumping effect, the greater the DD and the resultant DCR. You can also vary the DCR by advancing/retarding the camshaft up to 4 degrees. The higher the DCR, the greater the knock sensitivity, and therefore the greater the fuel octane requirement.

You will notice that this one engine can in fact produce a Dynamic Compression (last column in the chart) ranging from 7.70:1 to 9.49:1. It may not seem immediately obvious, but it is unlikely that all of these engines will survive using fuel of the same octane rating. From my own experience I know that a dynamic compression of 7.99:1 will have a "knock rating" of approx. 4.4 @ 6000 RPM. This is very close to being marginal for 100 octane fuel, but with no more than 28 deg. of total distributor advance. However, changing the camshaft to one that produces a dynamic compression of 8.99 will require a fuel with an octane rating of 105 in order to maintain a knock rating of 4.4 or less. Now if we jump up to the most aggressive camshaft, with a dynamic compression of 9.45:1, nothing less than 112 octane racing fuel will do.

As you have probably noticed by now, compression ratio cannot be viewed alone, without taking into account ignition timing and fuel octane. I suggest that all who are interested knowing more about "fuel octane" numbers go to the following link and read the information carefully. Here in 3 pages is a very good explanation of the different rating measurements and how to compare different rating systems.

http://www.btinternet.com/~madmole/Reference/RONMONPON.html

After you have read and understood this information, the following will make more sense.

Note: I am running my own engine at a computed compression of 13.5:1, The camshaft that I an using computes to a DCR 10:57:1. With DCR levels this high you MUST use very high octane. For this motor I use Sunoco 112 octane leaded racing petrol. In the interest of "longevity" I have reduced the static compression ratio to 12.5:1, which has lowered the DCR to approx. 9.67:1

Valve Train and Camshafts

4.1 Valve Train Friction

The valve train is one of those almost forgotten items in a race motor. Like most other things, there is lots of folklore about what is good and what is not. Over the years I can count the amount of time that I have spent playing around with valve train components not in hours or days, but more likely in months.

- First, it is important to understand that the valve train assembly consists of everything from the cam bearings to the valve seat, and everything in between.
- Second, the valve train also accounts for the highest percentage of total parasitic loss within an engine. Whatever can be done to reduce parasitic loss WILL make a difference, albeit small at times.

Cam Bearings - Letâ€™s start with the cam bearings. As Scuderia Topolino does on great deal of work on A112 type motors, I will focus on this type of installation, but the general ideas are applicable to almost any internal combustion engine. There are basically three cam bearings. The front one is pre-sized, but the middle and rear must be bored-to-fit in the block. This is a very important step, as the specifications for the cam bearing clearance are 0.001-0.0015 inch (0.025-0.037mm). The first task in optimizing the valve train, is to check the straightness of the camshaft. Knowing the bearing tolerance, the camshaft cannot be out-of-true by more than 0.0005 inch (0.012mm) or it will bind in the center bearing or have insufficient clearance. Next, the camshaft lobes should be in good condition and/or polished lobe surfaces . Ultimate friction reduction would involve DLC coating the bearing races and the lobes, to where the Ra value of the surface approaches 0.1.

Lifters and Lifter Bores - Next we need to look at the lifters and lifter bores. The foot of the lifter should have a slight convexity. This means a new lifter will have a slight crown to the foot. If you want to visualize this, take two new lifters and place the feet against each other. You will notice the curvature more readily when you do this. This curvature insured that the lifter turns a small amount each time the lobe lifts the lifter, to distribute the wear. If you ever find a lifter with a distinct wear pattern, then it likely that it has lost its convexity and is not rotating. In the race engines that I build in generally hone the lifter bores with a special small cylinder hone that has cork contact areas. I use a special compound, deposited on the cork and use this to provide a very light surface polish to the lifter bores. Again, if the ultimate in friction reduction is desired, then you could DLC coat the stem of the lifter. I do not recommend drilling any holes in the lifter. This significantly weakens the lifter and failure of the lifter can be the result. Be sure to inspect the push rod seat in the bottom of the lifter. This should have a shiny appearance and should not have any defects.

The Push Rod - The next item to consider is the push rod. I know that all of you will have seen advertisements for all types of alternative push rods, ranging from one aluminum ones to carbon fiber to metal-matrix-composite. Just keep the following in mind. The push rod should be as stiff as possible, in order to maintain proper cam lobe to lifter tracking, at a price you can afford. That said, yes, a push rod made from 3M aluminum matrix composite (not aluminum tubing) is lighter and stiffer than an equivalent 7mm 4130 steel push rod. However for most competitors it looks less appealing when the price of over $200 per push rod is realized. Because of the nature of a tube, as compared to a solid rod, a tubular steel push rod provides a good balance between performance, durability and cost. The last thing we want is for the push rod to flex. This produces unwanted harmonics in the valve train (valve spring to be exact) and will cause valve train failure.

If we divide the valve train into the components that are on the lifter side of the rocker arm fulcrum and those that are on the valve side of the rocker arm fulcrum, then it is more important to reduce the weight of all of the components on the valve side.

The Rocker Arm - Next is the rocker arm itself. Its weight is divided on either side of the fulcrum, and for most instances it is a bout a 50/50 proposition. Taking weight out of the tip side (where it contacts the valve stem) will help. Be careful how far you go, as you do not want to lighten it so much that the rocker becomes unreliable. In the rocker arm alone are three distinct boundary layer friction interfaces. There is the adjuster-to-push rod cup, the rocker-to-shaft, and the tip-to-valve stem.

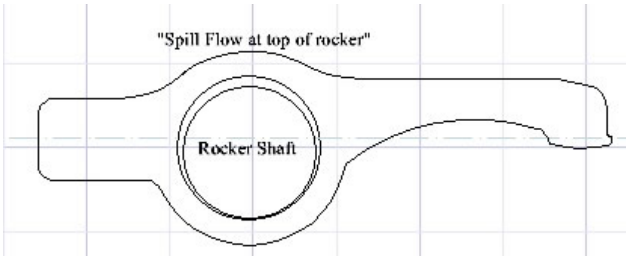
To recap, we had a cam-lifter friction interface, a lifter-bore interface, a lifter cup-push rod interface, plus the three associated with the rocker arm. Not hard to see why this is such an important area, as none of these interfaces are hydrodynamic film interfaces* and have some sliding friction action associated with them.

The push rod cup-adjuster interface is one where little improvement can be made. I have considered making a very small orifice in the rocker arm which would exit on the push rod side of the rocker arm, behind the adjuster. This would allow some oil to squirt onto the adjuster and run down into the push rod cup. This would provide a level of oil cooling to this interface, as otherwise it simply replies on whatever random oil happens to splash into the cup. If you have a situation where the top of the push rod is discolored (brown or blue), then it is getting much too hot and additional cooling is required. This is generally a result of excess valve spring pressures and or lack of lubrication.

The actual rocker arm is the next item that we should take a look at. As you know, many used Fiat rocker arms have excessive play, even when installed on a new shaft. Even worse, as the metallurgy of the rocker and the shaft is similar, they both wear equally, making the problem more pronounced. I recently measured about 40 rockers and NONE had the required 0.002-0.0025 clearance, if used with a new shaft. New rockers are available (at about $40 per rocker) but are in limited supply. Of course a new rocker arm, without any metallurgical improvements, would suffer the same fate.

The problem of rocker longevity, and damage, is one that rarely affects standard road going Fiats, as these are generally not subjected to the extra stresses imposed by high lift camshafts and the associated parts. Yes, a street engine, poorly maintained with 50,000 miles on the odometer will have worn rocker arms for sure. In racing applications, where we run with stronger valve springs, much more aggressive lobe designs and much higher valve lifts, the dynamics are totally different. ALL of these put additional stress on the rocker/shaft boundary layer interface. As I mentioned before, even standard rockers from road cars will show signs of damage if the oil to the shaft has been less than adequate for some reason.

If we look at the dynamics of the rocker-to-shaft interface, the first rule of thumb to remember is that oil will take the <u>course of least resistance</u>. Therefore, if there is an excess of clearance, the likelihood of high spill flow is very likely. If there is no pressure (only flow) then, <u>when spring pressure is applied to the rocker and all of the clearance moves to the top of the rocker, the oil flow will go to where the clearance is </u>. Thus the very area that needs the oil, THE BOTTOM OF THE ROCKER, will not get any.

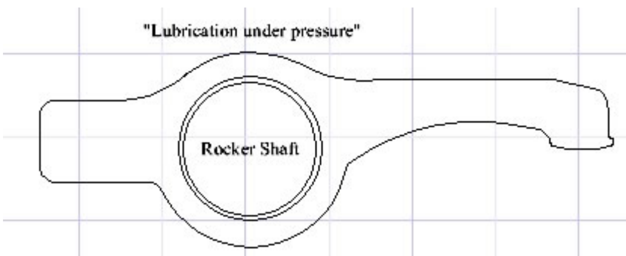


The second issue with the standard shaft, in a racing environment, is that the supply hole is too small to allow sufficient flow to service the 8 rocker oil delivery holes. As such, there is a pressure drop at this small inlet orifice and this will restrict the flow of oil. If, as I described above, there is excess clearance and the flow to the rocker is reduced, then all of the available flow will escape to the top of the rockers (where the excess clearance is) of the rockers immediately adjacent to the rocker shaft supply inlet hole. In this case, when the oil is hot, it is quite possible to have NO OIL PRESSURE build-up at the rocker arm/shaft boundary interface. This will then mean that surface friction will go up dramatically and heat will not be carried away as there is no oil flow at the rocker/shaft boundary interface. The end result is a seized rocker arm. (Note: Please read the section in blue later on for more clarification)

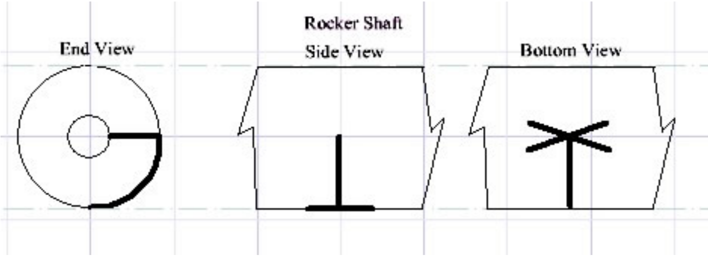
The solution is therefore to be found in four modifications.

1. Increase the supply hole to the shaft (remember there are two, even though one is used at a time) to 0.150mm (3mm) diameter. This is equal to the total area of the rocker supply holes** (0.027 inch [0.7mm] each) in the shaft. In this manner the input and output flow capabilities of the shaft will be balanced and insure that adequate flow and pressure are available to the rocker/shaft boundary interface.

2. Insure that the rocker-to-shaft clearance is approx. 0.002 inch(0.05mm), but no larger than 0.003 inch (0.075mm). This will insure that the supply flow to the rocker is greater than the "spill flow", and allow pressure to build up at the rocker/shaft boundary interface. This will promote better lubrication and cooling of this interface. As a secondary issue, the spill flow must be under sufficient pressure to cause oil to "splash" into the push rod cup. This is the only method for both lubrication and cooling this vital pressure interface. The surface area of the adjuster, where it rides in the push rod cup, takes the full force of the lobe opening pressure and must be adequately lubricated and cooled.



3. Modify the shaft, by providing a partial groove from the rocker oil delivery hole to the bottom of the rocker shaft, and then scoring an "X" at this point to spread out the oil. If the correct rocker/shaft clearance is maintained, then this will provide a wide cushion of oil for the rocker to work against.



4. Finally, where rules allow, the engine preparer may opt to install a small oil spray bar inside the valve cover, fed from a small line external to the motor. This will help insure that both the push rods and the valve springs are adequately oil cooled by high pressure oil. This is a common modification in many OHV motors, particularly in NASCAR, where flat tappet designs with very high valve lift are commonplace.

Further Information and Calculations

When a rocker is under tension, it is up against the bottom of the shaft and, there will be no oil flowing into the rocker/shaft boundary interface pressure point (presuming that the feed hole is on the bottom of the shaft). Considering that each rocker is under tension approx. 250 degrees out of every 720 degrees of crankshaft rotation, it can be assumed that, for the remaining 470 degrees of engine rotation, oil will flow through the rocker/shaft boundary interface clearance (approx 0.002 inch). This flow would re-establish a lubrication supply to the rocker/shaft boundary interface in preparation for the next 720 degree engine rotation cycle for that particular rocker arm, and, most importantly, carry away heat generated during the previous cycle at this boundary interface.

According to my observations there are 4 rocker arms in tension (to some extent anyway), at the rocker/shaft boundary interface, at any given point in time. During this "tension period" all the clearance is at the top of the shaft, with virtually nil clearance at the boundary layer pressure point (also the oil feed hole) on the bottom of the shaft. As 4 of the eight rocker feed holes are occluded, either partially or completely, any oil flow would be diverted to the remaining four rockers not under tension, with equidistant circumferential clearance.

Based on my earlier computations of cross-sectional flow area, each rocker's 0.070 inch (1.75mm) feed hole will flow about 10% greater volume than the spill volume of the 0.002 clearance between the rocker and shaft. The spill volume computes to an area of .004 sq. inch (2.58 Sq mm) per rocker arm. As such, the aggregate spill volume for 4 rocker arms would be 0.016 sq. inch. The single oil supply hole to the shaft must be 0.145-0.150 inch in diameter ,to supply sufficient flow, to service any **four** non-occluded rocker arms at any given point in time.

The metallurgical inconsistencies associated with a steel rocker arm against a steel shaft are not acceptable to for a high performance application. A better alternative would be a harder rocker shaft, made of 4130 steel and hard chrome finished for a surface hardness of RC65, and then an associated rocker arm with a pressed in bronze bushing. This combination of differing metals will prevent rocker arm seizures due to microwelding.

Scuderia Topolino has already undertaken steps to produce this new type of rocker shaft and rocker design.

With a conventional Fiat rocker arm the pad on the end of the rocker arm is larger than the stem of the valve that it contacts. Many times I have see this pad marked with a small â€œhalf moonâ€? as indication that there is a great deal of pressure concentrated in a small area in this interface. I will talk some more about the causes for this pad damage in the camshaft dynamics later on. It has always been my contention that it would be better if we could make maximum use of the total surface area of the contact pad, and hence many of the motors that leave Scuderia Topolino have â€œlash pads or lash capsâ€? over the end of the valve stems. This serves two purposed, both to protect the end of the valve stem and the pad on the rocker arm. Yes, it does add a minute amount of weight to the valve train on the â€œsensitiveâ€? side of the rocker arm fulcrum, but the benefits far outweigh the weight penalty. The pad and the associated lash cap should be polished to a fine finish to reduce parasitic losses.

One other possibility is to use an alternative rocker arm. Scuderia Topolino has available an aluminum rocker arm with a rollerized tip. The aluminum rocker arm body is made from 2024-T6 aluminum and is then hard anodized. This provides a surface hardness of RC62, and combined with the ductility of 2024 aluminum, provides a usefull wear surface against a steel shaft.

* *There is some evidence that at higher RPM the lifter-cam lobe interface converts from a boundary layer interface to a hydrodynamic interface.*

Valve Spring Retainer â€" Here is the first area where we can make a real weight savings. By changing to a titanium retainer we can cut the weight in half. It may not seem like a great deal, but it will make a decided difference in the overall dynamics. Scuderia Topolino uses special retainers that use special collets, with an included angle of 6 degrees, as opposed to the standed Fiat ones which are 5 degrees. We can also provide these retainers with 7 degrees collets, and then the collets can be provided in titanium as well.

Valve Springs - We finally pay attention to these when we install a new camshaft, to check that we do not have coil bind. There are however a number of other considerations that must be examined in terms of valve springs.

Most standard valve springs are selected on the basis of basic performance and longevity. Obviously when the 4 cylinder Fiat engine was first developed, what with a whopping 27 horsepower, the requirements of valve spring performance was not very comprehensive. After all, with a camshaft with a total lift, at the valve, of less than .300 thousands of an inch (7.6mm), the real criteria was to use a spring that would last a long time. These springs had a sufficient number of coils so as to not be very highly stressed.

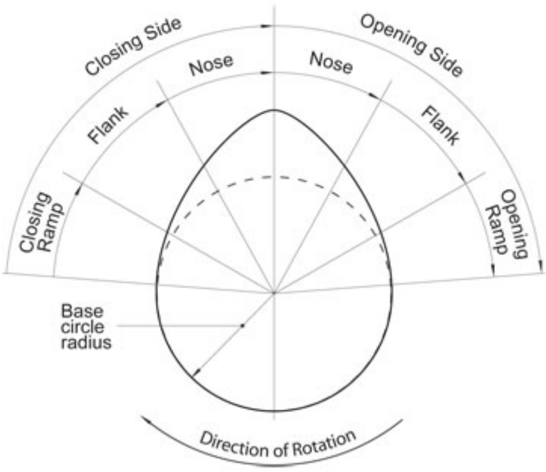
In a racing situation the requirements are just the opposite of that of the standard road car. First, whereas the standard engine uses relatively low RPMs, therefore spring harmonics play a minor role. Not so in a competition engine. Modern racing camshafts not only open the valve more and for a longer period, they also open the valve quicker as well. This means that the modern valve spring must be able to control rapid valve train acceleration. We need to find a good balance between spring pressure, harmonic control and minimum parasitic loss. Without a great deal of trial and error, the only other alternatives is to computer model the valve train, or to follow the recommendation of the camshaft supplier. The cam grinder will almost ALWAYS be very conservative in choosing a valve spring. In most cases he would rather err on the side of a too heavy spring, than one that is too light.

Valves â€“ Here again, we are confronted with a choice between ultimate weight savings and cost. One way to reduce the weight of the valves is to use a small diameter stem, say from 7mm to 6mm. Also, many racing oriented valves, because the way they are formed, are inherently lighter than the standard Fiat valves. Finally, if the pocketbook will allow, you could go for titanium valves. This will greatly reduce the weight. I am fairly comfortable with using titanium intake valves, but titanium exhaust valves have a very short lifespan, and should be replaced at least once each season. This of course adds to the cost.

Valve Seats â€“ While technically there is nothing that can be done to lighten valve train components with the valve seat, if you plan on using titanium valves, then there will be a cost impact that is not insubstantial. Unfortunately you cannot use steel valve seats, as the titanium valves are subject to a phenomenon call â€œmicroweldingâ€?. Tiny amounts of steel are transferred to the valve during operation and eventually the valve no longer seals. This means that either special copper or berrylium valve seats must be used. Unfortunately these are from 6-15 times more expensive than a steel seat.

4.2 Camshaft Dynamics

Camshaft and Induction Dynamics



In an effort to better understand camshaft technology, I asked a number of fellow Abarth competitors what type of camshaft they were using. I wanted to take these various camshaft designs and put them through a computer based engine simulator to compare the various grinds. Below you will find a spreadsheet of the information that was provided.

Manuf	Adv	Lobe	Cam	Intake	Overlap	Open/Close

	Duration	Center	lift (mm)	L/C		Adv.
CatCams	305	108	7.21	108	78	39/79 79/39
PBS A8	305	108	6.98	108		
PBS A6	292	106	6.85	104	72	40/72 80/32
SLR286	286	106	8.43	106	74	37/69 69/37
SLR300-106	300	106	8.43	104	88	46/74 78/42
SLR300-110	300	110	8.43	105	80	45/75 85/35
Kent FT6	304	106	7.11	106	92	46/78 78/46
Alquati	316	110	6.85	108	96	48/88 88/48
Abarth	316/304	105	7.21	102	100	53/83 77/47
Abarth	336	105	7.72	105	120	60/96 96/60

As you can see, almost every competitor is using a different camshaft. I did find that two people were using the same Kent FT6 camshaft. Duration ranged every where from 286 to 336 degrees. Certainly there were some interesting surprises when I ran some of these profiles through my analysis program. My first reaction was that the larger the duration, the more horsepower. Not quite correct. At the end of this study I will rate each of the above cams that I was given "advertised" duration information for.

First I have to state the assumptions that I used to make all of the comparisons that follow.

Bore	68m
Stroke	74mm
Rod length	110mm
Compression	13.8:1
Cylinder head	PBS 8P
Inlet Valve	31mm
Exhaust Valve	27mm
Carburetion	2 x DCOE40

There is a complex "ballet" of inter-acting numbers that define a particular camshaft and what it is capable of delivering. As with anything in life, a good "plan" is always worth the time it took to develop. Such a plan is also required when deciding upon a camshaft and the other components it will be required to interact with. In my mind there are three principal elements. It is very much like a 3-legged stool. It cannot stand unless all three parameters are well thought out and developed.

A) The maximum RPM that will be used. - It is of little value to specify a camshaft where horsepower is likely to be produced above the RPM range where it will be required.

B) The RPM where maximum torque will be expected. - This may have more to do with shift points and such, but it is something that can be adjusted for to some extent.

C) Best Available Octane Fuel â€“ This will affect which camshaft can actually be used without encountering detonation.

From the chart above, it is obvious that different competitors had different things in mind when choosing a camshaft. I am of the firm belief that, given the design of the small Fiat motors with only three main bearings, that a top RPM around 8500 is not only prudent, but the only way of assuring reasonable reliability. Therefore, I will go out on a limb and say that anything over 304 degrees duration is may too great, as it puts the power production too high in the RPM range. Now there will be people who disagree, but don't dismiss the idea just yet.

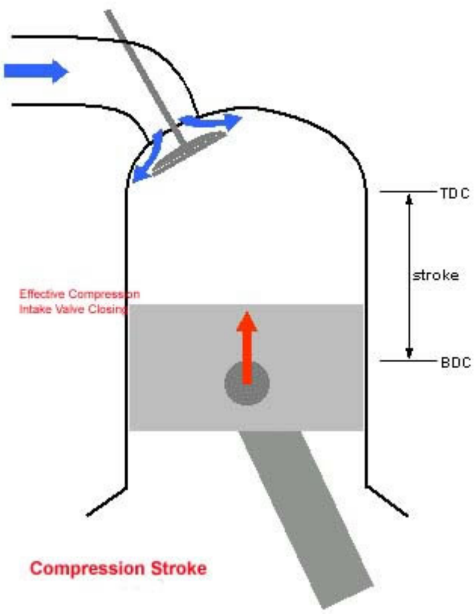
The PBS head and intake manifold/Weber carburetor combination has an overall inlet tract length of around 330mm. Generally the carburetors would have 30mm chokes. We want to maintain a mean a peak torque number somewhere between 5500-5800 RPM. Below is a chart of the calculations that I did for a 1050cc motor (68mm bore 74mm stroke, volumetric efficiency of 85%) for the diameter of the inlet port at the head/intake manifold interface.

Peak torque RPM	Inlet Diam.(mm)
5000	21.8
5250	22.8
5500	24.1
5750	25.1
6000	26.2
6500	28.4
7000	30.7

All PBS heads are machined for a 25.4mm port, so it would appear that there is a good match. This size port assumes an air velocity of just under .6 mach, or 650 ft/sec (196m/sec). So to match these head characteristics we will be looking for a camshaft that will deliver peak torque between 5000 and 6000 RPM.

Finally we need to look at third leg of our three legged stool, namely Fuel Octane Rating, and this will then tie directly to Dynamic Compression Ratio (DCR), which is quite different from the Computed Compression Ratio (CCR). We should all be familiar with how CCR is calculated. Basically it is the displacement of the cylinder, plus the displacement of the combustion chamber, then divided by the volume of the combustion chamber.

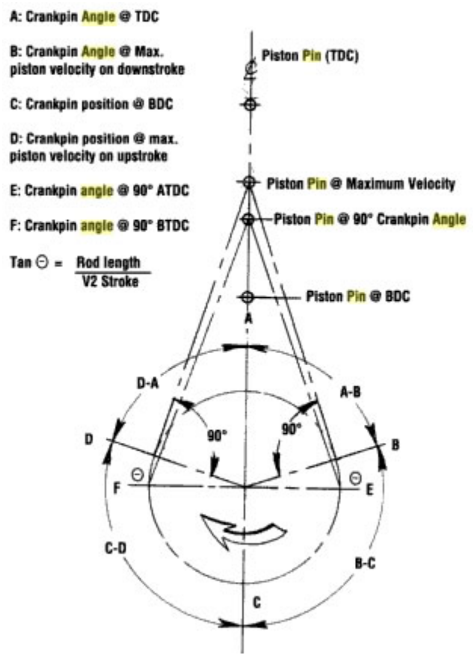
For our "standard" engine I stated that the compression ratio would be 13.8:1. Of course the intake valve is NOT closed for the entire compression stroke (BDC to TDC). In fact, even though the piston has already passed BDC, air is still flowing into the cylinder due to scavenging effect. The key is adjusting the intake valve closing position so that it coincides with the point where intake flow ceases.



According to the camshaft specifications in the chart the intake closing is anything from 65 to 96 degrees. With such a wide variance, it will be interesting to compare.

A small story *- At a recent race at Hockenheim I ran across Bram Paardenkoper. During our conversation he mentioned that he had installed an Alquati 316 degree cam in his car for the weekend, as he was trying to find some extra pace. He indicated that over the years he had done quite well, but now the competition was starting to reel him in. He was however disappointed as the car, with a cam with 12 degrees more duration, was actually slower than before!! The Alquati cam is very similar to the 316 can in the list, and the cam Bram used previously is very close to the Kent FT6 grind. This led me to do some further analysis.*

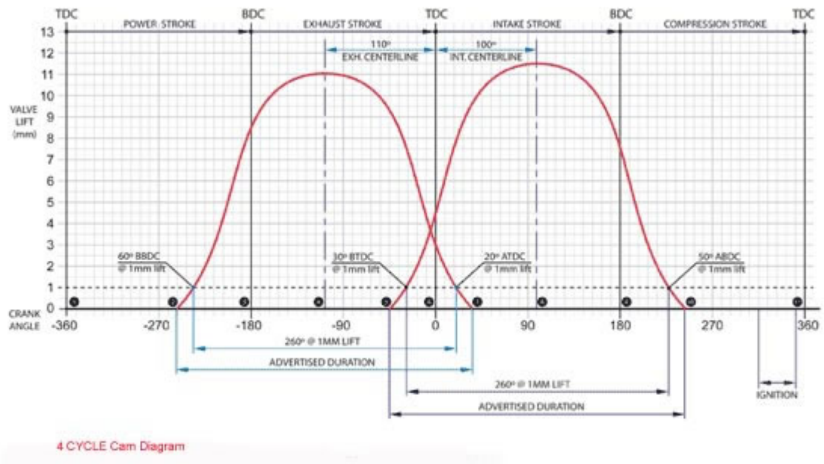
I started out by looking at where the rod and crank throw were at 90 degrees to each other on the compression stroke, in terms of degrees of crankshaft rotation, and how many degrees this was from the point of maximum piston acceleration. Perhaps this diagram will help in understanding the the reason why the rate of piston acceleration is not uniform over the entire stroke.



It turns out that for the A112 engine this is at 71.4 degrees before TDC. Conversely, this means that on the compression stroke the point of maximum acceleration for the piston is 108.6 degrees from BDC.

The most important consideration for any engine is the timing of the closing of the intake valve. The valve must be closed before this point of maximum piston acceleration to minimize the effects of pressure reversal in the cylinder. Of course the earlier that we can close the valve, and still meet our design objective with regard to power generation, the better the performance will be. After modeling many camshafts for use in the A112 motor, with its particular bore and stroke characteristics, it would appear that a valve closing period between 65 and 78 degrees ABDC produces the best results, at least for engines like the A112. Coincidentally this also places the closing event prior to the crankpin achieving a 90 degree angle with the connecting rod centerline, so that the intake valve is closed prior to the fastest portion of the piston acceleration.

The challenge is then to find a combination of lobe center, overlap and duration that will maximize DCR. Remember that DCR can only be the amount of the stroke from the time the intake valve is closed to TDC.



Suppose we have a cylinder volume of 268.75cc per cylinder (68mm stroke x 74mm bore), with a combustion chamber volume of 21cc. This would provide a CCR of 13.8:1. Now if we use the Alquati 316 degree cam (110 deg L/C, 6.86mm lift at the cam), with an intake closing of 83 ABDC, we can compute the position of the piston above BDC and determine the actual cylinder volume remaining at that point. In this case, at 83 degrees the cylinder volume (VE) is 170cc. So according to the formula "(Cylinder Volume + Combustion Chamber Volume)/Combustion Chamber Volume", this makes for an DCR of 9.2:1. The DCR is always lower than the CCR, it is only important how much lower.

Now if we take the Kent FT6 304 degree camshaft (106 deg. L/C, 7.11mm lift at the cam) with a intake closing of 74 degrees (installed 4 degrees advanced) this results in a VE of 192cc and a DCR of 10.2:1. This is a full compression point higher. In addition the overlap on the Kent camshaft is also 4 degrees less.

Small story continued *- Seeing the above results I could easily see that the Kent camshaft was a much better cam for the A112 motor.*

Most camshafts are installed 2-4 degrees advanced, particularly if they use a cam chain which over time will stretch. This will change the Effective Cylinder Volume and also the Effective Compression Ratio. Below find a spreadsheet of the results of the computations for the various camshafts. Remember that Actual Cylinder Volume is 268.75cc per cylinder.

Manuf	Adv	Lobe	Cam	Intake	Overlap	Open/Close	VE in	CRE

	Duration	Center	lift (mm)	L/C		Adv.	cc	
CatCams	305	108	7.21	108	78	39/79 79/39	182.4	9.68
PBS A8	305	108	6.98	108				
PBS A6	292	106	6.85	104	72	40/72 80/32	196.88	10.37
SLR286	286	106	8.43	106	74	37/69 69/37	202.73	10.65
SLR300-106	300	106	8.43	104	88	46/74 78/42	192.86	10.18
SLR300-110	300	110	8.43	105	80	45//75 85/35	190.82	10.09
Kent FT6	304	106	7.11	102	92	50/74 82/42	192.86	10.18
Alquati	316	110	6.85	108	96	48/88 88/48	162.31	8.73
Abarth	316/304	105	7.21	102	100	53/83 77/47	173.67	9,27
Abarth	336	105	7.72	105	120	60/96 96/60	143.34	7.83

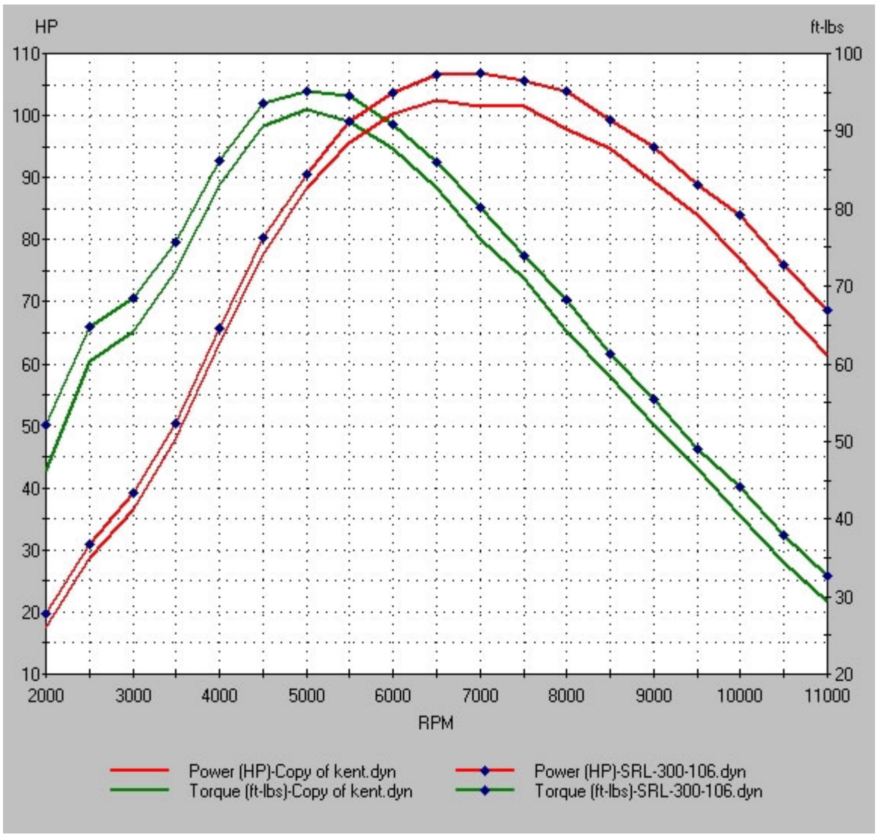
By looking at the above chart it becomes obvious that BIG duration, with small lobe centers, equates to a severe lowering of VE and the directly interrelated DCR. So a compromise must be reached. We can choose a larger lobe center, making sure the power band stays where we want, and also advance the cam to get back the VE and DCR numbers that we want.

This is what has been done with the SLR cams, Kent FT6 and the PBS cam and it appears that the "sweet spot" is between 71.4 and 78 degrees for intake valve closing. This produces a DCR above 10 in all cases. Noting is free however, and DCR numbers over 10:1 may give cause for alarm, as you would definitively have potential for detonation. It may be that slightly lower CCR may have to used, so as to lower the knock index number.

Note: I went back to an article that I had read some years ago about the short lived Pontiac GTO Trams-Am project. Pontiac's project engineer Tom Neil explained how they had gone about determining what their "road-race" engine required in the way of a camshaft. At the end of the day they also determined that the "secret" lay in the closing times of the first the intake valve and secondarily the exhaust valve. As it turned out they settled for a 300/310 camgrind on 105 centers with ,500 inch (12.5mm) lift. Intake closing was slightly different, because of the short stroke and long connecting rod. However when computed backward, it falls right into the range that I found to be effective for the A112 motor.

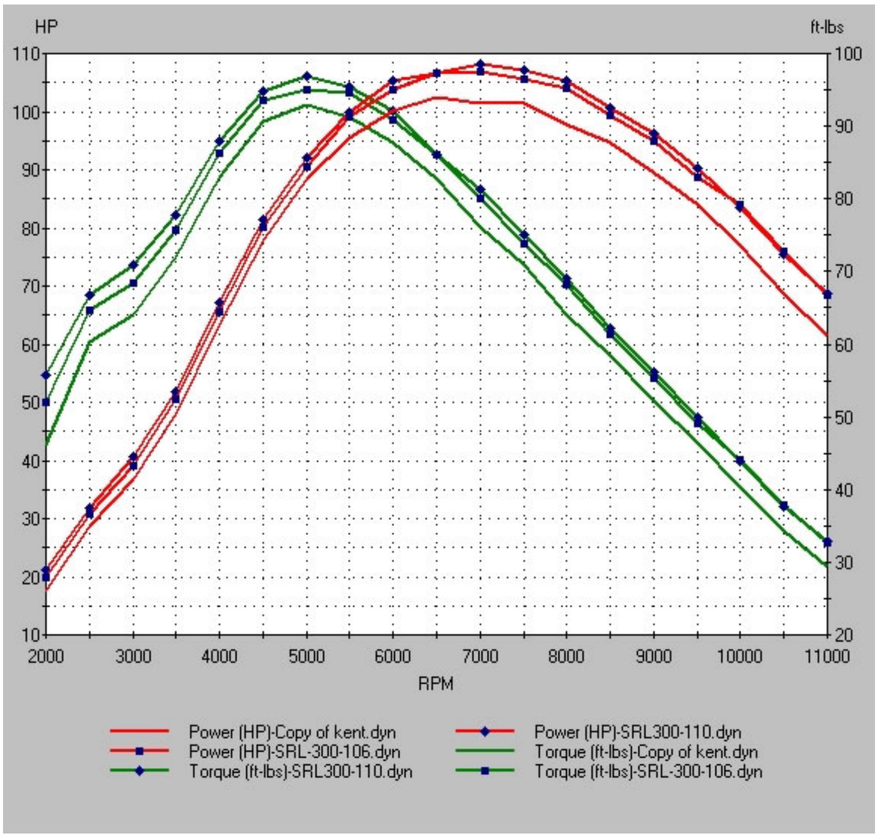
There is one additional difference and that is that the SLR cams all have smaller exhaust/intake overlap and appreciably greater valve lift than the other cams in the list. This means the lobes on these cams will be more aggressive as far as lift per degree of rotation and so exhibit greater lift "earlier", providing a greater area "under the curve". I used one of my engine design packages to illustrate this in graphical form, as in number form it becomes too cumbersome.

First, let's compare the two most directly comparable camshafts on the list for which I have data. This would be the Kent FT6 and the SLR300-106. These are 304 and 300 degrees respectively (both on 106 L/C), with the Kent FT6 winning out in duration and the SLR on valve lift and less overlap. Lets see how they compare.



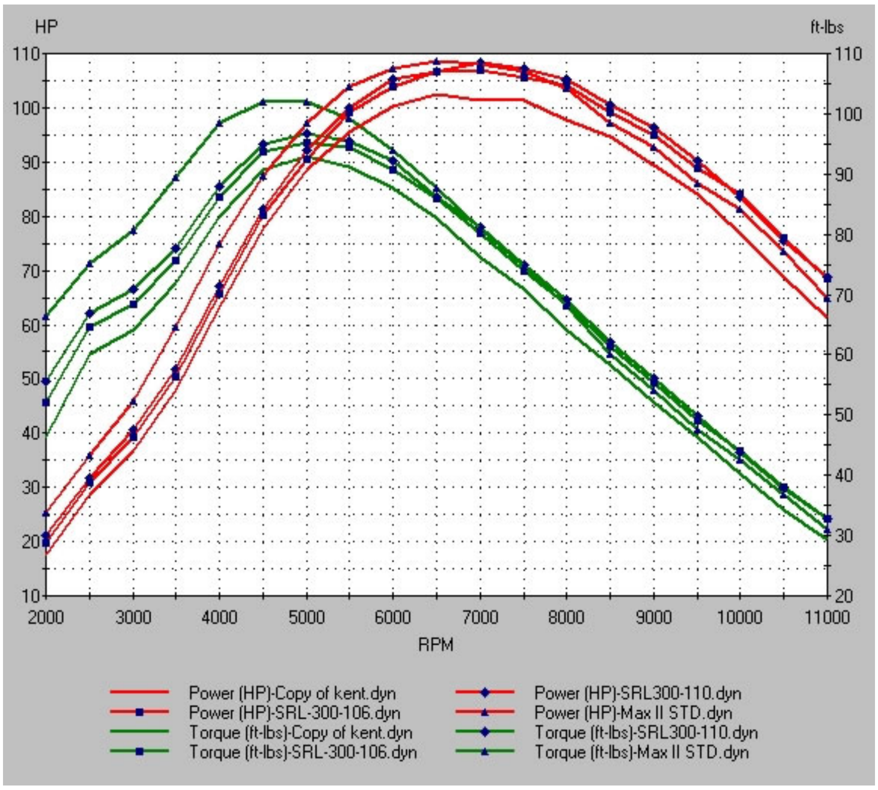
Interesting !! Here you can see that cams with the same VE and DCR, can have different power curves. Obviously the higher, and inherently earlier, cam lift has increased the area under the curve for the SLR300-106 cam. The SLR300-106 cam also has 4 degrees less overlap. Both the horsepower and torque show an reasonable increase, about 5 HP and 2 lb/ft of torque.

What would happen if we spread the lobes apart to 110 degrees, thereby reducing the overlap by another 8 degrees? Would this reduction in overlap, theoretically reducing pumping losses, have beneficial effects?



From the results you would have to say that the SLR300-110 cam does appear to do better. While the horsepower increase is relatively small (maybe 1-2), it has moved the horsepower peak higher in RPM, and there is also a noticable increase in peak torque, even though the VE and DCR numbers are marginally less than the SLR300-106.

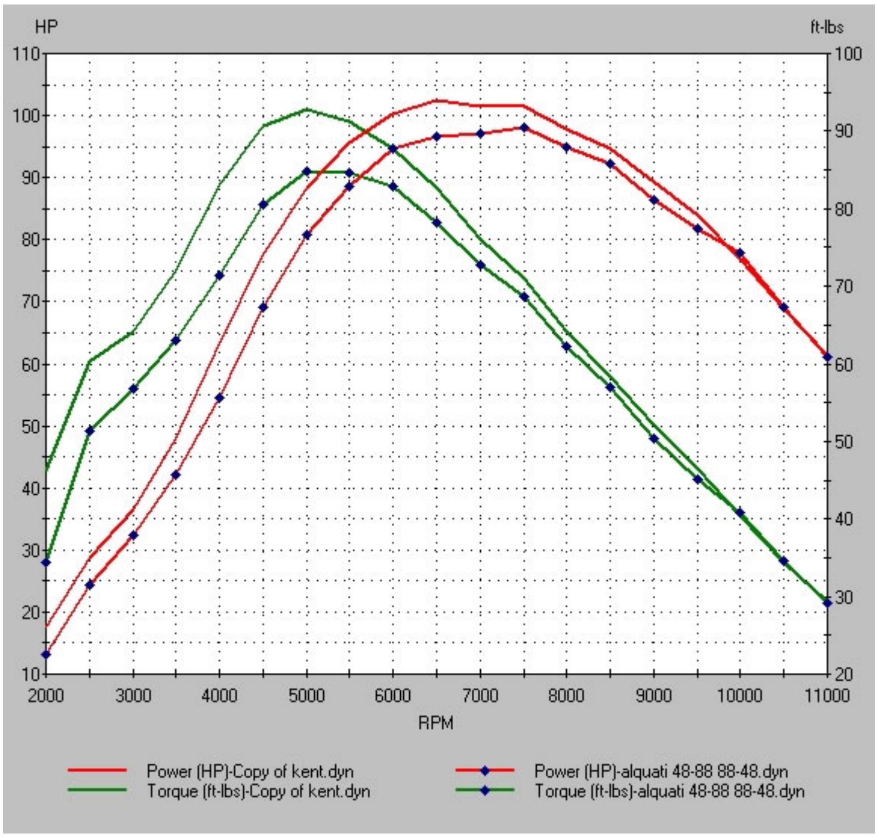
What about the SLR 286 cam (also known as the Max II). It had higher VE and DCR numbers than the other two SLR300 cams and the Kent FT6, yet it is another 4 degrees less in duration. How does it compare?



Did you expect this result? Let's see if we can judge why. The torque is considerably higher, based on having the lowest overlap of any of the cams we tested. Likewise, because of its shorter duration, it does achieve peak horsepower as early as 6500 RPM, and by 8500 has dropped off considerably.

As far as the original design criteria of a cam that had peak torque between 5000 and 6000 and carried power to between 8000-8500, this may not be the best cam, although it is what I used in 2001 when I ran in the Coppa MIlle. On the other hand if you were doing fast slaloms or perhaps even hill climbs, where good torque response at lower RPMs are required, then the SLR286 would get my vote.

Small Story Final Episode - *Just to finish the story, let compare the Kent FT6 cam that Bram would regularly run with the 316 degree Alquati camshaft that he decided to try at Hockenheim.*



Well as you can see the Alquati cam comes up well short of the original Kent FT6 and all of the SLR designs would probably outperform the Alquati as well.

So to summarize, it is probably more important to maximize the "area under the cam curve", by increasing both the aggressiveness and lift of the cam lobe and making sure that the valve closing is somewhere in the range of 65-78 degrees ABDC. Likewise, using slightly larger lobe seperation (108-110 degrees) will also reduce overlap and minimize possible pumping losses.

4.3 Optimizing Valve Train Reliability

1. **Valve Spring Forces** - The answer is both simple and complex. The simple one is "enough to keep the valve from floating or bouncing off the seat". The more complex answer takes into account the weight of various components and the aggressiveness of the opening and closing ramps.

   In order to answer this fully you would have to run the engine on a Spintron machine. Then with a high speed camera and a strobe you could isolate each of the movements of the camshaft action and the impact on the valve and spring. Since almost none of us have access to this type of equipment, we have to make some educated assumptions. These are that we need sufficient spring pressure to make sure that the lifter accurately follows the cam lobe and that the valve is not lofted off the lobe. *(There are some cam designs where the valve is PURPOSELY lofted off the valve lobe in order to achieve higher opening lifts, however this may have other consequential effects which under normal circumstances would be catastrophic.)*

   We already know that with medium lift camshafts (6-7mm at the lobe) that standard valve springs are quite adequate. Abarth made some springs that dealt with cam lobe lifts of 7-7.4mm. Finally, with aggressive camshafts with 7.5-8.2mm lift we need springs with both more preload, and with more free travel. The more aggressive cams will have valve openings between 11.5 and 12.3mm.

   I have seen some valve installations where the seat pressure was less than 30 lbs and the nose pressure only 120 lbs. The Abarth springs are rated at about 45 lbs seat pressure with about 160 lbs across the nose. For very aggressive camshafts seat pressures go up to about 60-70 lbs and nose pressures may be in excess of 230 lbs. This clearly illustrates that the rocker/rocker arm lubrication boundary interface is being stressed much harder with aggressive camshafts and springs.

   We can do things to minimize the spring pressure however. First and foremost would be to reduce the weight of those items on the valve side of the rocker arm. This includes the valve, retainer, spring and the rocker arm itself. In a secondary fashion the valve spring is also responsible for controlling the lifter contact with the camshaft. It is in this area where there may be significant gains that can be made.

   If we were to make a spring holder that would sit on top of the lifter, with an orifice for the push rod to go through, we could put a compression spring there to control the action of the lifter, and a portion of the push rod. The other end of the lifter spring would be held captive by a notched plate on the underside of the cylinder head, and positioned in the lifter galley. *In "hot-rod" circles this would be referred to as a "rev-kit".* This would relieve valve spring of this responsibility and the spring tension of the valve spring could be reduced. This would also mean reduced pressure on the adjuster/pushrod and rocker/rocker shaft interfaces, thus reducing the parasitic losses associated with these interfaces.

   The resultant redistribution of spring pressures would either allow more RPMs before valve float would occur, or would allow lighter valve springs to be used with the current limit on RPMs being observed.

   My current test have indicated that if a spring is mounted on the lifter, then the valve spring pressure could be reduced by as much as 25% or more.

   As with most things in life, there are really no "free lunches", and the same goes for this idea. It is true that the redistribution of spring pressures would have a beneficial effect on the rocker arm/shaft boundary interface. However there are some trade-offs.

   1. 1. The spring in the lifter idea will add three extra components. All of these items will add to the weight of the total valve train.
   2. 2. The addition of a third spring will also add a third harmonic element to the valve train. So we have a dampened pair of springs on the valve and an undampened spring on the lifter.
   3. 3. The chilled iron lifter will now be constantly spring loaded on the cam lobe. Principally this will have an effect on the heel, or base circle, element of the lobe. This means that additional lubrication may be required to deal with this. As the standard lifter tends to collect oil, it would be possible EDM a small hole in the bottom of the lifter to provide a "drip" oiling system for the cam lobe. It is not known if the metallurgical structure of the chilled iron lifter would take well to any "interruption" in the lifter foot surface, no matter how small. It could be that this "defect" could cause be the beginning of catastrophic lifter failure, and also perhaps camshaft failure.

   Having looked at all of the "pros and cons", I have come to the conclusion that this is not an avenue that is worthwhile pursuing, at least at this time, simply because the attendant risks outweigh the possible rewards.

   However, what it did reinforce was that a better understanding of the spring forces required for the proper operation of a camshaft is required. To that end I went back to one of my engineering programs that allows me to model for valve train dynamics. This involves recording the lift of the camshaft at 1 degree intervals (Ideally you would want to record the data at much smaller increments with a Cam Doctor or similar device, but for this exercise this level of accuracy will be sufficient) and recording this information in a "camshaft file". This information, along with the weight of each individual component in the valve train (valve, spring, retainer, collets, lash cap, rocker arm, push rod and lifter) can then be modeled to determine what the minimum seat pressure and full lift pressure that are required to have accurate tracking of the lifter to the cam lobe.

   At this point I have to again indicate that all of this came about because of a lubrication problem at the rocker arm/shaft boundary interface, but it also important in terms of the overall performance of the engine.

   For examination purposes I used our SLR300 camshaft as our test sample, principally because this is the camshaft where the rocker arm/shaft lubrication problem first emerged and it would appear to be linked to the increased spring pressures exerted by the uprated springs, supplied by Scuderia Topolino, for use with the SLR 300 camshaft. Remember, the primary valve spring criteria is that the cam must not float the valves within the operating range of the engine.

   So just how much spring pressure is required to control the valve action with a SRL300 camshaft, yet NOT overstress the rocker arm/shaft boundary interface? I decided to draw on some empirical data, provided by customers, and to model all of this to try an determine an answer. To this end II had to set some standard weights for the various valve system components. These are within one gram +/-.

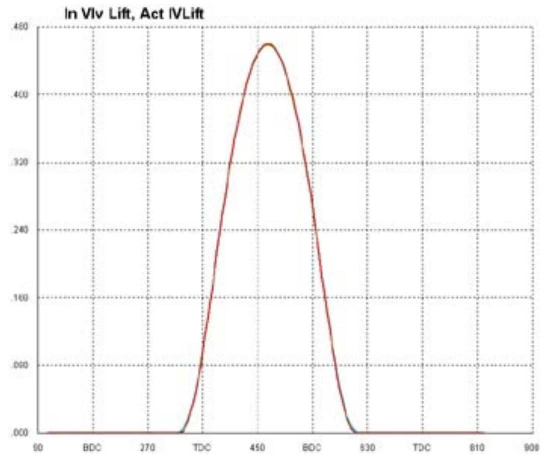
Push Rod 48gm, Lifter 29gm, Intake valve 44 gm, Exhaust valve 42gm, Titanium retainer and collets 8 gm, Spring 50gm, Lash cap 3gm, Rocker arm 52 gm.

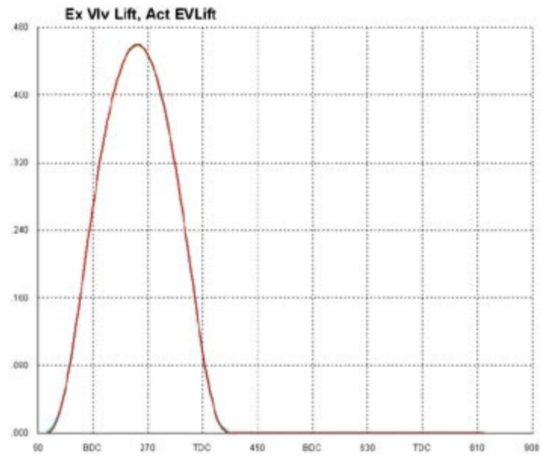
One of my customers (Customer A) used the SRL300 camshaft with a set of the valve springs we supplied, rated at 90 lb (41 Kg) seat pressure and 245 lb (111 Kg) full lift pressure. This engine suffered from a rocker arm/shaft lubrication problem, with the far rear rocker seizing to the shaft. Without knowing the condition of the rocker and shaft in terms of wear and clearance when the engine was assembled, it is difficult to estimate just how much the limits had been exceeded. This is of course always a problem with high performance camshafts, as the cam designer has no knowledge of the condition of the remainder of the components that must work together to make an effective camshaft installation. Obviously, this amount of pressure, given the Fiat lubrication system, was too much and the rocker arm failed.

Another customer (Customer B) used the SRL300 camshaft with a set of Schrick springs rated at 50 lb (22.7 Kg) seat pressure and 151 lb (68.6 Kg) full lift pressure. This engine did not suffer from a rocker arm/shaft lubrication problem. From this empirical result it would be safe to assume that if the rocker arm/shaft boundary interface were to factory clearance specifications, that a rocker arm failure would not occur. *

Obviously, the answer must lie somewhere in between.

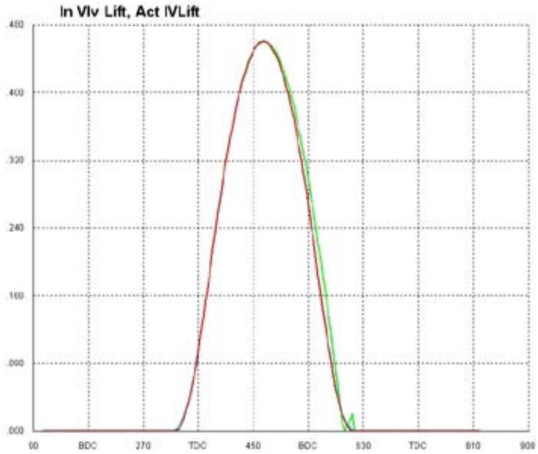
I modeled both the intake and exhaust valve valve lift and actual dynamic valve lift characteristics for Customer A first. This was a SLR300 camshaft with the Scuderia Topolino supplied valve springs. Test RPM was 7500 RPM.

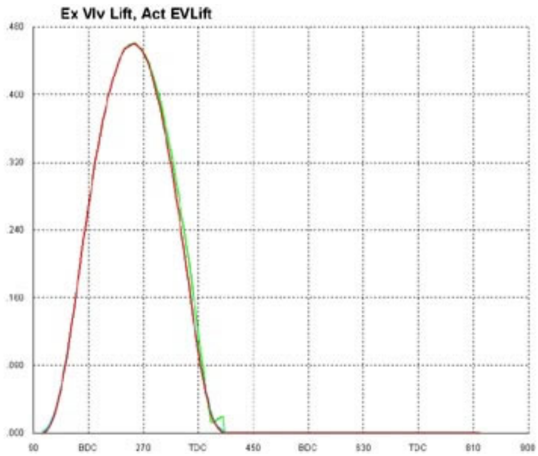




As you can see in both graphs the two traces for (red and green) are perfectly superimposed. This would seem to indicate that the valve spring supplied by Scuderia Topolinbo has sufficient pressure to control valve. **What it does not say is whether the spring pressure could be less, and still do an adequate job of controlling the valve train motion.**

Next I modeled the same SLR300 camshaft, as used by Customer B, using the spring specifications for the Schrick valve spring that he used, again at the same test RPM of 7500 RPM.

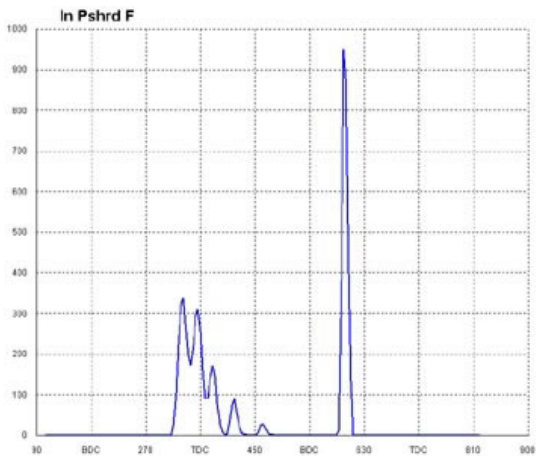


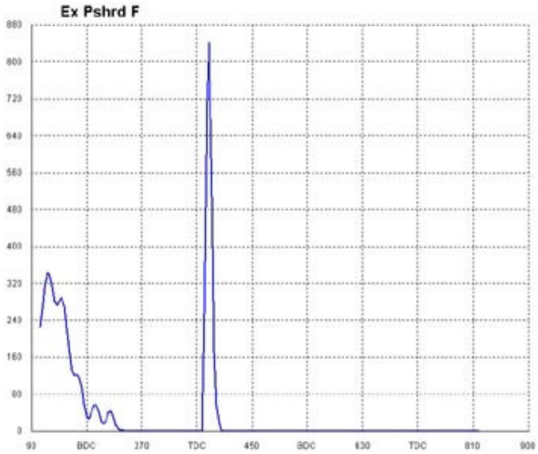


Here the result is quite the opposite. The two traces are easily distinguished. Both the intake and exhaust valve are being "lofted", indicating that the lifter is not properly tracking the cam lobe. This is not necessarily "valve float" in the traditional sense when the spring goes into a harmonic condition, but it may in turn cause some unwanted spring oscillation to occur. Obviously this is spring is not sufficient to control the valve train motion of the SLR300 camshaft. The lifter is for all intents off the closing side of the lobe altogether and comes back down with the lobe at less than 0.050 lift. You can also see that it is predicted that the valve will bounce when it does come down with considerable destructive force.

** The engine was assembled by a knowledgeable mechanic. The cam was degreed in and found to be installed to specification, with minimum valve-to-piston clearance of 0.100 inch (2.5mm). He did set the valve clearance at 0.008/0.010 inch (0.2-0.25 mm) instead of the recommended 0.020/0.022 inch (0.5-0.55mm), as he thought the clearance was too great and "could not be right". The closer clearance of course has two effects. 1) It effectivecly increases the duration of the camshaft about 20 degrees. 2) It compromises the effectiveness of the "lobe clearance take up ramps". 3) This combined with the lower pressure Schrick valve springs led to the lifter crashing on the end of the closing ramp and bouncing off the seat.*

This engine did lose compression in 3 of 4 cylinders after about 15 minutes of running. What can we deduce from the above information? First it is obvious that both the intake and exhaust valve are not in contact with the cam lobe on the closing ramp, and totally overshooting the clearance take-up ramp on the camshaft. This would cause a very high spike in pushrod pressures. (Note: On engine disassembly the camshaft had at least one damaged cam lobe) See the following graphs.





These graphs illustrate the effect of valve "lofting". The closing ramp forces are more than three times higher than the opening ramp forces. This camshaft was going to fail, it was only a matter of time.

Valve lofting has other unintended consequences. First, the inertia contained in valve lofting could cause the associated valve spring to go into coil bind, because it does not have sufficient tension. This will place severe stresses on the retainer and collets. Further, if the amount of lofting is in excess of the assembled piston-to-valve clearance, then almost certainly the exhaust valve will come into contact with the piston. From the spring technical data it appears that the spring can compress an additional 3mm before it is in coil bind. If the valve is lofted into spring bind (.120 inch [3mm] more than the actual lift of the cam lobe), then there would be interference of about 0.020 inch [0.5mm]) between the piston and valve. This situation is more acute for exhaust valve, due to the motion of the valve relative to the piston and would be worse if the cam were installed advanced 3-4 degrees to accommodate any chain stretch. The test data suggests that the valve lofting will occur between 7500 - 8000 RPM if the Schrick spring is used with the SLR300 camshaft.

Conclusion - If the engine was assembled with a valve-to-piston clearance of 0.100 inch (2.5mm), then certainly at 8000 RPM their is a very good likelihood that the pistons will clash with the exhaust valves. The problems associated with the engine of Customer B seem to bear this out.

Note: <u>Under no circumstances would I imply that there is physically anything wrong with the Schrick valve spring. This company makes very good products. It simply means that it is not the correct valve spring for a SLR300 camshaft. In fact, in other tests that I conducted with different camshafts with less duration, there appeared not to be a problem at all with this spring. The margin of difference is very small.</u>

<u>So what is the answer.</u>

After further modeling, if a valve spring with a rate of 220 lb/inch (3.9 Kg/mm) were used, then lofting is not longer an issue. Please note that there is a difference between the â€œspring rateâ€? and the actual seat spring pressure and the over the nose pressure. As a matter of safety, I would probably opt for a spring with a rate between 250-260 lb/inch (4.47-4.65 Kg/mm) to provide a little extra safety margin. This valve spring would have 52 lbs (23.6 Kg) of seat pressure (at 1.250inch [34.29mm] installed height) and 172 lbs (78 Kg) of pressure at 12.18mm valve lift.

This spring combination would be a 24% reduction in valve spring force, as compared to the spring supplied by Scuderia Topolino, and recommended by our cam grinder. This is not the complete story however. As the rocker arm ratio is 1.45:1 the reduction in pressure on the valve adjuster is close to 35%.

It goes with out saying that a reduction in spring pressure, as well as the attendant reduction in other parasitic friction losses associated with using a spring with less pressure, will mean an incremental increase in horsepower. Between 7500 and 8000 RPM this may very well mean 2-3 more horsepower is available to drive the wheels.

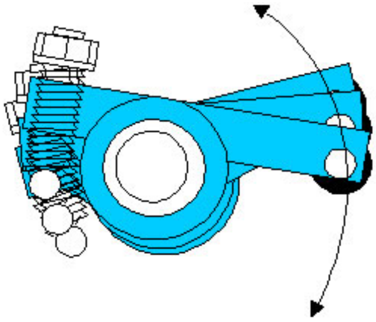
2. **Rocker Arm Geometry** â€" One of the misunderstood and mostly forgotten functions of the push rod is to adjust rocker arm geometry. After the cam has been reground, the head and block decked, new valves installed in the head, why would you assume the standard push rods are still the correct length? After all regrinding the cam lowers the base circle of the cam, thus lowering the position of the push rod to the rocker arm. Now, if you also machined the head 0.040 inch (1mm), and perhaps the block also 0.020 (0.5mm), then you will have compensated for some of the material ground from the base circle of the cam. The point is, you never know how much of each, so when the engine is first test assembled is when you find this out.

   As a good rule of thumb, when the cam is at 33% of its total lift (so for the SRL300 that would be 0.110 inch [2.8mm]) the adjuster should be in a straight line with the pushrod, with the adjuster showing no more than three threads below the rocker arm. So if 5 threads are showing, and each thread is .8mm, then you would require a push rod that is approx. 1.6mm longer than the test push rod. If however the cup of the push rod is right up against the rocker arm, then you will need a push rod that is 2.5mm shorter than the one you are using for the test.

   Note: The easiest way to measure the effective length of a push rod is to put a small ball bearing in the cup and then measure the overall length with the ball bearing. Now measure the diameter of the ball bearing and subtract this amount to get the effective length.

   You are not finished however. Next you must look at the rocker arm pad where it contacts the valve stem. Rocker arm geometry is generally optimal when the travel or movement of the rocker arm tip on the valve stem is minimized. To understand how to achieve correct geometry, it must be understood that the rocker arm tip itself travels in an arc. At zero lift, the rocker arm tip is expected to be closer (or inboard) to the plane of the pivot point and as the valve starts moving down, the rocker arm tip starts moving outboard. If the geometry is close to ideal, then the rocker tip will be at its most outboard position at half or mid lift at which point the rocker tip starts moving inboard again as the valve reaches full lift. Simply put, ideal rocker arm geometry is achieved when the rocker tip is sitting on the valve stem tip at the same position at both zero lift and full lift.

In a perfect world, where the rocker shaft pedestal stand locations, the valve guide, and the rocker itself are all machined to exact specifications, the rocker tip is expected to be sitting slightly inboard of the valve stem center at both zero and full lift while the rocker tip will be sitting the same distance outboard of the center of the valve stem at exactly mid-lift. As this is not a perfect world, this sometimes does not happen.



It may be that the valve is installed at a slightly lower installed height. If so a lash cap may be required. Alternatively, if the rocker pad does not sit in the correct location on the valve stem and the rocker fulcrum point has to be moved closer to the center of the stem, and so a shim will have to go under the rocker arms stands. If the opposite is true then the rocker stands may have to be reduced in height a small amount.

All of these actions, will affect the length of the push rod required. Take the trouble to do it, as it is worth the effort to get the geometry correct.

Note: The condition of having the contact pad too far extended over the valve stem, is the cause for the little half moon wear marks, that I discussed earlier. This causes the rocker arm to extend â€œoverâ€? the valve stem at full lift, and instead of depressing the valve stem it pulls the valve stem sideways toward the rocker arm fulcrum. This increases parasitic friction losses and causes premature valve guide wear.

SCUDERIA TOPOLINO when only the best will do

Cylinder Heads

5.1 Types of cylinder heads

There are both OEM type and aftermarket cylinder heads that can be used in combination with the various types of Fiat/A112 blocks. The head bolt pattern is the same, with the exception that some models used 10 x 1.25mm head bolts, while the majority used 9 x 1.25mm head bolts. It should be noted that Fiat is probably the ONLY car company to use 9 x 1.25mm hardware extensively.

OEM Heads

Fiat 600 and 600D - The early Fiat 600 head is distinquished by 8mm rocker stand studs and 6mm rocker arm adjusting screws. The later 600D heads had 10mm rocker arm stand studs and 7mm adjusters, as the early 6mm ones did not stand up well to the rigors of competition.

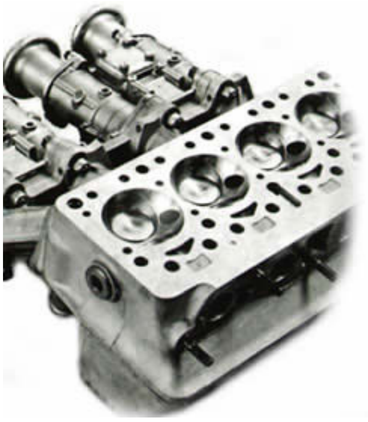
Photo – 600D head combustion chamber

Both 600 and 600D heads were used for most of the early Abarth coachbuilt automobiles. The head featured a deep “bathtub” type combustion chamber that required a piston with a “kidney” shaped dome on the piston to get the compression up. The valve sizes that would fit into the head were also limited, although at the time this was dictated by FIA rules and homologation papers. As engine displacements grew to 982cc, as for the 1000TC motor, this proved a real challenge.

Fiat 850 - The Fiat 850 head configuration was quite different. Still using a 2-valve side-by-side arrangement, the chamber was now much more open, with a good squish area to increase the combustion chamber turbulence as the piston came up to TDC. This same chamber configuration continued on and was eventually used for the A112 motor, with only minor modifications around the intake valve to assist flow.

850/A112 Combustion Chamber – Note that the chamber is not yet completely finished.

Abarth TCR and OTR - While considered and OEM head, the TCR and OTR heads did not look anything like the Fiat 600/600D or 850 heads. While the head bolt pattern is still the same, that is about where the similarity ends.



The general design of the head is a scaled down version of the head designed by Aurelia Lampredi for the Fiat 2300 saloon, and later adopted for the Abarth 2400 couple. It is still push rod operated, but with a hemispherical combustion chamber. Valves are operated by rocker arms, however intake and exhaust valves have separate rocker arm shafts. The TCR/OTR head is usually equipped with either dual Solex or Weber carburetors. The exhaust manifold is also different, as the port spacing and location does not coincide with either the Fiat 600/600D or Fiat 850 heads.

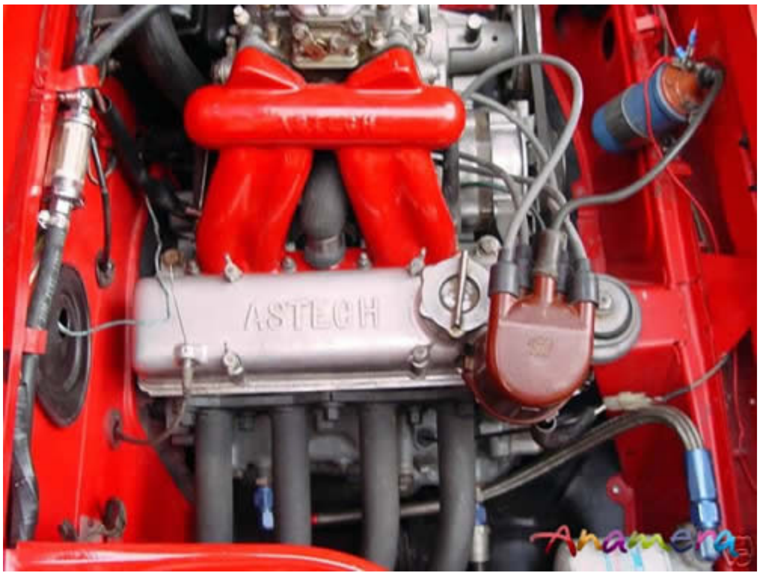
Aftermarket Heads

Vizza Motorsports

Vizza Motorsports Cylinder head

This head is a modified Fiat or Autobianchi standard head, and is sold by several people including Vizza Motorsports. The head has been welded up and then individual ports machined for each cylinder. As the ports enter the top of the head, the intake has to make a rather abrupt turn so that the dual Weber carburetors sit at the correct angle. I have also seen this head with two 40mm DCNF downdraft carburetors, and in this case the entry into the head is pretty much straight down. I have not had any experience with this head, but I would imagine the downdraft version would be more effective that the side draft version.

Astech

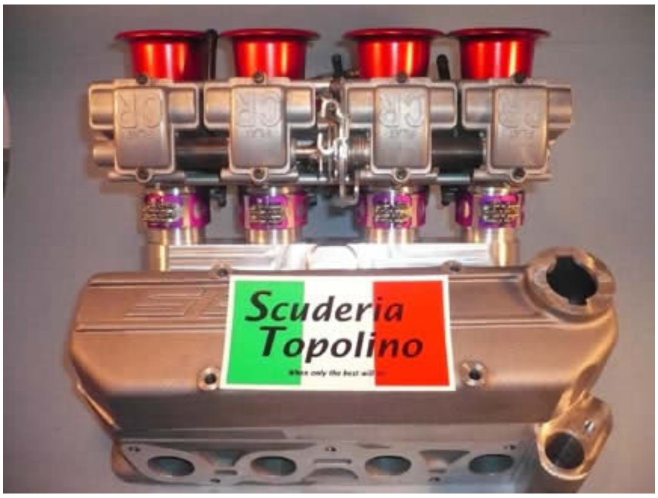


Astech Cylinder Head

This head was designed in the USA and produced for a number of years. It uses a log-runner intake manifold design for a single 40 or 45 Weber DCOE carburetor. The head is no longer in production. Given the design of the intake manifold, this type of installation would provide good torque and horsepower in the range from 4000-7000 RPM.

PBS 8P

Designed by Paul Swenson, this head has been in constant production since 1968. There was one revision of the casting models in the 80s to slightly change the head so that it would work on the A112 blocks,which had a different bore spacing. Scuderia Topolino is pleased to carry on the production of this head. The chief characteristics of this head are improved squish area, a "high angle intake port and revised cooling layout.



Here are two implementations of the 8P head. The first is a more conventional version with twin 40DCOE Weber carburetors. The second is a racing implementation using 4 Keihin FCR single barrel carburetors. In this case the carburetors sit at a 50 degree angle and the approach to the intake valve is very direct. Even with the Weber carburetors the curvature of the intake runner is very gradual, which is the reason for its good performance.

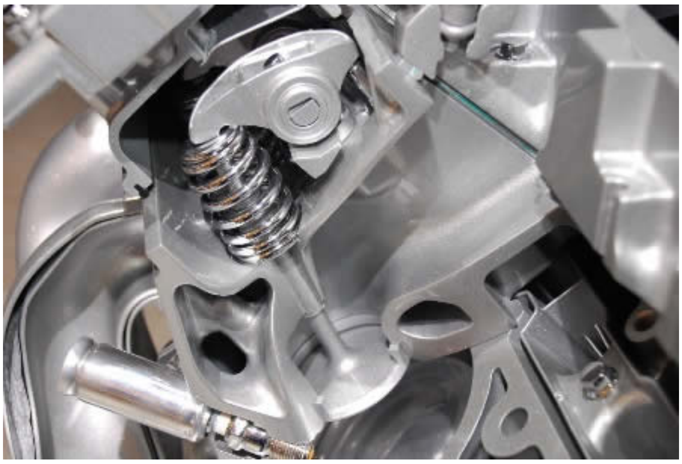
5.2 Head Layout and Modification

Valve size - It is possible to put larger valves in 850, A112, PBS and Vizza heads. I am not sure about the Astech head, as I have not worked on one. The maximum valve size is 32mm intake and 28mm exhaust. Using these sizes does pose some interesting problems. First, the intake and exhaust seats must be "nested" in one another (the exhaust is cut into the intake) and this can significantly weaken the seat area of the head. Second, by using such a large intake valve, the edge of the valve is shrouded by the edge of the combustion chamber and does not unshroud itself until the valves is nearly completely open. This causes a disruption in the flow of the valve. Third, using valve this large means that they "nearly" touch. If anything goes remotely out of sync, the valves could end up touching and this could be disastrous. The only way to rectify this is to move both the intake and the exhaust valve in the head. This would mean installing off-center valve guides, welding up the seat area, and cutting new seat pockets. This is a large, and exacting, job and not for the faint of heart. At Scuderia Topolino we consider 31 and 27mm valves for intake and exhaust respectively, to be the maximum size that we feel comfortable installing. Even then we go to some extra effort, on fully race ported heads, to take the combustion chamber wall out as far as possible. The shrouding of the valve is thereby kept to a minimum.

Intake port short side radius – Probably the most important area of the any of the heads, more so for the Fiat/A112 heads than the PBS 8P, is how you treat the "short side radius". As the intake charge enters the port, and subsequently flows around the valve head, the general idea is to induce a high helix swirl pattern. In fact there are probably several of these patterns being generated.

All too often we get carried away with making the hole in the head bigger without considering the short side radius. You want to leave the radius as large as possible, and make it smooth while removing the minimum of material. It is already a very short radius, so the last thing that you wish to do is lower the floor of the common chamber that feeds all four inlet valves. It would take a great deal of work, but it could be that there is advantage to be gained from actually raising the floor of the chamber slightly to that a larger short side radius is maintained.

Here is a good cross-section of a typical intake port. The one on the right has a larger physical port, but the one on the left will actually flow air more efficiently. The secret lies in the port floor and the short side radius. In the port on the left the floor is raised slightly, and the radius approaching the valve is larger and more rounded. The top of the port will still flow more air, but the ratio of flow between the top and the bottom is much smaller.



In the case of the PBS 8-P head, because of the 45 degree angle of the intake runner the short side radius is very well formed and the flow around the valve is almost equal.

The exhaust port on all of the heads, Fiat/A112/PBS is much the same . The exhaust port also has a very abrupt short side radius. However, because we are not simply relying on atmospheric pressure for getting the burnt charge out of the cylinder, the effect is much less pronounced. You still want to make sure that the flow radius is a smooth as possible, without any abrupt changes. For the most part the port at the exhaust manifold face is almost too big. Generally you want an exhaust valve that is between 83-86% of the intake valve. This makes the 27mm exhaust valve just right, as the actual valve seat diameter will be around 26.5mm. The remainder of the port is somewhat rectangular, ending at the exhaust manifold in a round shape that should be slightly smaller than the internal diameter of the primary tubes. This slight step will provide a small amount of anti-reversion and aid in exhaust tuning.

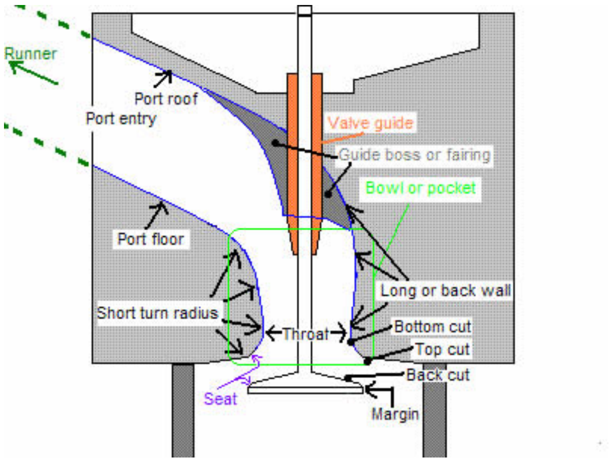
The Bowl – The next most important area is the shape of the bowl. This area can have a large impact on the swirl of the fuel charge as it enters the cylinder. Increased swirl generally means higher efficiency and increase flow. The disturbed characteristic of a swirling air mass also means that the fuel will stay in suspension better for improved combustion efficiency.

The bowl area also includes the portion of the valve guide that protrudes into the intake port. Some head experts like to remove this portion of the guide, while others like to leave a small amount of material there and use it as a small "air director" to get the air to assume a swirl attitude before it passes the valve seat.

The bowl on the exhaust port is no less important and should provide the best possible exit path. Due to cooling considerations, this may not always be optimum.

There is not set formula for how a bowl should look. It is one of those "black art" areas, and only a flow bench will partially answer the question of where to remove material. There are some things that we know work, but the last detail is often just a matter of trial and error.

Head seat configuration – If we can achieve a 83-86% intake/exhaust size ratio, then the next thing to consider is the seat angles in the head. The principal seating surface can be either 45 or 55 degrees.



Above you will find a typical intake port.

I then use a 45-55 degrees seat angle with a bottom cut to blend into the bowl area and a top cut to aid flow into the cylinder. If you go for a 55 degree seat, then you will have to run Beryllium or AMPCO copper seats, or you may have transfer of material from the seat to the valve surface, affecting the seal of the valve. For configurations where the intake valve may be shrouded this higher angle may be of some help, as it assists flow into the cylinder.

For the exhaust port I generally use a 45 degree seat, principally for reliability.

Valve seat and valve head configuration – This is a very complex interplay of components, theory, and practical experience. Each "expert" will have his or her ideas of what works.

The following link is a very good technical explanation of what you should try and achieve in modifying any head. While it is written around typical American production head work, much of it is directly applicable to Fiat heads.

http://www.tmossporting.com

Once you get on to this site, click on the Tech Articles link and go to the last item "Head Porting Principals".

5.3 Valve Lift and Flow

One of the best engine technicians was Smokey Yunick of stock car racing fame. He concluded from air flow observations that he made that to open a valve more than 33% of the valve diameter may not be productive due to the additional parasitic loss generated. Therefore for a 31mm valve, the maximum valve lift should be 10.9mm.

I know from my own tests on the flow bench that the "incremental" gains in air flow over 11mm of lift are indeed small. However the real advantage to higher lifts lies in the fact that in order to achieve these lifts a more aggressive cam grind must be employed. In doing so we increase the "area under the lift curve" in general, and because the lobes on the can are more aggressive, we increase the "rate of lift". This improves low lift flow.

Our biggest camshafts have 12.4mm of lift at the valve, and as long as low friction components are used in the valve train, and sliding friction losses are reduced to a minimum, additional benefit will result from the increased gross valve lift and rate of lift.

5.4 Head Gaskets

There are all types of head gaskets available and each has properties that are preferred for specific applications, but all depend on having an absolutely flat, clean and smooth head and block surface. This means a flat surface, at 90 degrees to the bore, with an RA (Average Surface Roughness) of between 14 and 20.

If you have a road car, with 10:1 compression, flat top pistons and a road camshaft, then any standard head gasket will be sufficient for your needs, PROVIDING you use the appropriate grade of fuel to prevent detonation. *Note: You will read a great deal more about the subject of detonation in the technical sections and the FAQ section, as it is a little understood, and often neglected, area of engine tuning.*

One step up would be to use the Spesso Competition head gasket. This is 1.6mm thick (uncompressed) and compresses to approx 1.2mm (0.048 inch). These head gaskets have special silicone beads around certain oil and water areas for improved sealing. In addition the "fire rings" are general made of stainless steel.

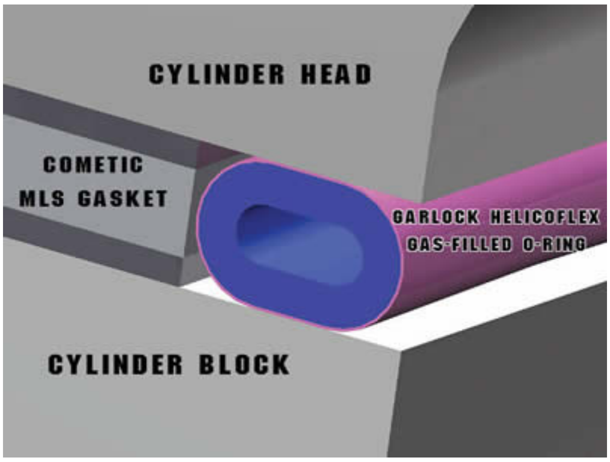
If you have a competition engine up to 12.5:1 compression, with a more aggressive camshaft with higher "dynamic compression" then you should consider the Spesso Competition or a Scuderia Topolino solid copper head gasket. The copper head gasket is 0.043 inch (1.09mm) thick. Care should be taken that the copper head gasket is properly softened (annealed) and coated with sealant on both sides before used.

The next step up is to use an Multi-Layer-Steel gasket, such as the one made for Scuderia Topolino by Cometic.

This gasket consists of a minimum of upper and lower stainless steel layers with a number of intermediate layers. In this way the gasket can be varied according the squish area requirements of the engine. The standard thickness for this gasket is 0.035 inch (0.9mm). The upper and lower surfaces are coated with a rubber material and not additional coatings or sealers are required.

Finally, if you have a very high compression motor (13.5:1 computed compression) with a very aggressive camshaft and high dynamic compression (10:1 or more), then all of the above options may eventually fail. In this instance you have an additional option.

The ultimate head gasket is a combination of two gaskets. It consists of a special MLS gasket with a separate 0.031 outside diameter stainless steel, pressurized o-ring, added as a pressure sealant, to contain the combustion pressures.



The "Helicoflex" o-ring is hermetically sealed during manufacture and has an internal gas pressure to maintain a positive contact against the block and head. As the ring heats up the pressure inside the ring increased, resulting in a higher pressure seal. The diagram above shows the combination of the two gaskets.

This combination of head gasket materials gives the best possible combination. The MLS head gasket acts as both a media sealer for oil and water, and a peripheral stop for the gas filled o-ring. The pressure filled o-ring acts as a super efficient pressure barrier for the combustion process. Another company that implemented this pressure ring technology is Coopers, who provided this technology for the Cosworth turbo-charged, FI motors.

If the piston is installed with "0" deck height, then the squish distance would only be 0.035 inch (.98mm). Given that connecting rods so stretch up to about 0.015 inch (.38mm), this would leave only 0.023 (.6mm) piston to head distance at high RPM. This would be the minimum piston to head distance for a 1050 motor, that I would recommend.

Postscript - Below are some photos of a head gasket that was destroyed by detonation. The detonation was caused by trying to run an engine with high dynamic compression(9.8:1) on 100 octane fuel. It probably would not have made any difference as to what type of head gasket was used, as even with a pressure ring the end result would have either been the same, or the next weakest link (probably the pistons) would have failed.

Three of the gasket fire rings have been deformed into a "teardrop" shape. This is a classic end result of detonation. In detonation the cylinder pressures would have spiked to over 3000 PSI. This amount of pressure would have lifted the head, releasing the clamping force on the head gasket and simply pushing it aside. The only reason the fourth cylinder did not suffer the same fate was that the car had already stopped running.

Cooling System

6.1 General Considerations

In any high performance, internal combustion engine one of the principle issues is managing the heat produced by the internal combustion process. This heat can be your friend, however it can also be a major contributor to three very destructive processes: pre-ignition, detonation and alloy temper degradation.

Even very efficient engines are rarely more than 35 % efficient in converting heat to motive energy. The other 65 % is lost either out the exhaust pipe, or has to be dissipated through the cylinder head, engine block and cooling system.

The circulation of coolant through both the cylinder head and engine block cooling passages provides the mechanism for transferring this heat to the radiator system. The cylinder head accounts for the majority of the heat generation and transfer to the coolant, namely 65%, whereas the engine block makes up the remaining 35 per cent. While these are numbers are estimates on my part, they are supported by the various temperature loads that are generated in the head and block respectively and will suffice for illustration purposes.

Within the cylinder head the "hot spots" are the areas directly around the exhaust valve seats and the area around the spark plug seats. These are also the areas that are likely to become super-heated. This super-heating is likely to result in the creation of super heated steam pockets. These steam pockets have three potential negative effects.

- If the steam remains trapped, then it will act as an insulator and this area of the cylinder head or engine block where the steam is located will prevent the transfer of heat to the coolant. This condition is "regenerative", and the steam pocket will eventually enlarge, and an ever-increasing overheating situation will occur.
- Once this steam reaches the water pump then it is possible for the entire cooling system to become ineffective, as modern centrifugal type pumps are not designed to pump steam, only liquid. In systems where the coolant temperature is very high, it is also possible for the water pump itself to boil the coolant on the suction side of the pump. As the pump rotates the "system" pressure is lowered, and thus if the coolant is sufficiently heated, the liquid may boil of its own accord and cause pump cavitation. This would lead to a catastrophic failure. Note: This "pressure drop" situation occurs when you open the radiator cap on a hot motor. *Instant pressure drop equals instant boiling.*
- Aluminum head castings are usually heat treated to T6 hardness specifications. This means that the head casting will have been heated to 1000F initially, then water quenched for a short period of time, and finally left in a heat soak oven at 320F, for a period of approx. 5 hours, and finally allowed to cool naturally to ambient temperature. They key is to make sure that the aluminum head does not exceed this 320F temperature in normal operation, as it will lose it temper. Once this happens then it will be very difficult to maintain proper head gasket clamping forces. In addition pressed-in valve guides and seats, particularly for the exhaust valve, will be compromised.

So the idea is to keep a sense of equilibrium within the engine cooling system, noting that it would be best to keep the average cylinder head temperature well below 300F.

Before going any further, let put to bed a myth *(I will admit that I fell victim to this early in my engine work).*

If the coolant flows through the radiator too quickly, it cannot transfer the heat to the radiator.

> *At a fluid velocity of "X" the coolant will release a given amount of heat to the radiator. If we double the velocity of the flow, it will release less heat to the radiator obviously because the coolant is in contact with the radiator for half the original period of time. Makes sense so far, right??*
>
> *However, since the water now makes* ***two revolutions*** *in the same period of time (remember we doubles the velocity of the flow), the net result is the same. So thermodynamic law says that in the same period of time the same amount of heat is released. This part of the myth is definitely busted*

One reason this rumor persists is that the 2 most often cures actually work sometimes. The first "cure" is to slow the coolant pump rotation with a larger pulley. The problem that is actually solved, in that many stock pumps will cavitate at even moderate RPM. A cavitating pump is very inefficient at best and may stop working altogether, at worst. Therefore slowing the pump the pump actually becomes more efficient. The second part of the "cure", restricting the outlet of the engine (supposedly slowing the flow), actually causes a pressure buildup behind the restrictor thus pressurizes the block by a few pounds more that the overall system pressure. Therefore, the boiling point of the coolant is raised by a small amount. This works to preventing local boiling in stagnant flow areas.

Remember that steam bubbles can slow, or stop, coolant flow through small passages. The idea of a restrictor is actually a worthwhile implementation in a competition motor. The increased pressure also creates some back-pressure and this, in turn, reduces the onset of cavitations. A more optimized solution would be to run the coolant pump at 50% of engine RPM and to place a restrictor (in a production car this would be a thermostat) wherever the coolant flow exits the block. By not overly slowing down the coolant flow we can introduce a level of "turbulence". This will have, as a side benefit, the ability to sweep some of the stagnant areas of any steam accumulation. So I guess that this part of the myth is at least "partially" true. Stewart Racing pumps actually built a coolant pump dynamometer and has demonstrated these effects. There is also a significant body of literature, within the Society of Automotive Engineers, addressing this very problem. I know that some of you reading this are probably saying, "OK, I understand all of this, but how does all of this get me more performance?" Stay with me a little longer, as it will all become clearer.

Next, we need to consider both heat generation and heat release mechanisms (engine and radiator). Let me walk you through a “mental” exercise. Consider the instance where the coolant flow-rate is just such, that the outlet temperature of the radiator is near ambient temperature. In essence, hot coolant (200F) going into the radiator, and ambient temperature (72F) coolant coming out. Not realistic but good for this exercise. This must be MAXIMUM EFFICIENCY for sure. **Wrong**.

Assuming that the coolant loses its heat linearly, the top part of the radiator, where the hot coolant enters the radiator, absorbs almost all of the heat, while the lower part, where the coolant is at, or close to, ambient temperature, the radiator absorbs almost none. The area in between works proportionally.

Similarly, in the engine, where the ambient coolant enters the block, maximum heat is absorbed by the cooling liquid. Where the coolant now exits the engine (at the cylinder head), only limited additional heat can be absorbed. Thus, the cooling ability is non-uniform. Worse, areas in the head that have the highest heat load (such as around the exhaust ports and spark plug areas) may suffer localized boiling. Once a film of steam forms, almost all cooling is lost.

Almost all current production systems utilize what I will refer to as a “bottom-up” approach to the cooling system. Coolant flow is moved by a pump, located on the cylinder block. This takes coolant fluid from the radiator and pumps it into the block first, then through the head gasket into the cylinder head and finally from the head back to the radiator system.

As discussed earlier both the absorption and release of heat is linear. This means that the “bottom” up approach severely compromises the coolant’s ability to deal with the more significant areas of heat generation in an engine, namely the cylinder head. As such, most cooling systems are much larger than actually would be required, were the system biased to where the majority of the heat is generated. We only have a given temperature delta (Δ) to work with within a closed system. This is the difference between the input and output temperatures of the coolant as it enters and leaves the radiator.

In terms of proportional heat generation, the block cooling passages account for about 35% of the heat generation, whereas the cylinder head cooling passages account for the other 65%. In a “bottom up” arrangement, by introducing the coolant into the block first, and thereby drawing a significant portion of the block heat into the cooling fluid first, we have already narrowed the temperature Δ available to deal with the much more substantial heat load generated by the cylinder head cooling passages. This will increase the likelihood of steam pocket formation at higher load levels encountered under racing conditions.

It may be that there is significant justification to consider adopting a “top-down”, reverse flow, approach to the cooling of the Fiat /A112 competition engine. This idea is far from new, although contemporary developments have lent some new insight. Pontiac, as early as 1956, used a coolant pump to pump water in through the heads first, and then to the block.

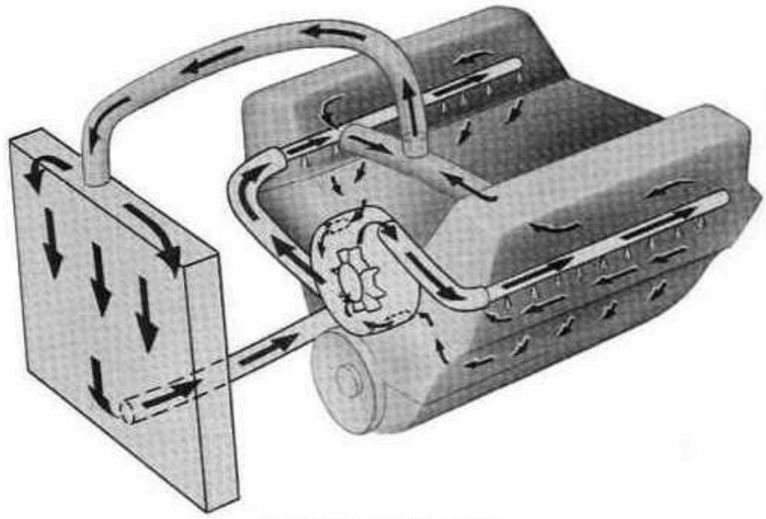


Fig. 6A-1 Engine Cooling System

More recently, John W. Evans filed a patent (5,255,636)) that makes very interesting reading. I may be more than coincidental that Mr. Evans has a pending, large lawsuit against General Motors for using his cooling ideas on the LT1 Chevrolet V8 installed in the latest Corvette. This motor is capable of running at 10.5:1 on unleaded gasoline, with far greater spark advance than normal.

You might ask, “Just how much advantage can be gained by reversing the coolant liquid flow?” In terms of overall cooling performance the percentage change may only be in the high single digits, but there are other “follow-on” benefits that are much more substantial. Since the coolant is now first going to the head (the place where the most heat generation takes place) there is the full coolant temperature Δ available to deal with the hottest part of the motor. Since detonation can take place at temperatures as low as 190F, controlling the areas around the exhaust seat and the spark plug seat provides large dividends in terms of minimizing steam pocket generation, but also allows more distributor advance and compression to be used. Head gasket surfaces are also likely to run at more even temperatures. If the overall temperatures at the cylinder head/gasket interface could be reduced by 10%, this would be a substantial additional safety factor.

A further benefit is derived with regard to the engine block. As the engineers at Chevrolet discovered, because of the relatively even temperature distribution, of the coolant entering the block from the cylinder head, the block was heated much more evenly. Even thought the temperature was slightly elevated, this turned out to be an advantage. In a “bottom-up’ cooling system, as low temperature coolant enters the front of the block and has to work its way to the rear cylinders, there is an inherent uneven heat distribution, with the rear cylinders always running hotter than the front ones. While with “top-down” cooling it is uniformly higher and therefore promotes uniform expansion. This assists in better ring sealing and lower piston/cylinder wall friction, all attributes that are important in any competition engine.

In reading Mr. Evans' Patent, it is obvious that some careful thought went into the design of the system, particularly for a road car. Here you have to account for quick warm-up and providing coolant to the cabin heater core. In principle the system is pretty simple. Here are the items that are required.

1. A pump capable of reasonable coolant velocity so as to keep the interior surfaces of the head cooling area and block cooling area well scrubbed of steam bubbles, yet not rotating fast enough to cause serious cavitations.

2. A method of routing any steam bubbles in the top of the head, away from the cylinder head and condensing them back to liquid, to then join the normal fluid flow again. The Patent information is informative in this regard.

3. A restriction on the output of the block, either in the form a thermostat, bi-directional thermostat, or a restrictor plate.

4. A condensing unit.

Items #2 and #4 are THE SECRET to making this work. Anyone familiar with refrigeration, or air conditioning, design will quickly realize what is required. A small line, or a larger line with an internal restriction orifice, must be attached to the top of the A112 head and then routed to the remote header tank in the engine compartment. Is that all there is to it?

Well, yes. You see, as the coolant pump is now pumping into the top of the head, and there is a restrictor on the side of the block (where the current water pump sits) there is a pressure differential between the coolant in the block/head and the coolant in the remainder of the system, including the header tank. By forcing the steam/coolant through a small orifice, it will be subject to a pressure drop and the surface area of the line, as well as the overall surface area of the header tank to which the other end of the line is attached, is sufficient to cool the steam and cause it to condense to a liquid state. If there was any doubt as to this you could always make a header tank that incorporated a finned heat sink. This is the same principle that takes place in a modern air conditioning or refrigeration system where gas is converted back to liquid, after carrying away heat.

Another option-

Another way to deal with the same problem would be to use a "dry deck" configuration. In this implementation a head gasket would be used that had NO water passages. Water would be channeled into and out of the head first and then routed to the block. This may also be a viable solution as no steam venting/condensing port for the head is required, although one could be implemented as a safety measure. Overall the amount of heat load to the radiator system would be the same as a conventional cooling system. This type of system would likely need an electric pump, remotely located from the engine.

I will not go into how I would implement these systems in detail, but I believe they hold much promise in being able to run higher dynamic compression ratios, and greater ignition advance, without the onset of detonation.

Acknowledgements:

John W. Evans – US Patents 5.255,636 and 4.550,694

John De Armond – Hot-Rod Magazine

6.2 Radiators

In modern vehicles radiators are most oftem made of either all aluminium or a combination of an aluminium cores with plastic end tanks. For competition purposes I do not recommend the second type, as the stress placed on them by racing may cause a premature failure.

For competition purposes radiators can be made from either aluminium or brass. The heat dissipation qualities of the two metals is similar, with of course the aluminium radiator weighing less.

Careful attention should be paid to the number of fins on the core tubes, and the overall thickness of the radiator. If the fin spacing is too close, or the core is too thick (or perhaps both), then insufficient air will travel through the radiator core. Radiator efficiency depends on air traveling through the core.

If you are going to run a front and rear radiator, then the flow should be as follows:

Engine thermostat housing to top tank of rear radiator
Bottom tank of rear radiator to top left of front radiator
Bottom right of front radiator to input to water pump
Output of water pump to input of engine.

If you are only using a front radiator on a rear engined car, then the following would apply:

Engine thermostat to top of rear mounted expansion tank
Bottom of expansion tank to top left of front radiator
Bottom right of front radiator to input to water pump
Output of water pump to input to engine

In both cases there should be a small, manual valve installed on the top left tank of the front radiator to allow for venting of any trapped air during system filling. As an alternative, a small tube could be run from the top left tank of the front radiator to either the top tank of the rear radiator or the top half of the rear mounted expansion tank. In this way and air or steam bubbles trapped in the front radiator will automatically be dealt with.

6.2.1 Air Flow – Radiators require air flow. The more flow the greater the efficiency of the radiator. This also means that all the air "caught" by the radiator should be made to go through the radiator, rather than around it. Air will always take the course of least resistance, so if there is a gap around the radiator, it will always flow around it rather than through it.

The radiator should be shrouded, to direct the air through the core. This hold true for systems where the movement of the car forces air through the radiator, as well as systems where the air flow is wholly dependent on an air conveyor as in the Fiat 600 and 850 type cars. Air conveyor systems do depend on engine RPM, and therefore may not be able to handle extreme competition temperatures as well.

6.2.2 Radiator Sealing – Most modern radiators have a radiator cap, and this includes Abarths and Fiats. What is generally not known is that the radiator neck on Fiat cars is different to most other cars. It is slightly deeper. Therefore the use of an aftermarket cap, even if it says 22 lbs on it, will generally only result in a 3-4 lb cap. So, if you are using a standard Fiat radiator, use a standard Fiat cap. It will at least give you 12-14 lbs of system pressure.

The other alternative is to have the radiator neck replaces with a standard aftermarket one. Now you can run any high performance radiator cap you want.

Here is an illustration of several stant competition caps that range in pressure from 19-22 lbs. While on the subject of radiators, all hose connections should have a raised lip on the end of the spout, so that once the hose is secured with a clamp, if cannot slide off..

The importance of being able to run higher system pressures will become obvious later in this section.

6.2.3 Radiator Hoses and Clamps – At a minimum I would recommend replacing the radiator hoses at least once each racing season. These items are often neglected and forgotten (at least until one fails and a motor is ruined). These hoses should be installed with NEW screw type hose clamps, and then the hose clamp wrapped in 3-M self-bonding electrical tape.

Other hose connections have been developed specifically for high performance purposes. These in include both fabric and stainless steel jacketed material with AN type screw on fittings, and Wiggins type connectors for joining and terminating solid cooling tubes.

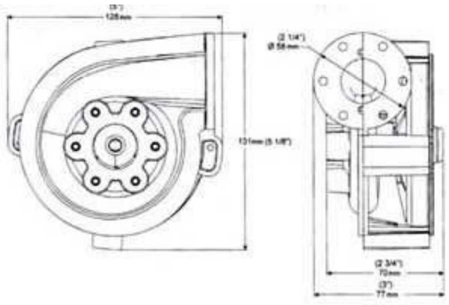
 

6.3 Water Pumps

Mechanical – All of the mechanical water pumps currently available for the various type of Fiat blocks (817.843,903, 965, 982, 1050) are quite sufficient in terms of transport of fluid, even if using a remote front mounted radiator, provided they are properly driven.

By this I am not referring to whether you use a v-belt or some type of toothed belt, but rather to the size of the respective crankshaft and waterpump pulleys, and the ratio between them. The crankshaft pulley should always be 50% the diameter of the waterpump pulley. This will insure that the waterpump runs at 50% of the engine RPM. This will not only mean that it uses less horsepower, but also that it be less likely to cavitate.

Electric – There are now several type of electrical water pumps available that can be adapted to the Fiat/Abarth type motors. Some are meant to run at constant speed, whereas others incorporate a “controller”, to regulate the speed of the pump. With a controller the water is only enough water is moved to maintain a given, preset water system temperature. I know of one competitor who has quite successfully used two “water guppy” type water pumps in his race car for over 10 years, so the idea is not at all new.



Technical Specifications:

Operating Voltage	4V DC to 14.5V DC
Maximum Current	7.5A
Flowrate	20L to 80L/min, (300 gal to 1300 gal/hr) 13.5V DC
Operating Temperature	-20C to 130C (-5 F to 270 F)
Pump Design	Clockwise centrifugal with volute chamber
Motor Life	2000 hrs continuous at 80 C (180 F) and 12V DC
Pump Weight	900 grams (2lb)
Pump Material	Nylon 66, 30% glass filled
Maximum Pressure	tested to 50 psi
Fits Hoses Sizes	32mm to 50mm (1-1/4" to 2")

Oil Coolers

Lubricating oil has two distinct purposes. Obviously, one purpose is lubrication, however a second purpose is to carry away heat from vital components within the engine. Components such as cam bearings, main and rod bearings can only stand so much heat before they fail. Likewise, boundary layer interfaces such as rocker arms and rocker shafts, also rely on sufficient oil flow though the interface to carry away heat.

So oil is an important "vital fluid" and its temperature must be maintained at an operating level that keeps it from breaking down. The "terminal temperature" for most organic oils used in automobiles is around 300 degrees Fahrenheit (148 deg. Celcius). For synthetic oils this number is slightly higher, however anything over 350 degrees will get dangerously close to the "flash point" temperature of most oils.

Fluid –to-Air Coolers

The most common oil coolers are similar to a conventional water radiator, only smaller. The idea, for any oil coller, is to provide enough temperature delta to maintain an oil temperature between 220-270 deg. Fahrenheit (104-132 deg. Celcius). In the fluid-to-air oil cooler oil is fed through a number of tubes, over which air flows to carry away the heat. To assist in heat dissipation, the tubes generally have small, thin fins attached.

Generally, oil from the oil pump should be fed to a filter first, and then to the oil cooler, before returning to the main oil gallery of the motor. In this way the cooler is protected from debris, and cooled oil is presented directly to the camshaft, crankshaft and connecting rod bearings.

Fluid-to-fluid Cooler

This type of cooler is better referred to as a "heat exchanger", and has been used on the maritime industry for many years. Here water, (fresh or salt water) is routed through the large fittings, and oil is routed through the AN fittings. This is then 2-way system.

In a racing application several different versions have been utilized. In one implementation the oil cooler element is placed in the low delta side of a standard aluminium radiator. Thus, using a dual pass radiator, the first pass of the cooling fluid through the radiator sheds much of the engine generated heat, then the water passes around the exchanger element in the left header tank and dissipates the oil system heat in the second pass through the radiator before returning to the motor.

The unit above is a self-contained exchanger and a portion of the water flow, at the low delta temperature point returning to the motor, is fed through the exchanger.

The beauty of this type of oil cooling method is that the oil cooler can be placed almost anywhere with in the vehicle, obviously keeping it as low as possible, and it does not need to be in the air stream in order to be effective.

In the original Abarth TC and TCR implementations there was a dual radiator, placed in the front-mounted radiator shroud. This required both water and oil fluid lines to be routed from the back of the car to the front, and back again. Using a self-contained heat exchanger could save some weight in eliminating one set of the long lines to/from the front of the car.

Both types raise the overall water temperature slightly, so it is important to make sure that the water cooling system is up to the additional task with sufficient temperature delta to handle the extra load.

Induction and Exhaust Systems

7.1 Induction and Exhaust Systems

Making more horsepower is all about getting more air and fuel into the cylinder. Now there are many ways to accomplish this, some more technically challenging than others. Put differently, horsepower is all about optimizing "volumetric efficiency" (VE).

Since almost all historic racing clubs do not allow the use of turbocharging, supercharging or fuel additives, I will not spend a great deal of time on these issues except just to review how they affect volumetric efficiency.

Turbocharging and Supercharging - Here the idea is to simply cram more fuel mixture (air and gasoline) into the cylinder under mechanically induced pressure greater than one atmosphere. Using these methodologies the only limiting factor is how much additional fuel and air can be reasonably burned, and this is usually dictated by the ability of the engine components and the cooling system to handle the additional heat load.

Fuel Additives - There is a distinction that has to be made here between those chemicals that can be added to fuel to increase resistance to knock, and those that can be added to increase VE. The best known of these is MTBE which has an oxygen content of 18.2% and is in use in California in reformulated lead free gasolines sold to the public. Others are Methanol (49.9% oxygen) and Ethanol (34.7% oxygen). These are all readily miscible with gasoline. There are also other chemical agents such as Propylene Oxide and Aniline, both known to be very dangerous and require special handling, that can be used as octane enhancing additives. Some of these additives have extremely low boiling points, such that on a hot day over 95 degrees Fahrenheit they would simply boil on there own. This adds to the danger of their use, as in a vapor state they become highly volatile and explosive. If you choose to use these take special precautions. Quite obviously, additives like Nitrous Oxide and Nitromethane will also provide a performance boost, but these are easily detected.

7.2 Intake Manifold Variations

Now lets get back to what can be done to increase volumetric efficiency through non-chemical means. The Otto cycle combustion process is all about "timing". This includes cam timing, valve opening timing, and as you will see intake charge timing. Just as a properly designed exhaust system utilizes the "scavenging effect" to extract spent gases from the cylinder, the same effect can be put to good use in maximizing the amount of fuel/air mix goes in to the cylinder. This will deal with a number of issues in the fuel induction system.

The whole idea is construct an intake arrangement whereby the intake pulses arrive at the intake valve just before the valve opens, at the desired RPM range. This has a supercharging effect which increases the amount of air/fuel entering the cylinder by as much as 20%.

To examine this we will look at three areas:

- Intake port diameter and length
- Plenum volume
- Ram Pipe length and Diameter (Helmholtz Theory)
- First intake port diameter and length - In general the diameter should be approximately 85% of the valve seat diameter. While this is am approximation, it is pretty close as this is based on assumption that the valve itself will provide some restriction even at full valve lift. If therefore the valve seat diameter is 31mm, then the port diameter as it approaches the valve seat should be around 26mm.

There are two types of heads in popular use on Fiat Abarth rear engined cars, the standard/modified head and a 8-port aftermarket head like the PBS 8P.

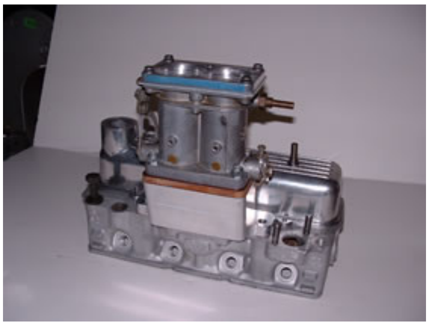
Now on a PBS head, with individual intake port runners, you have the ultimate flexibility in setting up your intake system. There might be a temptation to say that "big is better", but be careful. Large diameter ports may seem like they would flow a large volume of air, but they may lose a great deal of their velocity. I believe that in order to maintain low/mid range performance that the intake runner should maintain a certain amount of taper so that just before the intake valve seat the port is around 27mm, then blending out to the 31mm seat diameter. This means that the air will travel a path from the velocity stack bell (40mm) through the secondary venturi (30mm), through the throttle plate (40mm), down the intake runner to an area just before the valve (27mm) and through the valve seat. I have calculated that the theoretical length of the runners to achieve best performance around 7000 RPM should be 8-9 inches. The 8P design comes very close to this, and it would be difficult to change it. The design of the PBS 8P head promotes good midrange performance, while not restricting top RPM operation. I know of many engines that reliably run up to 9000 RPM.

The standard Fiat head is a different engineering exercise entirely. In reality we have an arrangement of two tri-Y intakes. If we divide the head into the right and left side, then inlet port of the head forms basically the bottom leg of the "Y" and the two short runners to the intake valves form the upward extensions of the "Y". This is identical for the left and right hand sides of the head. Just like in an exhaust manifold, this tri-Y arrangement has the effect of broadening the optimum RPM range over which the intake system is most effective. The intake manifold should be considered as a "plenum volume" common to all four cylinders. Unfortunately, as this head was designed originally for a 27 horsepower engine with limited RPM capability, the standard dimensions of the head do not lend themselves to a high performance application. It is possible to apply the port diameter theory in terms of making sure that the port diameter immediately prior to the intake valve seat is around 27mm. This will induce a secondary venturi effect just before the intake charge enters the cylinder. As far as port length goes, this is a combination of the two parts of the "Y" on each half of the head, being the very short individual port sections and the longer bottom of the "Y" section. Even so, this is not as long as ideal. As each cylinder that forms part of the Y fires alternatively to the other, the entire runner (both parts of the "Y") may be considered as the runner length for either cylinder. One could argue that if you had a two barrel carburetor with both venturis opening at the same time, then you could separate the two tri-Y areas. As we will see later on this may not be the case. The opening in the head should be of such dimension as to insure that adequate air flow is present and should match that of the bottom of the intake manifold/plenum.

Plenum Volume - Plenum chambers are designed to diminish the pulsing effect of the intake system, and this is particularly effective when more than one cylinder is fed from the same plenum. Plenums are not generally effective for more than 4 cylinders, although multiple plenums could be used for 6 and 8 cylinder engines.

For an 8-port head like the PBS unit there really is no plemum at all. Each port/intake runner combination acts as a standalone system. It is my view that this could be enhanced by adding a connection between the four intake runners just prior to entering the head itself. This would not be unlike the multiple tuned port arrangement in the BMW M44 engines, where an opening between the ports is opened at a certain RPM. This would then modify the effective RPM band to provide most efficient operation over a larger RPM band. Therefore the tunable range of the intake system of the 8P head will be fairly tight.

For Fiat Abarth heads, based on the standard Fiat head, the intake manifold provides for plenum chamber of sorts. To be ultimately effective around 7500 RPM the volume of the plenum chamber should be around 400cc on a 1000cc motor. This is the ideal number but is not critical. The plenum has the dual effect of dampening down intake pulse effects and also adding effective length to the intake runner. It is this very combination of intake port and intake manifold that then make up the effective port length that allow engines with standard heads to produce good power right up to 8000 RPM. Something that the original designer I am sure never intended.



Ram Pipe Length and Diameter (Helmholtz Effect) - The Helmholtz Theory was originally derived based on the harmonic effects of audio. He had postulated that a tone, or noise, was a combination of a primary frequency and a number of other secondary additive audio frequencies. Others applied this to fluid and plenum theory. There are three portions to the Helmholtz theory, namely plenum volume, ram pipe diameter and ram pipe length.

In the case of the PBS 8 port head the Helmholtz principle can be applied to each runner of the intake manifold. As such, about the only part that can be conveniently changed is the length and possibly the diameter of the velocity stack. Most velocity stacks concern themselves with providing a smooth air entry into the carburetor by ensuring a proper minimum radius on the bell of the stack. It may be that there is some advantage in actually making a velocity stack in which the bell opening, while having the suggested entry radius, might actually be smaller than the diameter of the secondary venturi. This would in effect increase the velocity of the air entering the carburetor and the adjustment in length would insure proper timing arrival of the pulse. Even with the PBS intake manifold there is a small difference in overall runner length for the outside cylinders, so one might consider using slightly shorter velocity stacks on these cylinders.

For the standard Fiat Abarth head, using a 2 barrel downdraft carburetor, the Helmholtz principle may have some further implications. In essence the volume of the area below the throttle plates, up to the back of the valve head is a plenum chamber. Therefore the velocity stack on the caburetor, along with the area of the secondary venturi in each carburetor throat makes up the entry to the "plenum". We can vary the tuned entry length both in diameter and length to get greater effectiveness at certain RPMs.

There may be an opportunity for further gains by employing yet another plenum chamber which would enclose the top of the carburetor and the radiused carburetor throat entry. Again, this plenum chamber should at a minimum be 500cc in volume on a 1000cc engine.

There are some basic generalizations that can be applied.

- A longer the intake tract will work better at lower RPMs
- A smaller diameter intake tract will promote better torque at lower RPMs
- A shorter intake tract will work better at higher RPMs
- A larger diameter intake tract will promote better HP at higher RPMs

Therefore, we need a good combinations that will promote good torque at around 5000 RPM, yet still provide good HP in the 8000-9000 RPM range.

7.3 Exhaust System Design

Exhaust systems, sometimes referred to as "headers", have been a subject of fascination for me for some time. Like other things, the design of header systems is sometimes is regarded as a "black art". However, when examined closely there are some basic rules that can be applied. This is not to say that the formulas and ideas that I am to present are the absolute truth on the matter, as every engine will have its own particular characteristics. The concepts put forward will however get you close to an optimum exhaust system.

Before I go too much further I have to give credit for much of the technical material in this article to the authors of a 1966 Hot Rod publication called "Supertuning". Bill Allen, the 1969 SCCA D-Sedan National Champion, used the same information to design an exhaust for his winning NSU. As the displacement of that car is similar to that of the Abarths that we still race today, I thought it a good place to start.

On all engines, including Abarths, a tuned exhaust system provides significant advantage, but this advantage is limited to a particular RPM band, depending on the design. Making a good tuned exhaust is not particularly difficult. The object is to get the reflected wave of one cylinder to help scavenge The next cylinder in the firing order. To better understand this it must be understood how waves are generated. The primary exhaust has a very high positive value. When this wave reaches to end of the exhaust pipe it is reflected back up the exhaust pipe as a negative pressure wave, which in turn is reflected again when the exhaust valve closes. Given the right timing, this secondary wave will help in extracting the residual exhaust gases. This wave movement provides further benefit in engines with camshafts with long duration and large overlap. In this case the intake and exhaust systems can drop below atmospheric pressure. When this occurs, the fresh mixture will begin to flow into the cylinder even before the piston initiates its intake stroke. This contributes to Volumetric Efficiencies in excess of 100%.

The real problem is in determining when the primary exhaust wave reaches full force, There is no sure way of determining this, but there is a fairly good empirical formula. I apologize in advance for not providing metric equivalents, but it became far too complex I should also mention that this model only deals with 4 into 1 type systems, although the computations are fairly close for 4:2:1 systems as well. It is:

L= V120/N

L is the length of the exhaust pipe primary from the head of the exhaust valve to the end of the merge collector (not including exhaust pipe)

N is the desired peak RPM

V is the speed of the wave in the exhaust gases (1700 ft/sec)

A "tuned exhaust" is generally only resonant over a relatively small RPM range, about 1500 RPM. If we consider that 8500 RPM is our absolute maximum, then subtract 1500 RPM, this leaves us with a midrange RPM of 7750. This would make the effective range from 7000-8500 RPM. At all other times the exhaust manifold would be less than 100% efficient. You may prefer to tune the exhaust somewhat lower in the RPM band, as with most Abarth engines peak torque is achieved at around 5500 RPM.

Let's model an Abarth 1050 motor as an example. Using the above formula with a peak 8000 RPM, we find that the ideal primary pipe length is 25.5 inches. Remember that this overall length includes the exhaust track in the head and the length of the merge collector, but not the secondary exhaust pipe. This is an approximation, but it will get you close. If you are using a cam with a lot of duration (say 300 degrees or more) then the pipe should be 1-2 inches longer.

Of great importance is the diameter of the primary pipes of the tuned exhaust. The diameter of the pipe will directly influence the velocity of the gas flow (note: not the wave) . The diameter should be such that, at the engine's power peak, the mean velocity of the exhaust gases is about 300 ft/sec. At this speed there is a balance, between the internal friction on the pipe on the one hand and the benefits of increased gas speed on scavenging on the other. When gas velocity is high enough, the sudden rush of exhaust products from the cylinder will tend to pull much of the residual gases along, and may even leave a slight vacuum in the cylinder, which will further aid cylinder filling during the intake stroke. If the primary pipes are made too small the the gas speed becomes so great that that the scrubbing along the inside of the pipe impedes gas flow and creates back pressure. The 300 ft/second point appears to be a good compromise.

The next step is to calculate the gas speed. Here is the formula:

V= (piston speed/60)X(D2/d2)

V is gas velocity in in feet per second

D2 is the piston diameter squared

d2 is the inside pipe diameter squared

Piston Speed is in feet per minute

Now then, for our Abarth 1050 engine lets start out with the following characteristics: 8000 RPM, 2.913 inch (74mm) stroke and 1.25 inch (31mm) pipe diameter. To determine piston speed in feet per minute we take the stroke times two (remember there are two strokes, one up and one down, for each turn of the crankshaft), namely 5.826 inches. This is them divided by 12, being the number of inches in a foot. This gives a result of .485 foot. This must then be multiplied by the desired crankshaft speed of 7500 RPM to give the piston speed. This computes to 3884 feet per minute or 64.73 feet per second. We then finish off the last steps by squaring the piston diameter of 2.657 (7.059) and the pipe 1.25 inch (31mm) pipe diameter squared is 1.562 inch (39.7mm. We proceed then by dividing 7.059 by 1.562 for an answer of 4.519, then multiplied by the piston speed of 64.75 ft/sec. The exhaust gas velocity computes to 292 ft/second.

Well, this is within 8 ft/sec. of our target. Almost perfect!!

Therefore we can now indicate that a 4:1 exhaust system for an Abarth 1050 motor having a 67.5mm bore and a 74mm stroke, with a mean peak RPM of 8000 RPM, will require exhaust primaries of between 27 and 29 inches, 1.25 inches in diameter. This would give a peak power RPM range between 7000 and 8500 RPM.

As the primaries come together in the collector, care must be taken to make sure that the tubes are fitted so that the next firing cylinder is adjacent in the collector. This will greatly increase the scavenging effect. Once in the collector, the exhaust pipe can be a minimum 15-20 inches in length. You will only find the "right" length by trial and error. There are different schools of thought as to whether the exhaust “pipe” should be a megaphone, or straight. Generally a tapered exhaust pipe will promote a slightly wider power band. You will remember that Abarth used such an exhaust megaphone on many of his race cars. On some engine designs, for sake of ease of installation, you may find having to use a “resonant multiple” for the exhaust pipe. So if a pipe of 22 inches (560mm) works well, but does not exit the car, then the next choice would be 44 inches (1120mm). The TCR exhaust is a good example of this.

A word of caution!! There is a temptation to make the exhaust primary tubing too large in diameter. Bigger is not always better, unless some thing else has been compensated for.

There we have it. Easy, right !! Well actually the math is easy. Now comes the hard part of finding the right tubing sizes, and making sure that the final design has equal length tubes, and will actually fit inside the engine compartment.

A Note of Interest - I went back and computed a complimentary intake length for this exhaust system and it indicates that the total length of the intake path should be 13.2 inches (33.5cm) from the bell of the velocity stack to the back of the head of the intake valve. If you look at a 1000TC intake manifold with a 36DCD7 carburetor and velocity stack, I believe you will find that this is very close to what Abarth used.

7.4. Carburetion and Fuel injection

7.4.1 Carburetion

The standard head, be it a Fiat 600/600D 850 or A112, have all been equipped with various 2 barrel, downdraught carburetors. The two manufacturers most often used as Weber and Solex.

The most often seen Weber models are 36DCD7 and DCN/DCNF carburetors.

The 36DCD7 can be found in two versions. One has both venturies opening simultaneously, whereas the other version has a progressive secondary. I have had experience with both types. At full throttle there is virtually no difference in performance, however the progressive carburetor may have a slight advantage in terms of fuel economy at partial throttle.

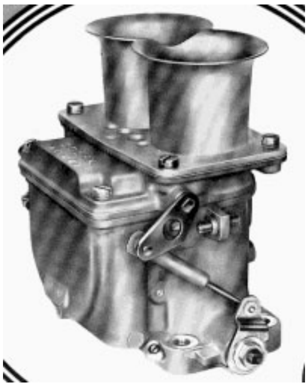
The DCN series of carburetors come in various varieties and have and have been used for many years on marques such as Ferrari and Maserati. For these engines the carburetors were cast in lightweightt aluminum, whereas most versions are cast from the conventional alum/zinc mix. All DCN type carburetors are non-progressive

Both of these types of carburetors should be mounted with phenolic spacer, to isolate it from heat generated by the cylinder head.

A good starting point for jetting would be as follows

28-29mm venture
130 Main
175 Air
45-50 Idle

For a number of years Solex supplied a carburetor specifically suited for competition purposes. This unit, the 36-40CCI carburetor was designed for competition. Again this is a non-progressive design, made from lightweight aluminum alloy.



These carburetors are always popular and when they do become available they are quickly purchased. They have been out of production for over 25 years, but they are still one of the best designs.

Abarth equipped the TCR head with dual Weber 40DCOE carburetor. This is probably the most widely implemented carburetor design for competitive applications.

With an individual throttle for each cylinder, this provides the ultimate in fuel control. The PBS 8P head, which also incorporates individual runners, is perfectly suited to use these carburetors.

A good starting point for jetting would be as follows

29-32mm venture
125 Main
180 Air
F11 Emultion Tube
45-50 Idle

There are other carburetor types that have been used successfully on Fiat/Abarth competition vehicles. These include Del'Orto units from Italy and various flat slide carburetors.

7.4.2 Tuning Weber Carburetors

All Weber carburetors have four circuits.

Idle
Cross-over
High Speed
Accelerator

The Idle and Cross-over circuits are inter-connected, in that the idle jet is responsible for the fuel supply for both circuits. It provides fuel to the idle mixture screw (responsible for idle supply up to about 1500 RPMO), and the crossover ports, which are hidden by the butterfly and come into play as soon as the throttle open, because the incoming air literally "sucks" the fuel into the airstream. The cross-over ports provide additional fuel in this critical phase and If the idle jet is too small, then there is insufficient fuel to satisfy the needs of both the cross-over ports, and of course the idle mixture screw as well, as both circuits are active simultaneously. This mechanism take care of the "sudden" rush of air when you open the throttle butterfly. If the fuel is not sufficient at this point, the engine will run lean and stumble momentarily. Of course once the butterfly is fully open, the vacuum on the intake tract drops, and the motor is running on the main jet and air corrector jets. The crossover ports and the mixture screw have done their job and are no longer active, at least until you close the throttle and it all starts again.

If the idle mixture jet is correctly sized, then the idle mixture screw should provide a stable idle when it is 2-4 turns from being fully closed (seated on the seat) at an idel speed of 1000-1200 RPM. Be careful when you screw it fully in, on the seat, so as not to damage anything. I suspect with the large venturi that you need to change the idle jet to the next larger size. It probably has a 45 in it now. I would change it to 50, or perhaps even a 55. Then you can back off the idle speed screw and you will then have to reset the idle mixture screw and it should come back to 2-4 turns from fully closed. This should get rid of the stumble at 1500-2000 RPM. I would also increase the main jet on both the primary and the secondary and raise it 5 points from what it is now. So if they are 125, I would change them to 135. The LAST thing that you want is to have the engine run LEAN at high RPM. It will run great for a short period of time, before the pistons melt. The carburetor also has an accelerator pump that squirts a pre-determined amount of fuel with the opening of the secondary venturi. Much like the cross-over ports help the primary venturi with a small additional amount of fuel, the accelerator pump does the same for the secondary venturi. If this were not so, then there would be an instant when the secondary venturi would run lean (on first opening) and this would cause a hesitation in the power delivery. It is important to get this part of the carburetor correct, as this is what gives the engine that "crisp" response so important when accelerating out of a corner.

Just in case you friend does not have the right jets, get on the Web and go to www.mcmaster.com or some equivalent Canadian company and order an assortment of miniature drill bits and a "Pin Vise". I would order drill bits from 1 to 2.5 millimeters in .05mm steps, or some 30 drill bits, and a .45, .50, and a .55mm drill bit as well. You are now equipped to deal with any jet in any carburetor (except those that use needles, but that is another story). They should be between $1 and $2 dollars each. Make a little holder from a block of wood and label each of the drill bit by its size. A 125 jet has a hole 1.25mm, and a 130 jet has a hole 1.30mm. As you can see, as a last resort you can make your own jets. If you need to go smaller, a soldering iron and some regular soft solder will close the hole, and then you can re-drill it for whatever size you need. The drill bits are also your "gauge" as you can with them measure any jet to make sure what size it is.

As far as high speed running is concerned a good rule of thumb is to start with 125 Mains and 175 Air Correction jets (a numerical spread of 50) if you have no idea where to start. The fuel and air as joined together in the "emulsion tube". In the cavity in the emulsion tube the air/fuel is mixed (emulsified) and aerated before it is fed to the opening in the secondary venturi. The air stream passing through the secondary venturi literally carries the fuel with it into the combustion chamber. I like F11 emulsion tubes for DCOE carburetors, but regardless of the type of carburetor, the emulsion tube is the LAST thing that you fine tune, if necessary. The engine should at least run on this combination. Next I recommend that you use a Air/Fuel meter to monitor high speed fuel situation. The key number is 13:1. Yes, a stochiometric mixture would be 14.7:, and this is fine if you are tuning to pass a smog test, or for ultimate fuel economy, but it is much too lean for a competition motor. If course if it reads 10:1, then you are much too rich. Be careful, you need to test the entire RPM range, as it could be that slow running is OK but wide open throttle is too lean, or vise-versa. If you do not have an A/F meter (also known as a Lambda meter), then you have to use a bit of intuition.

If the engine misses at high RPM, then it is entirely likely that the main jet is too small, particularly if the exhaust pipe is very light in color (white to very light grey). The engine is leaning out at high RPM and if this situation persists, then you will do major damage. You can have the same miss if the engine is grossly too rich, except that the tailpipe will now be black and it will be spewing large amounts of black smoke (unburnt fuel) and it may foul the plugs. If you are standing behind such a car, it will not be long before your eyes begin to water.

This brings us to how to adjust main jets. Adjustment steps in the main fuel jet are generally done in .05 sizes (from a 125 to a 130). Of course the larger the number the more fuel it will pass. Main jet changes affect the carburetor fuel delivery throughout the entire RPM range, once the carburetor is past the cross-over circuit (from about 2200 to ????? depending on how brave you are).

Air Correction jet use a different formula. In short, 4 steps of air correction change is roughly equivalent to a single incremental step in fuel jet. Change the air correction from 175 to 195, and that would be the equivalent to changing the fuel from 130 to 125. Confused? In essence you are letting more air into the emulsion tube in relation to fuel, so the mixture is "leaner". Here comes the interesting part. Whereas a fuel jet change affects the entire RPM range equally, changes in air correction jet have greater effect at RPMs above 5000 RPM. So if the engine is running fine everywhere, but a little lean at top RPM, then you could fine tune it by going 5-10 points smaller on the air correction jet.

Tuning Webers is as much an art as it is a science, as every now and then you will have an engine that needs something totally different, but this is VERY rare.

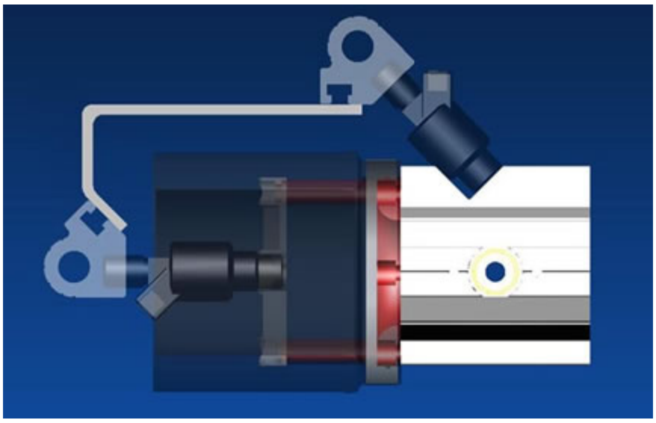
7.4.3.Fuel Injection

As we are talking about competition purposes, I will only discuss those variations of fuel injections that are directly applicable to competition vehicles.

There are basically two type, mechanical and electronic. From a design perspective they both employ a “throttle body” per cylinder. On the mechanical side the only one that was actually implemented on an Abarth vehicle was the Kugelfischer type. This had a “slide” type throttle, instead of butterfly type, and a complex belt driven mechanical, high pressure pump.

The latest ECU driven, electronic fuel injection systems therefore have a real advantage over the earlier mechanical system. If you look at the early Kugelfisher systems, these were really only totally efficient at near, or full throttle. It is for this reason that the injectors were placed well upstream. This aided high RPM performance, but did little for lower RPMs and made for a very narrow usable power band.

It would be possible to use an ECU that could be programmed to fire two banks of injectors, either singly or together. This would allow me to place one set of injectors just past the Injector Throttle Bodies (ITB) or as close as practical to the valve, and the other set above the inlet trumpets. (See diagram). I found a company called Extrabody that was working on something similar and we are now collaborating on a solution that could be produced as a kit for the PBS 8P cylinder head.



The Extrudabody ITB units can be mounted on a manifold designed for a Weber DCOE carburetor. This makes the system simple to mount to the 9P head. Because the system is modular, additional extrusions can be added, either before or after the ITB portion to custom tune the intake runner length to meet almost any design criteria. As the second drawing illustrates, the system can also cater for the two injector idea that I described earlier.

Using one of several aftermarket ECUs, a map can be derived that automatically sequences between the two injector banks, either using both injectors at the same time, or switching from one bank to another.

Now, I understand that this type of system is not allowed in many racing clubs, but where it is, it hold some reasonable promise, particularly when you add to this the ability of the ECU to control spark advance as well.

8.1 Ignition Systems

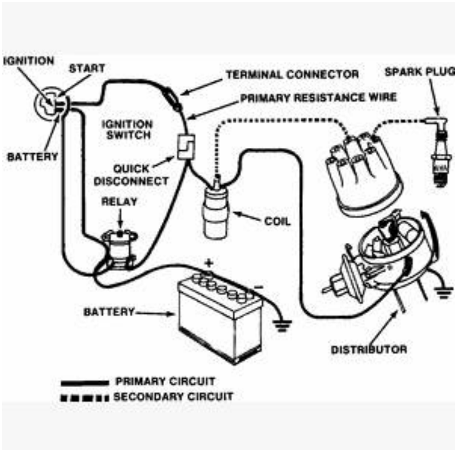
The ignition system sends an extremely high voltage to the spark plug in each cylinder when the piston is at the top of its compression stroke. The tip of each spark plug contains a gap that the voltage must jump across in order to reach ground. That is where the spark occurs.

The voltage that is available to the spark plug is somewhere between 20,000 volts and 50,000 volts or better. The job of the ignition system is to produce that high voltage from a 12 volt source and get it to each cylinder in a specific order, at exactly the right time.

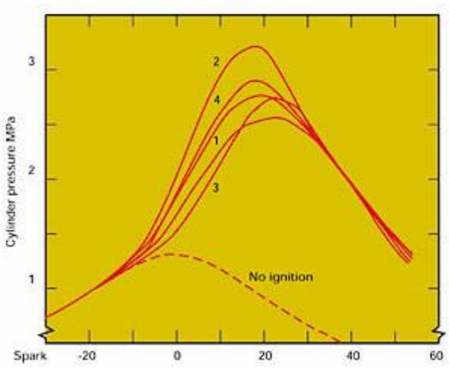
Let's see how this is done.

The ignition system has two tasks to perform. First, it must create a voltage high enough (20,000+) to arc across the gap of a spark plug, thus creating a spark strong enough to ignite the air/fuel mixture for combustion. Second, it must control the timing of that the spark so it occurs at the exact right time and send it to the correct cylinder.

The ignition system is divided into two sections, the primary circuit and the secondary circuit. The low voltage **primary circuit** operates at battery voltage (12 to 14.5 volts) and is responsible for generating the signal to fire the spark plug at the exact right time and sending that signal to the **ignition coil**. The ignition coil is the component that converts the 12 volt signal into the high 20,000+ volt charge. Once the voltage is stepped up, it goes to the **secondary circuit** which then directs the charge to the correct spark plug at the right time. Here is a diagram of such a system.



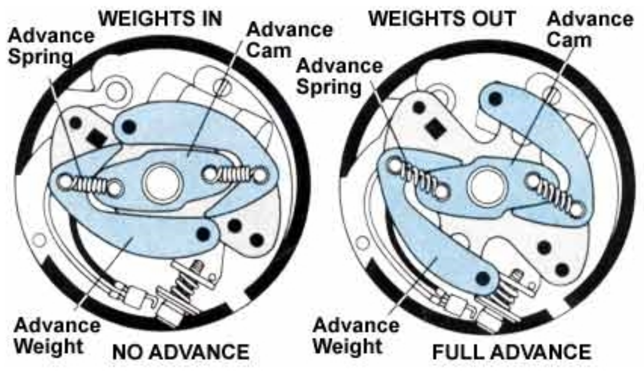
Without exception, almost all ignition systems were “mechanical” systems, up to the early 70s. Engines, to operate efficiently, require the spark to fire at some point BEFORE the piston reaches TDC.



This is to allow the explosion to build enough pressure (push) on the top of the piston, at just the right time, to provide optimum power. If it is started too soon (advanced) then this explosion reaches piston while it's still traveling upward and you lose power, (trying to push the piston the wrong way) waste energy, and create heat in the combustion chamber area (and usually knocking or detonation from an explosion instead of a nice smooth flame traveling from the upper cylinder to the piston top). If started too late (retarded) then you loose power because the piston is already traveling downward, before the flame explosion can "push" it. This also creates heat in the surrounding combustion chamber because remember, heat is energy. This energy, if not used to push the piston, is released either into the surrounding water jacket or the exhaust manifold instead of powering your vehicle. Both are inefficient as far as maximum power is concerned, but it makes an effective heater! As the engine RPM's increase, given that the flame propagation speed remains the SAME, then the combustion cycle needs to be started earlier to achieve the desired "push" on the top of the piston. Also, as the pressure (more fuel/air) inside the cylinder increases, then the less advance the engine can handle at a lower RPM (bigger explosion). So as you can see it depends upon the speed (RPM) of the engine, AND the amount of air/fuel mixture (throttle position) that the engine is operating at. OK, elementary internal combustion education is out of the way.

These systems all used an "advance" mechanism to alter the timing of the spark pulse to the cylinder. A good starting point for setting the initial timing (static timing) is 10 BTDC. You might call this "idle advance" at 1200 RPM or below. As the engine RPM increase, the amount of time available for ignition decreases, so the initiation of spark must occur earlier. There is a limit to the amount of advance, but for standard engines this would be around 28 degrees.

So how do we get from the 10 degrees at idle to the 28 degrees at higher RPM?

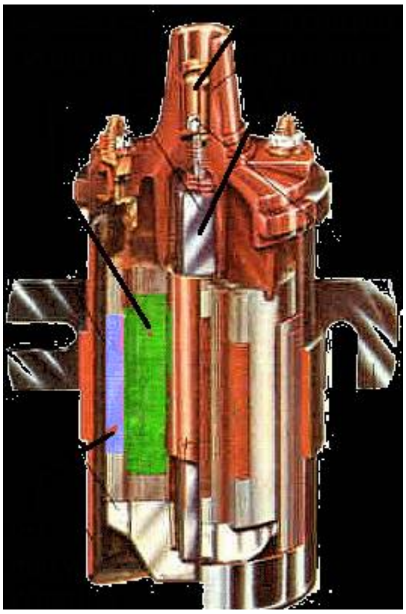


Above is a good illustration of the standard mechanical advance mechanism. The point plate in the distributor has two advance weights on separate pivot pins. The advance weights are connected to the advance cam via small coiled springs. These springs provided tension to keep the cams against the cam. As the distributor spins, the centrifugal forces generated cause the weights to want to move away from the centerline of the distributor, however the springs provide a resistance to this centrifugal effect. As the weights swing outward the points plate is rotated so that the "points opening event" occurs earlier. By varying the cam design, and the tension of the advance springs, it is possible to change the advance characteristics of the distributor. In most competition engines, using a mechanical advance mechanism, the distributor is generally fully advanced by 3000 RPM.

Now say your cruising at 2000 RPM little load, again low cylinder pressure, optimum advance (30 deg) engines happy. Suddenly you snap open the throttle. Now you have maximum cylinder pressure, low engine speed and advance needs to be at say 12 deg to prevent detonation. If the advance were purely mechanical again, and set for optimum advance (30) at the no/low load condition, then we would have too much advance for this high load condition, and one unhappy engine because of detonation. However, during high load conditions, the intake manifold pressure drops to zero (equals outside manifold pressure or no vacuum). IF the mechanical timing were now optimized for high load, low speed conditions (12 deg@2000 RPM), then the vacuum unit can optimize the timing at light or no load conditions (30 deg) because it is in effect not operating at high load conditions, and the mechanical advance can be optimized for high cylinder pressure or maximum load conditions.

So in this case, when you stomp on the pedal, the timing (at 30 deg light load, relatively high vacuum) would drop back to 12 deg, because the vacuum is now not operating, as stated before, the manifold pressure increased (vacuum dropped to zero) and the diaphragm returned to it's no vacuum position. In this way, timing can be optimized for all engine conditions. For racing, and max power applications, you don't really need a system for controlling advance at low or no load conditions because these engine are operating at maximum power most if not all the time. (and is one reason why some tend to overheat at idle) Also, another reason that early emission systems with idle retard, or advance cutouts have a provision that during extended idle periods, when the engine begins to overheat, it restores PROPER advance to prevent that overheating! Note: High performance engines generally do not use vacuum advance mechanisms, as the vacuum generated is relatively low due to the longer camshaft duration and overlap.

This would be a good time to speak about ignition coils. To put it simply a coil is nothing more than a "step-up" transformer. It has a primary side (12V DC from the car's power supply) and secondary side. Each side of the transformer has a number of windings, and the number will be sufficient to step up the 12V to between 20,000-50,000 Volts. It is this secondary voltage that causes the spark in the cylinder when it jumps from the center electrode of the spark plug to ground.



8.2 Points Type

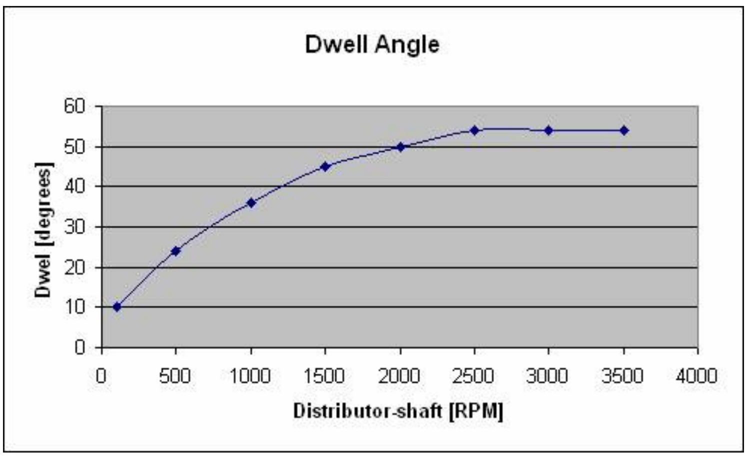
Ignition points are a set of electrical contacts that switch the coil on and off at the proper time. The points are opened and closed by the mechanical action of the distributor shaft lobes pushing on them. The points have a tough job, switching up to eight amps of current many times per second at highway speed. Indeed, as engine speed increases the efficiency of your ignition system decreases, thanks to heating problems and fundamental electrical laws. This declining efficiency has a serious effect on your spark voltage and results in poor high-speed performance, incomplete combustion and other drivability problems.

Condenser: Those same principles of inductance create a kind of paradox, because when the points open and the magnetic field collapses it also induces a current in the primary as well. It's not very much because there are only a few windings in the primary, but it's enough to jump a small air-gap, such as the one between the just-opening points in the distributor. That tiny spark is enough to erode metal away from the points and you'll 'burn' the points. It prevents the points from arcing and prevents coil insulation breakdown by limiting the rate of voltage rise at the points.

Ballast Resistor: This is an electrical resistor that is switched in and out of the supply voltage to the ignition coil. The ballast resistor lowers voltage after the engine is started to reduce wear on ignition components. It also makes the engine much easier to start by effectively doubling the voltage provided to the ignition coil when the engine is being cranked. Not all car manufacturers used a ballast

A points type of distributor uses a set of spring loaded points to "initiate" the spark event. When the points close, current through the coil primary increases from zero to maximum in an exponential manner, rapidly at first, then slowing as the current reaches it's maximum value. At low engine speeds, the points are closed long enough to allow the current to reach a higher current level. At higher speeds, the points open before the current has time to reach this maximum level. In fact, at very high speeds, the current may not reach a level high enough to provide sufficient spark, and the engine will begin to miss. This current through the coil builds a magnetic field around the coil. When the points open, the current through the coil is disrupted, and the field collapses. The collapsing field tries to maintain the current through the coil. Without the Condenser, the voltage will rise to a very high value at the points, and arcing will occur. Another possible problem with points type distributors, particularly at higher RPMs, is that you may get "point bounce". This would be and indication that the points spring is not strong enough to run at higher RPMs.

Setting the Dwell Angle - The Dwell-Angle is the number in degrees of rotation of the distributor-shaft, whereby the breaker-points are closed. *(The same for an electronic ignition module, discussed immediately below), only here the breaker-points are replaced with a control-module and the lobes on the distributor-cam are replaced by a reluctor. The reluctor induces pulses which are past on by the ignition-signal sensor to the ignition-module. The ignition-module "tells" the power-transistor to turn the current through the primary-coil on or off. The time in which the power-transistor turns the current "on", is also expressed in degrees of rotation of the distributor-shaft.)*



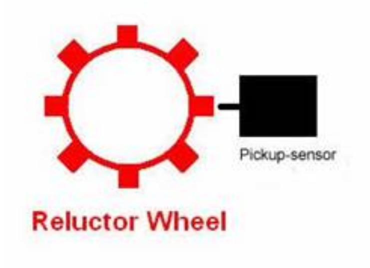
For a mechanical system, the Dwell-angle is during operation a fixed number, about 50 degree for a 4-cylinder engine. This means that the current flows through the primary-coil for 50 degrees of distributor-shaft rotation, regardless the RPM. For example when running idle: 500 rpm, the crankshaft makes 1 revolution in 120ms. The distributor-shaft, at half speed, 240ms. It takes 4ms to charge the ignition-coil till saturation. The required Dwell-angle is (360° / 240ms) x 4ms = 6°. In reality the ignition-coil is charged for 50° duration, 44° more than required. The 50° Dwell-angle is required if the crankshaft makes 1 revolution in (360°x 4ms) / (50x2) = 14.4 ms. This is 4,166 rpm, nearly full throttle!!! So above the 4,166 rpm the ignition coil is charged below saturation, and the spark intensity will therefore be less. Certainly at 9000 RPM we may be pushing the limit of the ability of the coil to reach sufficient saturation to support combustion for extended periods of time.

8.3 Electronic

Electronic Ignition systems are not as complicated as they may first appear. In fact, they differ only slightly from conventional point ignition systems. Like conventional ignition systems, electronic systems have two circuits: a primary circuit and a secondary circuit. The entire secondary circuit is the same as in a conventional ignition system. In addition, the section of the primary circuit from the battery to the battery terminal at the coil is the same as in a conventional ignition system.

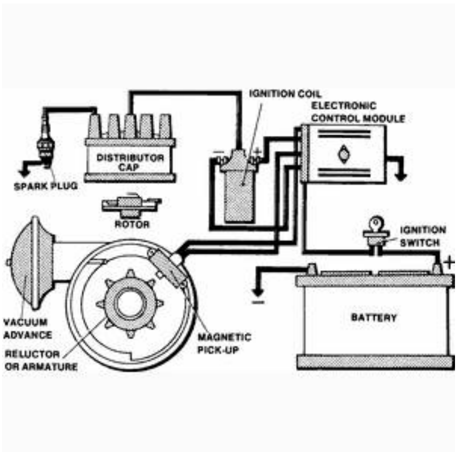
The primary circuit of the electronic ignition systems operate on full battery voltage which helps to develop a stronger spark. Whereas, Breaker point systems needed a resistor to reduce the operating voltage of the primary circuit in order to prolong the life of the points.

Electronic ignition systems differ from conventional ignition systems in the distributor component area. Instead of a distributor cam, breaker plate, points, and condenser, an electronic ignition system has an armature (called by various names such as a trigger wheel, reluctor, etc.), a pickup coil (stator, sensor, etc.), and an electronic control module.



Essentially, all electronic ignition systems operate in the following manner: With the ignition switch turned on, primary (battery) current flows from the battery through the ignition switch to the coil primary windings. Primary current is turned on and off by the action of the armature as it revolves past the pickup coil or sensor. As each tooth of the armature nears the pickup coil, it creates a voltage that signals the electronic module to turn off the coil primary current. A timing circuit in the module will turn the current on again after the coil field has collapsed.

So in the case of the Magneti Marelli “Marelliplex” system. The distributor still has a mechanical advance system, on which is mounted the magnetic pickup. Because the signal from this pickup is extremely low voltage, it has to be amplified, so that it is sufficiently strong enough to trigger the coil. In addition the amplifier module, located on the large aluminum heat sink (to which the coil is mounted) has the timing circuitry to turn off the coil primary current. The module used by Magneti Marelli is actually a standard unit made by Delco, and the coil is an ordinary coil without a voltage drop resistor.



It would therefore be perfectly feasible to build your own electronic ignition system from readily available parts. Here is a short shopping list.

A) Standard Fiat 850/903/A112 distributor

B) Pertronics (or equivalent) sensor module and reluctor

3) Delco or Chrysler Ignition amplifier

4) MSD Coil (without ballasts resistor)

8.4 Capacitive Discharge & Transistorized Ignition

An advantage of the capacitive discharge ignition system is that the energy storage and the voltage ‘step up' functions are accomplished by separate circuit elements allowing each one to be optimized for its job.

Capacitive discharge ignition systems work by storing energy in an external capacitor, which is then discharged into the ignition coil primary winding when required. This rate of discharge is much higher than that found in inductive systems, and causes a corresponding increase in the rate of voltage rise in the secondary coil winding. This faster voltage rise in the secondary winding creates a spark that can allow combustion in an engine that has excess oil or an over rich fuel air mixture in the combustion chamber. The high initial spark voltage avoids leakage across the spark plug insulator and electrodes caused by fouling, but leaves much less energy available for a sufficiently long spark duration; this may not be sufficient for complete combustion in a “lean burn” turbocharged engine resulting in misfiring and high exhaust emissions.

The high voltage power supply required for a capacitive discharge system can be a disadvantage, as this supply provides the power for all ignition firings and is liable to failure.

Ignition in lean fuel mixtures by capacitive discharge systems can sometimes only be accomplished by the use of multi-spark ignition, where the ignition system duplicates the prolonged spark of inductive spark systems by sparking a number of times during the cycle. The MSD unit is a good example of this. At engine RPMs below 3000 the MSD provided multiple spark events. Above this RPM it reverts to a single spark event.

Below is a link to a very good discussion of the aspects of electronic CDI ignition.

http://www.mclarenelectronics.com/Products/All/App_Act_Ign.asp

8.5 Crankshaft Triggered or Distributorless Ignition Systems (DIS)

The third type of ignition system is the distributorless ignition. The spark plugs are fired directly from the coils. The spark timing is controlled by an Ignition Control Unit (ICU) and the Engine Control Unit (ECU). The distributorless ignition system may have one coil per cylinder, or one coil for each pair of cylinders.

Some popular systems use one ignition coil per two cylinders. This type of system is often known as the waste spark distribution method. In this system, each cylinder is paired with the cylinder opposite it in the firing order (usually 1-4, 2-3 on 4-cylinder engines). The ends of each coil secondary leads are attached to spark plugs for the paired opposites. These two plugs are on companion cylinders, cylinders that are at Top Dead Center (TDC) at the same time. But, they are paired opposites, because they are always at opposing ends of the 4 stroke engine cycle. When one is at TDC of the compression stroke, the other is at TDC of the exhaust stroke. The one that is on compression is said to be the event cylinder and one on the exhaust stroke, the waste cylinder. When the coil discharges, both plugs fire at the same time to complete the series circuit.

Since the polarity of the primary and the secondary windings are fixed, one plug always fires in a forward direction and the other in reverse. This is different than a conventional system firing all plugs the same direction each time. Because of the demand for additional energy; the coil design, saturation time and primary current flow are also different. This redesign of the system allows higher energy to be available from the distributorless coils, greater than 40 kilovolts at all rpm ranges.

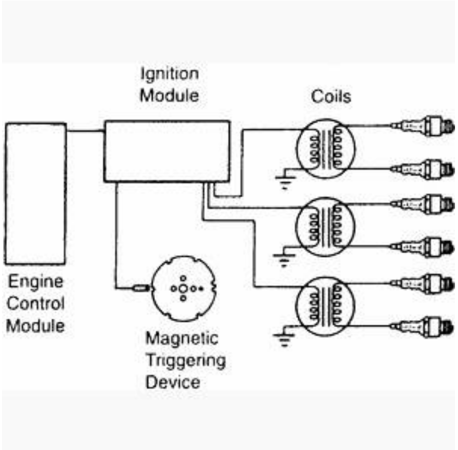
The Direct Ignition System (DIS) uses either a magnetic crankshaft sensor, camshaft position sensor, or both, to determine crankshaft position and engine speed. This signal is sent to the ignition control module or engine control module which then energizes the appropriate coil.

The advantages of no distributor, in theory, is:

- No timing adjustments
- No distributor cap and rotor
- No moving parts to wear out
- No distributor to accumulate moisture and cause starting problems
- No distributor to drive thus providing less engine drag

The major components of a distributorless ignition are:

- ECU or Engine Control
- Unit ICU or Ignition Control
- Unit Magnetic Triggering Device such as the Crankshaft Position Sensor and the Camshaft Position Sensor
- Coil Packs



8.6 Ignition Timing and Combustion

Under ideal conditions the common internal combustion engine burns the fuel/air mixture in the cylinder in an orderly and controlled fashion. The combustion is started by the spark plug some 5 to 40 crankshaft degrees prior to top dead center (TDC), depending on engine speed and load. This ignition advance allows time for the combustion process to develop peak pressure at the ideal time for maximum recovery of work from the expanding gases.

The spark across the spark plug's electrodes forms a small kernel of flame approximately the size of the spark plug gap. As it grows in size its heat output increases allowing it to grow at an accelerating rate, expanding rapidly through the combustion chamber. This growth is due to the travel of the flame front through the combustible fuel air mix itself and due to turbulence rapidly stretching the burning zone into a complex of fingers of burning gas that have a much greater surface area than a simple spherical ball of flame would have. In normal combustion, this flame front moves throughout the fuel/air mixture at a rate characteristic for the fuel/air mixture. Pressure rises smoothly to a peak, as nearly all the available fuel is consumed, then pressure falls as the piston descends. Maximum cylinder pressure is achieved a few crankshaft degrees after the piston passes TDC, so that the increasing pressure can give the piston a hard push when its speed and mechanical advantage on the crank shaft gives the best recovery of force from the expanding gases.

Detonation: A violent explosion; also called combustion knock. This usually occurs near the end of the combustion process when highly compressed, high-temperature end gases spontaneously ignite, radically increasing the cylinder pressure. This pressure spike moves at the speed of sound in the combustion chamber, and the pressure can cause damage to pistons, cylinder walls, and the head gasket.

Pre-ignition: The onset of combustion before the spark plug fires. This is generally caused by some type of glowing ignition source such as a hot exhaust valve, too-hot spark plug, or carbon residue. Pre-ignition is especially damaging to engine components like pistons and head gaskets, since excessive cylinder pressures can occur even before the piston reaches top dead center (TDC).

These are the classic definitions of detonation and pre-ignition. Perhaps a more fun definition of detonation would be to imagine the piston screaming up to TDC while you whack that piston as hard as you can with a 10-pound sledgehammer. The clang that you would hear is the same noise that occurs when your engine goes into detonation. Even if detonation doesn’t break any parts, as soon as an engine experiences detonation, the power drops way off. If you ever have a situation where at a certain point, when you give it more gas, the car physically slows, then STOP.

If you get the idea that detonation and pre-ignition are bad, that’s good. Of all the things that can kill an engine, detonation should be right at the top of your Public Enemy Number One list. The quickest and easiest way to cure detonation is to use a high-quality, higher-octane gasoline

Perhaps the easiest and least- expensive way to reduce an engine's sensitivity to detonation is to cool the engine-inlet air. Not only is cooler air more dense, which makes more power, but cooler air is also less prone to detonate. The classic performance rule-of-thumb is that for every 10 degrees you reduce the inlet air temperature, the engine makes 1 percent more power. This is why drag racers use ice to cool the intake manifold and why all those cold-air inlet systems work on late-model cars. Forcing your engine to breathe hot underhood air will also make it more prone to detonate, so make sure your carburetors have ready access to air that is at least at ambient temperature. In addition, keep your fuel as cool as possible as well.

Ignition timing is another cheap and easy area to work on. If your engine detonates at low engine speeds at part throttle, consider retarding the initial timing by 2 or 3 degrees and then adding that amount back into the total by increasing the mechanical-advance curve. For example, let's say you have 10 degrees initial timing with a total of 30 degrees and your engine rattles a little at part throttle, especially right off idle. You could cut the initial back to 8 degrees and add 2 degrees to the mechanical advance. The total remains at 30, but now the engine doesn't death rattle every time you let the clutch out..

Camshaft timing also plays a huge role in dynamic cylinder pressure, especially with street-driven performance engines. As you increase intake duration, this means the intake valve now closes later than it does with a shorter-duration cam. This later-closing intake valve bleeds some cylinder pressure back into the intake manifold at lower engine speeds. The longer the duration of the cam, the later the intake closes. This reduces cylinder pressure at lower engine speeds, which reduces the tendency for the engine to detonate.

Late closing of the intake can also be accomplished by retarding the camshaft's installed point. For example, many small-block Chevy cams are installed with the intake centerline at 106 degrees after top dead center (ATDC). This tends to close the intake valve sooner, which improves low-speed torque by increasing cylinder pressure at low speeds. But if the engine rattles at low speeds, retarding the closing point of the intake valve can by 3 or 4 degrees (from 106 to 110 degrees ATDC) softens the engine's need for higher-octane fuel.

Obviously, this is a little more difficult to do than playing with ignition timing but may pay off by allowing you to run a lower-octane fuel. If you do retard the cam, it's important to go back and perhaps add a degree or two of initial ignition timing.

You can also experiment with camshaft overlap. Unfortunately, this requires a new camshaft. Tightening the lobe separation angle, from 114 degrees to 110 degrees, for example, increases the amount of overlap since the exhaust valve closes slightly later and the intake valve opens a little sooner. This tends to bleed off cylinder pressure at lower engine speeds, which could be beneficial since this is a little like built-in exhaust gas recirculation (EGR) in the intake manifold.

There are several other ideas that you can try to reduce your engine's sensitivity to detonation and allow it to live on lower-octane fuel. Any kind of oil contamination in the combustion chamber is bad news. Oil is a great breeding ground for creating detonation. The best way to avoid this is to ensure your combustion space enjoys the benefits of tight valve-to-guide clearances and good leak-free valve guide seals. Of course, you want to seal up that intake so it doesn't suck oil into the cylinders, and your short-block should be in good shape.

8.7 Spark Plugs

Spark plugs provide one of the elements, without which an engine simply will not run. Well, at least a non -diesel engine any way. Most of us that dabble in cars would recognize that the primary purpose of spark plugs is to ignite the air/fuel mixture that enters the cylinder.

Spark plugs provide you with a window into the combustion chambers and also provide you with an evidence trail of what is going on in there. Spark plugs and their condition are one of the most important diagnostic tools. They tell you what happened in the cylinder. They will help you in tracking down what the root cause is for many problems and to maximize the air/fuel ratio

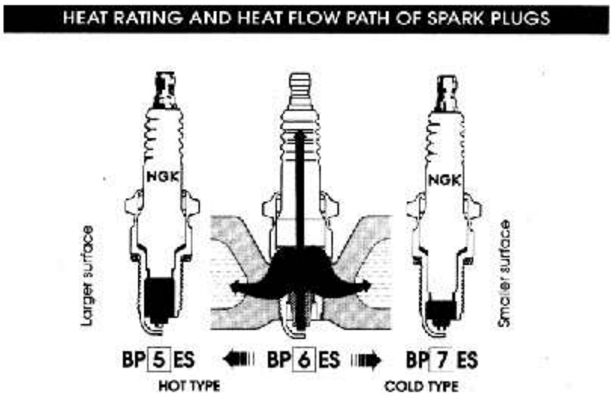
The two things that spark plugs do are:

1) Ignite the air/fuel mixture
2) REMOVE heat out of the combustion chamber.

If a sufficient amount of voltage is applied to the spark plug, so that the resultant spark spans the gap between the electrode and ground, then it is said to have sufficient *electrical performance*. In addition, the temperature of the spark plug's working end must be kept low enough to prevent pre-ignition, yet not too low so as to permit fouling. This is often referred to as the thermal performance of the hear range selected.

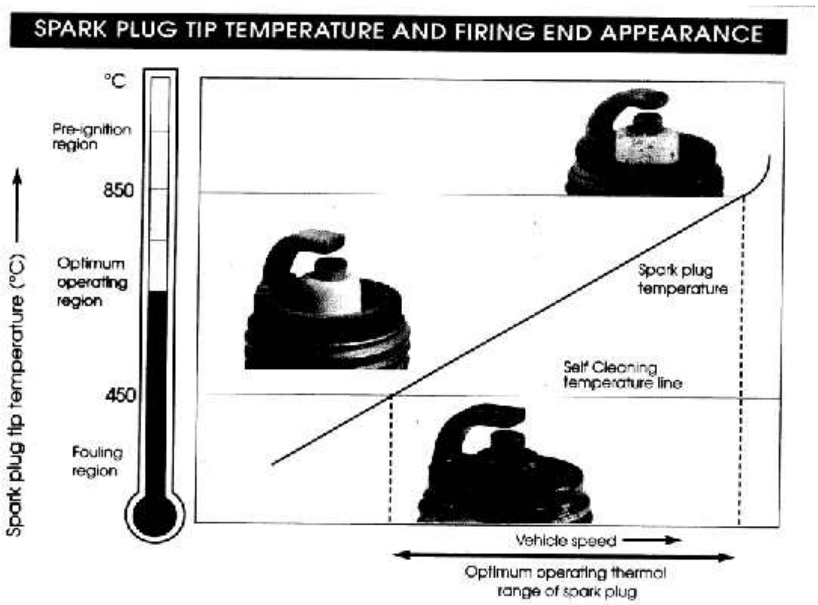
One popular misconception is that spark plugs CREATE HEAT. This is absolutely incorrect, and in point of fact spark plugs can only REMOVE HEAT. The spark plug is like a radiator, taking heat out of the combustion chamber and transferring it to the engine's cooling system. Therefore the heat range of a particular plug indicates the plug's ability to to shed heat. The rate at which a plug shed's heat to the cooling system is determined by:

1) The insulator nose length
2) The volume of gas round the insulator nose
3) Materials/construction of the center electrode and porcelain insulator.



The hear range of a particular spark plug has no relationship to the actual voltage transferred through the spark plug. As stated earlier, the heat range is simply an indicator of the spark plug's ability to remove heat from the combustion chamber. This heat transfer effectiveness is a function of ceramic insulator nose length and material composition of the insulator and center electrode.

The insulator nose length is the distance from the firing tip to the point where the insulator meets the metal shell of the spark plug. The insulator tip is the hottest part of the spark plug and therefore plays a crucial and primary role in both pre-ignition and fouling. No matter what the application, from lawn mower to race car, the tip temperature must remain between 450-850 degrees Centigrade. If the temperature is below 450C, the plug will not burn off carbon and combustion chamber deposits, including lead deposits if high performance fuels are used. This will lead to a misfire. On the other hand if the combustion temperatures are over 850C then the ceramic tip will overheat and fracture and the cause the electrode to melt. Once pre-ignition sets in, major damage can be done to an engine in a very short period of time. I know this to be the case, as I have personally experienced both ends of this scale.

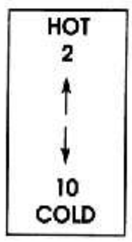


Presuming you use the spark plugs with the same electrical/mechanical characteristics, moving one heat range colder will allow the plug to remove between 70-100 degrees from the combustion chamber.

Projected tip plugs run about 10-20 C degrees hotter than standard plugs. By the same token retracted tip plugs MAY runs somewhat cooler, but much will depend on their construction.

The optimum is to find a plug heart range that will work in the crossover range between fouling and optimum operating range. This is sometimes referred to as the "self-cleaning" range and is somewhere around 600 degrees Centigrade. Here there is little chance of pre-ignition or detonation, yet there is little buildup of carbon and combustion chamber deposits.

Let's get back to the length of the insulator, as this sometimes causes much confusion. The longer the distance between the tip and the spot where the insulator meets the spark plug body, the longer the heat of the combustion chamber has to travel before it can be dissipated to the cooling system. Hence this would be a "HOT" plug. The shorter this path, therefore the colder the plug and the colder plug will remove heat more quickly and reduce the chance of pre-ignition/detonation. A good rule of thumb is to use the coldest plug that is available that does not foul. THen you can work with carburetor jetting to get the air/fuel mixture to where it produces maximum results.



WARNING: Each plug manufacturer has their own way of denoting heat ranges. By example NGK uses a low number (2) to denote a HOT plug, whereas a number 10 would be a very COLD plug. Quite the opposite, Champion uses low numbers to denote a cold plug and high numbers for hot plugs. Make up your own cross reference chart so as not to get confused.

Credits: Some of the information on this page was derived from NGK literature.

Recommended Torque Specifications

SPARK PLUG THREAD SIZE	CAST IRON HEADS WITH TORQUE WRENCH	CAST IRON HEADS WITHOUT TORQUE WRENCH	ALUMINUM HEADS WITH TORQUE WRENCH	ALUMINUM HEADS WITHOUT TORQUE WRENCH
FLAT SEAT w / GASKET				
18mm	3.5 kg-m ~ 4.5 kg-m 25.3 lb-ft ~ 32.5 lb-ft	1/2–2/3 turn 180°–240°	3.5 kg-m ~ 4.0 kg-m 25.3 lb-ft ~ 28.9 lb-ft	1/2–2/3 turn 180°–240°
14mm	2.5 kg-m ~ 3.5 kg-m 18.0 lb-ft ~ 25.3 lb-ft	1/2–2/3 turn 180°–240°	2.5 kg-m ~ 3.0 kg-m 18.0 lb-ft ~ 21.6 lb-ft	1/2–2/3 turn 180°–240°
12mm	1.5 kg-m ~ 2.5 kg-m 10.8 lb-ft ~ 18.0 lb-ft	1/2–2/3 turn 180°–240°	1.5 kg-m ~ 2.0 kg-m 10.8 lb-ft 14.5 lb-ft	1/2–2/3 turn 180°–240°
10mm	1.0 kg-m ~ 1.5 kg-m 7.2 lb-ft ~ 10.8 lb-ft	1/2–2/3 turn 180°–240°	1.0 kg-m ~ 1.2 kg-m 7.2 lb-ft ~ 8.7 lb-ft	1/2–2/3 turn 180°–240°
TAPERED SEAT				
18mm	2.0 kg-m ~ 3.0 kg-m 14.5 lb-ft ~ 21.6 lb-ft	1/12–1/8 turn 30°–45°	2.0 kg-m ~ 3.0 kg-m 14.5 lb-ft ~ 21.6 lb-ft	1/12–1/8 turn 30°–45°
14mm	1.5 kg-m ~ 2.5 kg-m 10.8 lb-ft ~ 18.0 lb-ft	1/12–1/8 turn 30°–45°	1.0 kg-m ~ 2.0 kg-m 7.2 lb-ft ~ 14.4 lb-ft	1/12–1/8 turn 30°–45°

All engine manufacturers have recommended torque specifications for spark plug installation, and most are represented in the chart above. Install the spark plug finger tight until the gasket or taper seat contacts the cylinder head. Then give it the recommended turn or angle.

Some content attributed to the following authors.

Vincent T. Ciulla

1.1 - Using the correct components.

As with any project, one of the most important steps is properly identifying the correct components that need to be incorporated. This process starts by carefully outlining the goals to be achieved. What the intended use of the engine is will guide you as to what may need to be done.

If a motor is for a street car, with a moderate increase in power, then you may choose to use some standard components. If you are building a motor for a "historically" correct restoration of a car, such as an Abarth 750, then you will have a different focus. On the other hand, if your intent is to build a motor for competition use then you will want to use a great many specialty parts, to insure both good power and reliability. This means buying the best possible parts available for the purpose that your budget will allow. As Scuderia Topolino deals principally in competition parts and services, most of what I discuss will have to do with developing competition engines.

It goes without saying that, if you are building an "all-out" competition motor, you have already spent some time developing the rest of the car. This means chassis preparation, gear ratio analysis, and brake development. All too many people start a race car project by building the motor first. I do not recommend this. My suggestion would be to build the car first and get it to handle correctly. This does not require a high horsepower motor and perhaps a standard motor could be used to do this development. This does not require a great deal of horsepower until the car is fairly well developed. Only then do you need a powerful motor to test the ultimate limits of vehicle development.

Once you have identified the components that you need, they should be assembled and laid out to make sure that everything that you need is there. Then each component must be double checked to make sure that it meets the dimensional and quality standards you have set. Do not simply assume that the part is "right", just because you bought it from a company that has a good reputation. Everyone makes mistakes, so make sure that your parts are made to specification before using them. Making the proper measurements means having your own measuring tools or at least having access to someone who can make the measurements for you.

1.2 - Preparation

Once you have assembled all of the parts and determined what you project is going to be, make a list of all of the operations that have to be performed in preparing all of the components that have to be assembled. For purposes of illustration, I will go through the steps involved in building an Autobianchi 1050 motor, assuming that everything has to be checked, measured, and renewed. Many of the steps will applicable to other engine combinations, including Fiat and Abarth 750, 817, 843, 847, 903, 965, and 982cc motors. For specialist Abarth "Bialbero" motors, those with double overhead camshaft installations, there are some other special considerations.

The list of operations below is simply an outline, and not particularly in any order. Some items marked with an "*", may be done during assembly. As I work through each particular area of the motor, in following chapters, I will expand on each of the subjects in the outline. It goes without saying, that you are going to start out with a block that does not have any obvious internal, or external, damage. The fact that it is well used, or rusty and dirty, is really inconsequential.

A. Hot tank block and head. Not a chemical hot tank, as you do not wish to destroy installed cam bearings.
B. Crack-test and pressure-check the engine block that you wish to use.
C. Check block "squareness".
D. Check block for main bearing and cam bore alignment and bore diameter size.*
E. Check crankshaft for cracks and bearing size.
F. Regrind main or rod bearing surfaces if required.
G. Clean all crankshaft passages.
H. Check camshaft for straightness.
I. Modify front cam bearing for oil pump drive oiling.
J. Check lifters for proper lifter surface curvature.
K. Check lifter bores for damage and clearance.
L. Check push rods for straightness.
M. Check connecting rods for straightness.
N. Check connecting rods for bore and pin size*
O. Check pistons for proper bore and pin dimensions.
P. Check piston/rod combination for proper deck height.*

Q. Check rockers and rocker shaft for proper alignment and clearance.
R. Check valve guide/valve stem clearance.
S. Install new valve guides if required.
T. Check valve/seat for proper seal
U. Calibrate valve springs
V. Check valve installed height.
W. Check valve spring for coil bind.
X. Check valve spring seat and "over the nose" pressure.
Y. Check rocker arm geometry.
Z. Check for correct push rod length.
AA. Check center cam bearing alignment and oil feed orifice to cylinder head.
AB. Debur block inside and outside.
AC. Remove any excess block material not required.
AD. Bore/Hone cylinders for new pistons
AE. Align hone main bearing bores.
AF. Debur, port and surface finish all head surfaces.
AG. Surface block and cylinder head.
AH. Install new cam bearings if required, line bore and check clearance.
AI. Check camshaft for clearance with connecting rods.
AJ. Check main and connecting rod bearing shell thickness.
AK. Debur any sharp edges on piston crown, check valve pocket size, depth and radial clearance.
AL. Check piston ring to ring groove clearance.
AM. Check piston ring end gap in bore.
AN. Check piston to valve clearance at 20-30 deg. BTDC and ATDC.
AO. Double check main and rod bearing clearance.
AP. Check crankshaft thrust clearance and replace bearing if required.
AQ. Install pilot shaft bushing/bearing in crankshaft.
AR. Balance crankshaft, flywheel, pressure plate and front pulley/nut.
AS. Balance pistons/rods.
AT. Determine the cubic capacity of each of the cylinder head chambers and equalize.
AU. Determine the cubic capacity of each piston crown or depression and equalize
AV. Check clutch assembly clearances
AW. Make sure that all gaskets, seals and other items are available

1.3 - Motor Assembly and Environment.

After you have all of the components individually prepared, cleaned and inspected, you are ready to begin assembling the motor. If there is anything that is not ready, then you will not be able to finish the job. That is not the end of the world, but it will simply mean that you will have to stop at some point, and then continue later.

The room that you use for engine assembly should be clean. By this I mean that if you have access to a dust free environment, use it. It does not have to be a "cleanroom environment", but this area should absolutely not contain any machinery or abrasive materials. If there is reason to modify any component, then this should be done outside of the assembly room, and the part thoroughly cleaned before it is returned to the assembly room.

Assembling a motor is like a surgeon doing a major operation. There is a certain sequence and order to be maintained.

Make a running record of all specifics about the motor as it is being assembled. This includes numbering and dating the block and head used. This will assist you, later on, when the engine needs to be rebuilt again. As you then inspect the motor it will give you a better understanding of the things you find when you disassemble the motor. Your record keeping should include all clearances maintained and torque, and/or stretch, settings for all fasteners in the motor.

Remember that we are assembling a competition motor, which will be subject to severe stresses. These records will be vital to make sure the components still comply with the original component specification.

> As an example, you will want to make a record of the overall length of each connecting rod bolt, and the location in the motor. As these bolts are designed to stretch up to 0.005-0.006 inch (0.1-0.12mm) when properly installed, you will want to make sure that they are within 0.0005 inch (0.0127mm) of their original length when removed. If they are not, then it is an indication that they have been stressed beyond their elastic limit and should be replaced.

1.4 - Post Assembly Test Procedure

Testing a motor is all about "comparison". Sure it would be nice to say that a motor develops 115 horsepower. The problem is, by what standard? Actually the finite number of horsepower is not nearly so important, as whether improvement is being made. Even "improvement" can have different meanings.

Therefore, what is important is to make sure that the same test procedure is used in all cases. Only then can you be sure that an improvement was made or not.

By preference, here is a list of performance testing options.

A. Engine dynamometer with a temperature/humidity controlled environment.
B. Chassis dynamometer
C. A stop watch.

By far the best test procedure would be to test each engine build on the same engine dynamometer. Here you have control over as many testing parameters as possible. You will be able to asses a multitude of performance parameters, not the least being torqued and horsepower.

The problem is that 95% of amateur racers will not have such a device. However, I am sure that they would have access to a chassis dynamometer. The chassis dynamometer has the additional advantage of being able to test the entire driveline package. So if you are also looking at reducing parasitic losses in the power transmission system, it can be of help there as well.

Finally, find a track where you can test. An accurate stop watch should be able to tell you if you are turning better lap times or not. You may not fully understand the reason for the better lap times.

As you can see each successive testing option introduces more variable that have to be dealt with. On the engine dynamometer "driver" variables have no impact, as compared to track testing, but each will tell you different things.

Next you must develop a repeatable testing procedure. Here are some of the steps that I include in my procedure.

1. Engine Run In
2. Carburetion settings
3. Power Runs

Engine Run In - To begin with each motor should be "run in" for a period of 20-25 minutes, in an RPM range from 2500-4000 RPM under light to moderate load with no more than 26 degrees total ignition advance. I recommend that this run-in procedure be conducted using a non-synthetic, petroleum based oil, with sufficient levels of zinc and phosphorus. This is vitally important to ensure that items such as camshafts and lifters are properly broken in. DO NOT run the engine at idle for extended periods of time. It goes without saying that you would monitor oil pressure, oil temperature and water temperature during this run-in period. See Section 3.2.2 Oils and Additives for more information on oils with the proper levels of zinc and phosphorus.

Even though I have carburetion settings as the next item, you should at least make sure that in the run-in RPM range the engine is running at an air-fuel ratio of around 12:1-13:1. If either too rich (less than 11:1) or too lean (more than 14.7:1) then this must be addressed BEFORE continuing with the run-in procedure. To rich a mixture may cause a decrease in cylinder lubrication and cause piston damage, and too lean may bring on pre-ignition or detonation.

Once this run in procedure is completed then the cylinder head should be retorqued, and the valve clearances re-set to the camshaft manufacturer’s specification.

Carburetion Settings - The most common carburetion will be either a single two barrel downdraft carburetor or, as used with the PBS 8P and TCR heads, a set of dual side draft carburetors. These could be manufactured by Solex, Del’Orto or Weber.

In the range of 2500-4000 RPM the engine will be running on either the "cross over" orifices or the main metering system. For the run-in procedure you need only concern yourself that each cylinder is operating in the correct air-fuel mixture range. In the next section I will go into more detail the carburetion tuning options for maximizing performance. For a complete discussion of carburetion settings, please refer to section 3.4.1 Carburetion and Fuel Injection.

Power Runs - Here is where we find out what the optimum settings are for horsepower (and of course torque). This will be a combination of a multitude of settings, including ignition, fuel, cam timing, valve settings and other variables. What is important is that you develop a methodology that can be repeated so that improvements can be noted.

You must make sure that certain basic parameters are properly adjusted before any power runs are attempted. These are:

> Oil Change - If you wish to change to synthetic oil, then this would be a good time. Again, make sure it is oil with elevated levels of zinc and phosphorus (1200-2000 PPM).
>
> Camshaft timing - Usually set during engine assembly (for chain driven camshafts this is usually 4 degrees advanced).
>
> Ignition Timing - Use a total advance of 28 degrees as a starting point.
>
> Fuel Pressure - Maximum pressure of 3-4 PSI, with adequate flow.
>
> 1. Carburetion Testing - Optimizing carburetion settings for Idle, Cross-Over, Full Throttle and Acceleration.
> 2. Venturi air flow optimization.
> 3. Ignition timing variations.

Power runs may combine changes in any of the above elements to finalize what the best combination of settings may be. It is important to maintain good air-fuel ratio measurements during all of these power runs, as changes in ignition timing will affect carburetion settings and vice-versa.

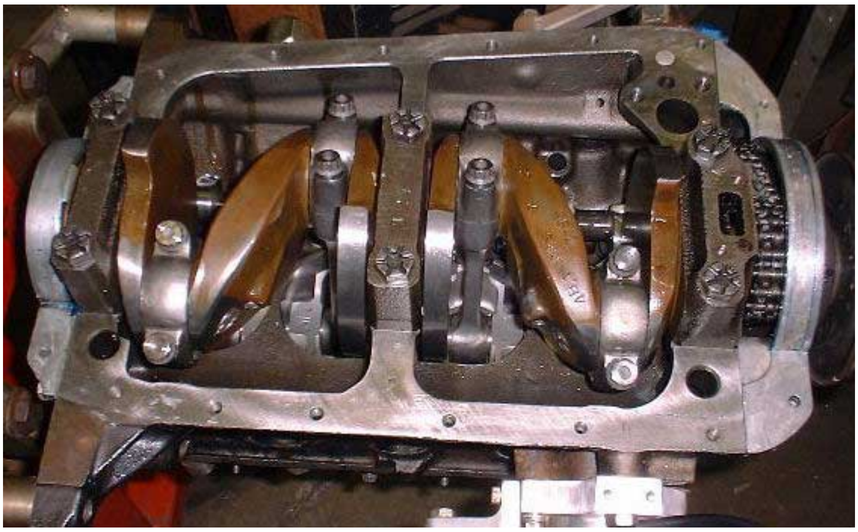
For more detailed information on each of the elements involved in power runs, please consult applicable subject categories.

As before, recordkeeping is a vital element of the testing methodology. Without it you will not be able to cross-reference your findings and get the best benefit of your testing.

2.1 - Cylinder Block Preparation

Caution: Racing is like any other hobby, if you want to be good at it you must spend time at it. The same goes for engine building. You cannot read too much, you cannot listen to much, but eventually you will have to get down to brass tacks and put that motor together. A good friend of mine who builds a lot of small block Chevrolets put it quite well when he said, **"The Cheapest way to Build a Really Good Motor is to do it Right the First Time".** The recommendations that I give below are just that, recommendations. There is no express or implied guarantee of performance.

Now it has to be said, that it would be really neat to just wave a magic wand and end up with that 1000 cc killer motor you always wanted, but that is simply not possible. Even more to the point, while it is possible to build such a motor, unless you are doing all-out racing it may be better to try for something more conservative. Now that I have gotten my reservations out of the way, I am hopeful of working through a complete engine buildup for an all-out racing motor. Along the way I will suggest alternatives for something more conservative as well.

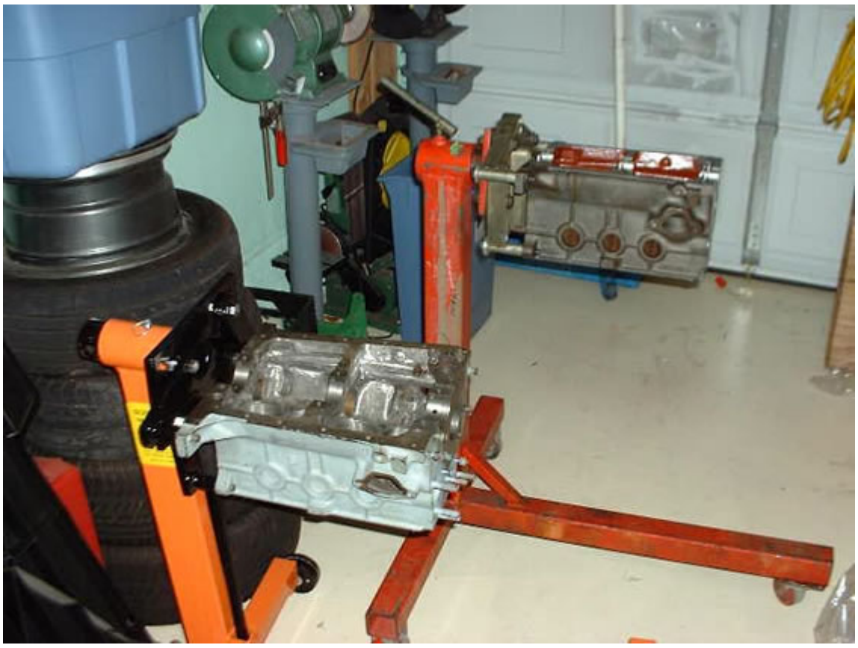


Bottom end of motor that went to Europe in 2001

Note: You should recognize some of the components. This is the bottom end of the motor that is currently in the car in Europe. As you read on you will see that we have changed our mind as to what the preferred way of doing this is. This motor incorporated prepared 850 connecting rods. The current motors have Scuderia Topolino rods, with special ARP2000 connecting rod bolts . We are continuing to use Abarth billet crankshafts, as long as they last and/or can be found.

2.2 - Block Preparation

I cannot stress strongly enough how important this first step is. The largest single item is the block. Unless you want to build an engine that is historically correct for an Abarth 750 GT, or the like, I would stay away from using a 600 block. For 1000cc motors these blocks require a great deal of work. There are two worthwhile alternatives, either an 850/OT1000 block or one of the 903/A112-1050 blocks. (For those wanting a real shortcut for a street motor, just buy a 70 hp A112 1050 motor and put it in your 600 and it will transform your car. This is still not a straight swap, but the work is quite easily done. The clutch end of the cranshaft will have to be drilled for a bronze input shaft bushing. This is not a hand drill operation and will require the removal of the crankshaft from the block. You can also start off this way and then if you want more performance, upgrade for more horsepower later.)



Two new blocks being prepared

The 817/843/OT1000 family of blocks are what I refer to as the "short deck" variety of block, whereas the 903 and 1050 are "long deck" blocks. The difference being that the long deck blocks are about 4-5 millimeters taller. Other than this the blocks a dimensionally similar, with the exception of the main bearing bore. The 817/843/903 use a smaller diameter main bearing bore, as compared to the 982 and 1050 blocks.

Before proceeding any further, clean the block in a parts washer and then have the block thoroughly magnafluxed for cracks. 1050 blocks are particularly prone to cracks in the front surface of the block, between the front main bearing and front cam bearing bores, and all blocks can suffer block cracks around the center main bearing web. Doing this now will save a great deal of wasted work. If the block is at all doubtful, find another one.

Check how far the block has already been bored . I consider the maximum safe cylinder diameter is 66.4mm or so for all but the A112a2000 (A112 70HP) block, which can be bored to 68.00mm. If the block is already to 1mm over standard, consider a different block. There are many blocks out there that are still standard, and these provide a good bases for development.

Note: US machine shops will probably measure blocks in thousands of an inch, so a good formula is that 1mm equals very close to .040 inch. Therefore, a .25mm larger bore is the same as .010 thousands of an inch.

Next determine if the deck of the block has been decked and is flat. If the top of the block has been previously machined, you may find that a standard connecting rod/piston combination will stick out above the block slightly. This is not the end of the world, but just means more work. If it is not flat, but at close to standard height, do not worry about it for now. The standard height for a 817/843/1000OT block is approx. 172mm, whereas a 903/965/A112A1 or A112A200 block will be around 177.8mm in height.

Now determine if the deck is parallel to the crankshaft bearing bores, the cylinders are at 90 degrees to the top of the block, and the front and rear block surfaces are precisely at 90 degree to the crankshaft bore. It is important that all of these dimensions are exact if we are to extract the maximum amount of horsepower out of the engine. If any of these dimensions are out, bring them back into specification. If the ends of the block are not square, and are out by more than .005 thousands of an inch, find another block.

Note: Do not assume that the factory manufactured the block correctly. Remember that this was a production motor and the tolerances for a race prepared motor are much tighter.

Next, knock out all of the freeze plugs (sometimes known as welsh plugs) from the water jackets and the one at the end of the camshaft gallery. These may look like they are OK from the outside, but as they are in contact with the cooling water on the inside, you may find considerable erosion. Put in new ones. You will also find small plugs at the end of the oil galleries and these should also be removed. It is likely that you will want the block hot-dipped, unless you do not plan on replacing the camshaft bearings. These will just disappear in the hot tank process. The oil galley plugs must be removed for cleaning no matter what. Further, remove all bolts, studs etc. so that the machine shop gets a totally naked block, as this means you will get it back with all of the threaded holes thoroughly cleaned. This will save the machine shop time and you money.

Have the machine shop check the block for main bearing saddle alignment and dimension, otherwise you may never get a good bearing fit. The 1000OT and A112 blocks register the main caps with either dowel pins or hollow dowels. I think doweling is a really good idea.

Do not automatically assume that a block needs to be line-honed, but if it appears that the block does, then this is a warning flag that it has had a rough life. Have another good look at this block before you go further. Once this is done, have the machine shop double check the ends of the block for squareness.

The next step is to examine and clean every thread in the block. This means running a tap, or better yet a "thread chaser" down each threaded hole and then blowing it out so that all of the debris is removed. Only then will you be absolutely sure that the fastener will properly torque to the required specification. I have found that many older blocks require some threads to be repaired with helicoils. I no longer take any chances with the head bold threads, if they look at all suspect, they get a helicoil inserted. Inserting helicoils is a precision job. DO NOT attempt to do this with a hand drill, as the threads will not be straight. Later on in this article I will talk about head bolts versus head studs, and you will understand exactly why this is so important.

Two different doweling methods.

Your machine shop should also be able to pressure test the block for you. You will have to make sure that all of the freeze plugs and oil galley plugs have been replaced. If the block passes the water and oil pressure tests, then you can proceed.

Next, the machine shop will need the pistons that you plan to use. Follow the piston manufacturer's instructions for piston fit, but a good rule of thumb is .001-.0015 (0.025-0.038mm) for each inch (25mm) of piston diameter. Thus a 65mm piston would have about .0025-.003 (0.063-0.076mm) of piston to cylinder wall clearance. (Max. recommended clearance .0035" [0.090mm]) Any honing of the block should be done with a deck plate bolted in place and torqued to the required head bolt torque. Cylinders will distort slightly and using such a plate insures that when the head is torqued down the bores will be round. Generally, bores should not have more than .0005 taper to them. To get accurate, straight cylinder bores requires the use of a Sunnen honing machine, or similar equipment. I do not recommend hand honing, except as a last resort and then only if you are well experienced at doing so. All too often, with hand honing, the hone spends more time in the center of the stroke and you end up with a "barrel" shaped cylinder. This makes ring seating very difficult and is very hard on piston ring lands, as the rings move horizontally during each piston stroke, very quickly wearing out the piston ring land and generating unwanted friction and heat.

Note: The dimension printed on the piston box usually denote the bore size required in the block for the enclosed pistons to meet specification. I say USUALLY. Do not leave this to chance. Whenever I have a block bored/honed, I ALWAYS have the pistons available for the machinist, so that he can personally check the size and fit.

Next, you will need to temporarily assemble certain components into the block to double check critical dimension. After making sure that the block is clean and dry install the crankshaft with the three main bearings and thrust bearings in the block. Install these dry, as at this point we are not assembling the motor yet, only checking dimensions. Lay the crankshaft in the block. Then place a dial indicator against one of the counterweights of the crankshaft and test for thrust bearing end play. I like an end play reading around 0.003 0.005 inch (0.076-0.127mm). Even clearance up to 0.010 inch (0.245mm) is probably OK, but you will have to double check other clearances more carefully.

2.3 - Lubrication System

The standard configuration for Fiat blocks is an oil pump within a wet sump, attached to the bottom of the block. This oil pump feeds oil to the various components of the block (and head which will be discussed later).

One problem associated with 817/843 and some early 903 blocks is the implementation of a partial oil filtration system, using a centrifugal filter in the front pulley. This implementation also did not have a direct oil supply to the center main bearing. Abarth realized this problem and in all of the Abarth derived blocks this was changed. I also recommend the following:

First, ANY orifices that have sharp edges on them should be radiused. Normally after oil is picked up by the pump, it goes to the pressure relief valve in the block. This area needs some serious cleaning up and blending to make sure that we get good oil flow. (1972 or later 903/A112 engines have the revised oiling system already and also have a filter mounted to the block.) For a race motor I recommend that you have a new crankshaft pulley machined from steel or aluminum. The pressed metal ones that are silver soldered have been known to come apart.

Follow the instructions below for reverse engine oil flow and pressurized center main bearing on earlier 843/OT1000 blocks:

1. Drill a 0.187 inch (4.5mm) diameter hole in the center main bearing saddle (use a center main bearing with the requisite hole to locate the position) on an angle to intersect with the bottom of the left main cap bolt hole. You do not have to drill very far, perhaps just a quarter of an inch or so.

This is the 3/16th hole going to the bottom of the main cap hole from the center main.

2. Remove and discard the oil galley plug from the outside of the block. (You should have already done so by now anyway for cleaning purposes) and use a long, straight 0.187 inch (4.5mm) drill bit, to drill an oil passage to intersect with the previously drilled hole. (I recommend that you do this on a vertical mill) If you allow the drill to go off-center, then you will get to start over with another block.(In my engines, as I use the rear-most boss on the oil galley for return pressurized oil supply, I tap this center entry in the oil galley for a 1/8th pipe fitting for an electric oil pressure sender.)

3. Carefully enlarge the outer hole at the oil gallery and tap for a fitting for the oil pressure feed line for the block. Remember this is cast iron and it does crack. A "-10" Aeroquip fitting with a pipe fitting on the other end is fine. Make sure that you install this fitting with a sealer or Teflon tape. This is where oil goes in to pressurize the entire engine.

4. Next turn the block upside down and find the hole where the oil pump outputs oil. Enlarge this hole for its full depth to 10mm. (The same goes for the gasket).

5. Next to this hole is another hole (bypass return hole) that must be permanently blocked.

6. Fabricate a main cap blanking plate for the #1 main cap and install with gasket.

7. If the crankshaft you plan to use came from a motor with a centrifugal oil filter, plug the hole in the snout of the crankshaft. Again, I recommend drilling and tapping for a Allen head plug, installed with loctite.

8. You will have already removed the pressure bypass valve in the side of the block. Inspect the oil passage and relieve/open up as required for better oil flow. Remember that this is going to be the output orifice for the oil pump.

9. Make fitting to go in place of the pressure bypass valve. Again, this will have an AN -10 or AN -12 Aeroquip fitting on one side and and a metric (20x1.5mm) thread with a flat face for an annealed copper or aluminum washer on the other side. Scuderia Topolino has available a kit to do this modification.

The oil now is picked up by the pump and exits the block via the special fitting in the bypass orifice in the block.

AN-12 Fitting installed in bypass hole
Be careful when installing. Use Teflon tape and do not over tighten.

2.4 - Oils and Additives

Recent changes in oil formulations have proven to be troublesome for older vehicle is general, and for cars with flat tappet engines in particular. Up until a year or so ago, almost all engine oils had small amounts of zinc and phosphorous as part of their oil chemistry. The typical amount would have been 1200 ppm (parts per million). Manufacturers have asked that the level of phosphorous be reduced, as it has a negative effect on the longevity of catalytic converters. Oil companies have responded by cutting these additives by 75%, as they are also expensive ingredients in oil-formulation chemistry.

Both phosphorous and zinc are specifically indicated friction modifiers, particularly applicable to "sliding interfaces". These would include the following interface junctions:

- Cam lobe/lifter,
- push rod/lifter,
- push rod/adjuster,
- rocker arm/rocker shaft,
- rocker tip/valve tip.

Most engine oils have had the level of these vital additives reduced to 400 ppm. The exceptions to this rule are as follows:

Manufacturer	Oil Type	Synthetic/Organic	Weight	Phos. PPM	Zink PPM	
Castrol	Syntec	Synthetic	5W-40	1000ppm		Vehicle
Castrol	Syntec Classic	Synthetic	20W-50	1200ppm		Vehicle
Castrol	TWS Motorsport	Synthetic	10W-60	1000ppm		Vehicle
Castrol	BMW Long-Life	Synthetic	5W-30	995ppm		Vehicle
Castrol	Power RS GPS	Synthetic	10W-30 10W-40 20W-50	1000ppm		Motorcycle
Castrol	Power RS R4	Synthetic	5W-40 10W-50	1200ppm		Motorcycle
Brad Penn Oils	Formerly Kendall	Synthetic	Various	860ppm		Vehicle
Swepco	306					

Royal Purple	Max Cycle	Synthetic	20W-50	1200ppm		Motorcycle
Amsoil	Harley V-Twin	Synthetic	20W-50	1200ppm		Motorcycle
Cosworth	Racing Oil	Synthetic		1150ppm	1250ppm	Vehicle
Shell	Rotella T CI4	Organic		1300ppm	1400ppm	Diesel
Pennzoil	Racing Oil	Synthetic	20W-50	1800ppm	1950ppm	Vehicle
Quaker State	Q Racing	Synthetic		1800ppm	2000ppm	Vehicle
Valvoline	VR1	Synthetic	20W-50	1200pp,	1300ppm	Vehicle
Valvoline	Racing Oil	Synthetic		1200ppm	1200ppm	Vehicle
Royal Purple	Racing Oil 21	Synthetic	5W-30	1130ppm	1961ppm	Vehicle
Royal Purple	Racing Oil 41	Synthetic	10W-40	1171ppm	1901ppm	Vehicle
Redline Oils		Synthetic	10W-40	1371ppm	1350ppm	Vehicle
Redline Oils		Synthetic	10W-30	1340ppm	1407ppm	Vehicle
Redline Oils		Synthetic	5W-30	1419ppm	1421ppm	Vehicle
Mobile 1		Synthetic		1223ppm	1376ppm	Vehicle
Joe Gibbs*		Synthetic	Various	6000ppm	6000ppm	Vehicle

Joe Gibbs Racing oil also has increased levels of Sulphur.

This was the information as of the middle of Jan 2008. Obviously almost all of the racing oils had increased levels of zinc and phosphorous, vital to the proper running of an Abarth motor. You should check on the brand that you are using, to make sure that it has not been reformulated. As a general rule, somewhere between 1000 -1200 PPM would be a good minimum number for both elements.

The follow-on question is then what weight of oil should I be using. The key to oil numbers is the second number. So, in 10W-40 weight oil, the "40" part indicates that this oil has a viscosity rating of 40 weight oil at 212 deg .F (100 deg. C) however it has a consistency of 10W oil. Now the question is, just what rating do I need? Do I automatically go for 10W-60? Well, maybe !!

The key is temperature and oil pressure. At whatever you oil temperature happens to be, you should be able to maintain 75 PSI (5 Bar) of oil pressure at 6500 RPM (this happens to be Ferrari's formula for high performance vehicles). If you can do this with 10W-30 then good. If not, then move up to 10W-40, or 10W-50 etc. In other words, depending on the state of the motor, you can tailor your oil grade to maintain a pressure level that is indicated. Using a grade of oil with a higher viscosity rating, beyond this, will only cost you horsepower. Now in the top leagues of auto racing they may use 0W oil for qualifying, but this usually means a motor that was broken in on the dyno on a higher grade, and then they only care if it last 3 laps.

For those who want to know more about oils and how they perform, please read this report. Draw your own conclusions. Yes, the report was sponsored by Amsoil, but the results really do speak for themselves.

http://www.syntheticoilnlubes.com/pdf/g2156.pdf

At Scuderia Topolino we break engines in on Shell Rotella T non-synthetic oil. This is for two reasons. First it has a good zink/phosphorous additive mix to protect the camshaft during break-in. Second, it is not "super slippery", and therefore rings will seat quickly in the cylinder bores. After this we change to Redline Synthetic Racing Oil or Amsoil 10-50 Motorcycle oil. Yes, motorcycle oil, as it has the higher level of zinc and phosphorous. Alternatively, Joe Gibbs oil is probably very good, but the price my put some people off.

Amsoil has also earned a very good reputation with their engine oils in motorcycle gearboxes with a rating equivalent to 80/90W gear oil. As it has no Extra Pressure (EP) friction modifiers, it may be a good solution for Fiat transaxles as well. This oil may work well with the standard synchro rings and the bevel gear pinion used in this transaxle. As it has superior gear wear characteristics, Scuderia Topolino hopes to test the oil on our dyno, and in the transaxle during the 2008 season, in preparation for a major effort in 2009.

Reciprocating Components and Engine Balancing

3.1 Connecting Rods

There is a fair amount of mystique associated with connecting rods, but one thing is certain, they are one of the most important components of an internal combustion engine. There are many opinions about what makes a good connecting rod, and I for one am not sufficiently versed in metallurgy to go about designing one. What I can do however is look to those companies that have a good track record and try and see what they do well.

Companies like Saenz, Pankl, Carillo, Crower and many others have been making connecting rods for many years, with great success. So what is so different about a specialty rod, as compared to a standard production rod. Three words describe the difference.

Fasteners - Fit - Finish.

Fasteners - In my view the most important aspect of any connecting rod is the bolts. For connecting rods that are to be exposed to the stresses of competition, only the absolute best will do. Whether you purchase the bolts you use with the rods, or buy them from after-market suppliers like SPS or ARP, the money spend on better connecting rod bolts will pay off every time.

Standard Fiat bolts are fine for normal road applications. They rate at about 90,000 to 100,000 PSI. For an engine that is going to see 8000 RPM or more this just will not do. After-market bolts start at about 160,000 PSI and go up from there. The ultimate strength of the bolt will be determined by both the metallurgy and the size of the bolt.

Due to size restrictions, the standard bolt for Fiat rods is a fine pitch threaded 8mm unit. If you are using standard connecting rods, use the 850/903/1050 units. (Remember - The 903 rods are 2mm longer) These can be cleaned up quite and then install a set of replacement rod bolts rated at 160,000 PSI. If all the components are lightened and balanced, this will give a very robust installation.

Stretchgauge

If you want even better then use connecting rods from Scuderia Topolino, Carillo, Crower or Pankl. All offer uprated bolts for their connecting rods ranging from 180,000 to 285,000 PSI. In all cases you will note that the shank of the bolt is cut down, so that it is slightly less than the original diameter of the bolt. There is a reason for this. All bolts must be able to stretch - but only a predetermined amount. Without stretch the bolt will not properly clamp the cap to the body of the rod. If the bolt stretches too much, or fails to return to its original length when loosened, DISCARD IT. It is also important "where" the bolt stretches. The reason for the shank reduction is this is the area where the stretch is supposed to take place. The diameter of the bolt in this area will be slightly less than the root, or bottom, of the threads. If this shank reduction were not there, then the bolt would have a tendency to stretch in the thread area. This would not be a good idea. In general an 8mm bolt should stretch between .005-.006 thousands of an inch at or below indicated torque. Each manufacturer will have his own specifications, and it is important that their recommendation be followed.

One thing that all of the premier companies agree on is how connecting rod bolts should be torqued and to do it correctly this takes a "rod stretch gauge". When I build a racing motor, I record the free length of each bolt and note it in the build records of the motor. If not, how would you ever be able to check whether the bolt failed to return to its free length when slackened off. The bolt is then fitted, finger tight, and torqued to 80% the specified amount. Now you check for stretch. If the bolt has stretched less than .005 increase the torque on the bolt until the required stretch is achieved, but do not exceed the maximum torque recommended by the rod manufacturer.

If you are fitting after-market rod bolts to standard connecting rods, or to rods for which they were not specifically manufactured, pay close attention the fillet radius of the underside of the bolt head. In many cases with standard connecting rods you will have to radius the edge of the hole in the rod. If you do not do this then the bolt will make contact with a sharp edge in the radius area and it is almost certain that an early and catastrophic failure will occur.

A WORD OF CAUTION - You must fully lubricate the connecting rod bolt before torqueing. This includes the threads, shank and the underside of the head. If the bolts come with special assembly lubricant, USE IT. You would be amazed at the different friction coefficients of regular motor oil, synthetic motor oil and special assembly lubricants. Also the torque recommendations will be different for all three.

One reason why I personally did do not use Carillo rods, is that the H-Beam design mandated the use of a 1/4 (approx. 6mm) bolt, where as in the Scuderia Topolino design a 5/16th (approx. 8mm) bolt is utilized. Carillo defends their use of the 1/4 inch bolt on the basis that the rod/piston combination's dynamic forces are will within the design limits of a 1/4 inch bolt. Call me a skeptic, but in the connecting rod bolt department "bigger is better" in my view.

Fit – Each aftermarket connecting rod manufacturer has their own technical solution. Carillo's H-beam design has been the standard for racing engines for many years (Even the Chinese are now making a knockoff of this design). The Scuderia

Topolino connecting rod also includes an H-beam component, but this is combined with a certain aspects of the I-Beam design as well. The result is a connecting rod that has the best features of both designs and which is lighter overall..

Using a custom connecting rod allows the engine builder to vary the length as well. This freedom, along with innovative piston designs, may make impressive horsepower gains, but more important are the gains to be made in engine acceleration due to lowered rotating mass and better rod angularity. One extra consideration is that all Scuderia Topolino rods have doweled caps. This rod caps will have two hollow dowel pins to locate the cap.

Many times I am asked why I go the trouble of buying custom connecting rods. Many competitors have done quite well with the standard 850 or A112 rods, suitably reworked. I guess it depends on how much time you have and what the ultimate objective is. If you consider that you may spend as much preparing a set of standard rods, presuming you account for your own time, then buying a set may not seem so outlandish, even to the amateur competitor. By the time you do the following:

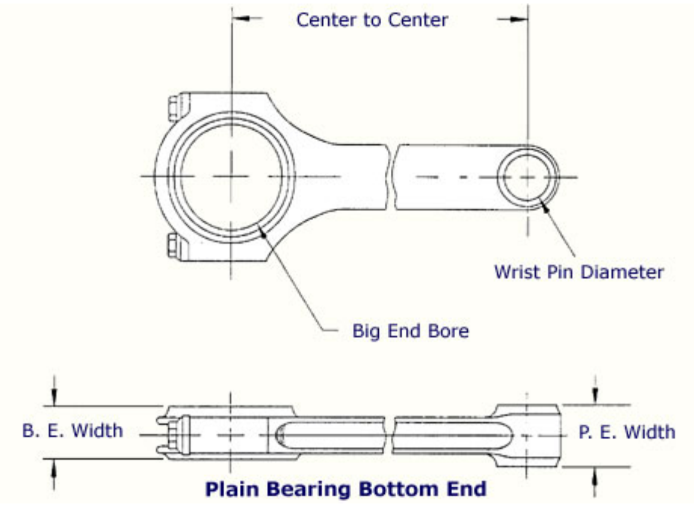
- Grind off the excess material (large blocks of metal at the big and small ends of the rod),
- Checked for axial twist and straightened if required,
- Sized or honed for bearing fit,
- Sized or bushed and honed for piston pin fit,
- Polish the rod and then have it shot peened (stress relieving after grinding); and
- Finally have the set balanced

then you may have spent more on your time, materials and outside machine shop labor than a custom set of connecting rods. A set of custom rods from Scuderia Topolino will have all of this done to specification.

Finish - In modern connecting rod technology there are two "finish" areas. One is the actual machine work quality, including proper alignment of oiling holes and making sure that the bolt holes have proper a proper edge radius. Finally, there is the overall finish of the connecting rod. While polishing really looks neat (at least to the person putting the engine together), no one else is going to see all that work, unless of course the engine disintegrates and there are lots of nice, shiny polished parts that are lying around on the track. I prefer a shot blasted finish which is the finish resulting from the stress relieving process.

The second half of the "finish" equation is surface coating. Companies like Calico Coatings now have function specific coatings, including for connecting rods they have special oil-shedding coatings that assist in getting the oil off the rods and back into the pan.

If you decide that custom rods are the way to go, then the following diagram will come in handy. All of the measurements are required in order to manufacture a custom connecting rod. One measurement not shown is whether the PE is offset from the BE. I personally do not use any offset, but if you are ordering special connecting rods, you can include this specification. In addition to these measurements, if you are building a 1050 or 982cc motor, then the width of the shank of the connecting rod will also be critical. Even if the width is no greater than a standard rod, the camshaft may still have to be relieved to prevent the rod from striking the camshaft core and the sides of lobes 1,4,5 and 8.



So why would we want to lengthen the connecting rod? Well, the short answer is of course the ever present holy grail of increased horsepower. But, just what is it that we are going to accomplish with a longer rod. Here is a short list.

1. Decrease in piston skirt side force - reducing friction and reducing parasitic losses

2. Increase in Dwell Time at TDC and BDC. This allows for better combustion control, particularly at high RPM, but from a negative perspective it may affect exhaust scavenging if the head used has a poor exhaust port.

3. More equalized piston acceleration with reference to TDC and BDC - not piston speed.

For high performance engines, where the builder has already explored better porting, valve angles, exhaust systems, camshafts etc., going to a longer rod will have more plus benefits than negative ones. Certainly longer dwell time at TDC will allow for additional time for flame front travel with high RON fuels. This will allow a small increase in static ignition advance, while still maintaining the maximum pressure crank angle on the combustion stroke. Even if everything is just right, the maximum gain that one should expect from longer rods is just around 1%. In small engines this might be 1 to 1.5 horsepower.

Connecting rod angles

Many engine builders will tell you that an optimum rod angle is 1.75. Thus, the stroke/rod length combination for the 843cc motor is pretty close to ideal. Even the 903 is pretty good, but the 982cc combination is getting pretty marginal. However the 843cc engine does have a downside. By moving the piston pin lower more side load will be produced on the piston skirt. There would be a way to optimize the situation to give the 903 and 843 engines the same high piston pin location, namely using a longer connecting rod. This give the following combinations:

- 74mm stroke - 0.995 inch (25.25mm) piston pin location - 982cc - 110mm rod length - Rod angle = 1.486
- 69mm stroke - 0.995 inch (25.25mm) piston pin location - 903cc - 112.5mm rod length - Rod angle = 1.63
- 63.5mm stroke - 0.995 inch (25.25mm) piston pin location - 843cc - 115.25mm rod length - Rod angle = 1.81

If we now look at what effect the same changes will have with the "tall deck" blocks, you will see that there is even some advantage for the 903/1050cc motor. These blocks are approx. 5.6mm taller than the short deck blocks. This means that the piston pin will be approx. that same amount lower again in the piston. Note: The 843cc motor combination would not be advisable !!!

- 74mm stroke - 1.219 inch (30.93mm) piston pin location - 982/1050cc - Rod angle = 1.486
- 69mm stroke- 1.318 inch (33.45mm) piston pin location - 903cc - Rod Angle = 1.594
- 63.5mm stroke - 1.426 inch (36.19mm) piston pin location - 843cc - Rod Angle = 1.732

Now, if we again modify the rod length to maintain a high pin position, the following results are obtained

- 74mm stroke - 0.995 inch (25.25mm) piston pin location - 982cc - 115.7mm rod length - Rod angle = 1.5636
- 69mm stroke - 0.995 inch (25.25mm) piston pin location - 903cc - 118.2mm rod length - Rod angle = 1.713
- 63.5mm stroke - 0.995 inch (25.25mm) piston pin location - 843cc - 120.9mm rod length - Rod angle = 1.90

In all cases the rod angularity is markedly reduced, which should result in some reduction of parasitic friction losses.

Quite obviously, in order to make these changes, custom connecting rods have to be used. These must of be the best possible materials and must use the highest grade connecting rod bolts, if the engines are going to survive 8000+ RPM limits. If a good quality 69mm crankshaft were available, then an interesting combination would be a 67.6mm bore with a 69mm stroke using a 118.2mm rod. This would produce a displacement of 990cc.

In an effort to "standardize" rod length configurations and to provide some economies of scale as far as manufacturing is concerned, Scuderia Topolino will make pistons for two rod lengths available, standard (110mm) and overlength (117mm). The Scuderia Topolino General Catalog lists all of the piston/rod combinations available as standard items. Of course if you need a connecting rod/piston combination that is different, please do not hesitate to ask.

> CAUTION - **Some of the rods will get quite long and only the absolute best of materials should be used**. In additional the bottom of the cylinder will likely require notching on most of these combinations and certainly on ones using a short deck block (600/850/1000OT) with a long rod.

All rods supplied by Scuderia Topolino are equipped with ARP2000 or ARPL19 connecting rod bolts. The A112 standard rod is the heaviest at 443 gram. By comparison, an Abarth 1000SP connecting rod (also used in the TCR motors) is lighter at 365 gram. Even after some grinding and polishing, it is still the heaviest. For its size, the 1000SP rod is surprizingly light. Its design is I-beam, only much stouter. The H-beam rods from Carillo weigh in at 384 gram. The Scuderia Topolino connecting rod, with a combination of H and I beam characteristics weighs 350 grams in standard form and less than 330 grams in the "narrow" version.* Finally, you could decide for a titanium connecting rod and this would weight approx. 30% less than the steel equivalent or 220 grams.

3.2 Pistons

Piston metallurgy. There are basically four types of forged pistons on the market today. Three of these are made of an Aluminum/Silicon Alloy, whereas the forth is made of 2618 Aluminium.

- 2618 Aluminum - No Silicon
- Hypoeutectitic - Less than 12% Silicon (typically 9 %)
- Eutectic - 12% Silicon
- Hypereutectic - More than 12% Silicon

Basically, silicon adds "wearability" to the alloy and in street applications would be a great plus. One drawback is that it makes the aluminium somewhat brittle. For low RPM motors this is not a problem.

For high performance applications, almost all pistons, including those supplied by Scuderia Topolino, are made from 2618 Aluminium or similar, but without silicon. This means that they will expand somewhat more than an aluminium/silicon alloy piston, but they will be significantly stronger.

While I have not specifically addressed cast pistons, the same considerations would apply, plus the cast piston will have a slightly less dense structure and therefore may not be as strong. Although some would claim that the expansion characteristics of the cast aluminum material are also less than a similar sized forged piston, for ultimate strength a forged piston is recommended

Ring Configurations - All Scuderia Topolino pistons are manufactured with the following ring pack. We find that it provides a good combination of wall tension, seating, wear resistance and cost.

- Top Ring - 1.2 mm Chrome
- 2nd ring - 1.2mm Nodular Iron
- Oil control ring - 2.8mm multiple segment oil control ring.

Engine Displ.	Bore mm	Stroke mm	Rod length mm	Block Height mm	Main Brg Bore Diam mm	PistonPin Diam mm	Comp. Height mm	Comp Height Inch	3-Ring Pin -Oil groove distance mm	3 Ring Pin -Oil groove distance inch	Rod/ Stroke Ratio	Vehicle/Engine
1049.3	67.20	74	110	177.850	57.53	18	30.85	1.215	8.14	0.321	1.49	A112 - 1074, 1077, 1077B, 1077C, 1091
981.7	65.00	74	110	172.350	57.53	18	25.35	0.998	2.64	0.104	1.49	1000OT - 1074, 1077, 1077B, 1077C, 1091
964.2	67.20	68	112	177.850	54.42	20	31.85	1.254	8.14	0.321	1.65	A112 965cc - 1074, 1077, 1077B, 1077C, 1091
902.1	65.00	68	112	177.850	54.42	20	31.85	1.254	8.14	0.321	1.65	Fiat 903cc - 1077, 1077B, 1077C
842.4	65.00	63.5	110	172.350	54.42	18	30.60	1.205	7.89	0.311	1.73	Fiat 843cc - 1077, 1077B, 1077C
846.3	62.50	69	110	173.000	54.42	18	28.50	1.122	5.79	0.228	1.59	Abarth 850 TC - 1076

STD 3 RING PACKAGE		
Top land	4.00	0.158
Top ring groove	1.22	0.048
Second land	2.72	0.107
Second ring	1.22	0.048
Oil land	2.03	0.080
Oil Ring	2.52	0.099
Total Space	**13.710**	**0.540**

2 RING PACKAGE		
Top Land	4.00	0.158
Top ring groove	1.22	0.048
Oil land	3.00	0.118
Oil ring groove	2.52	0.099
	10.740	**0.423**

Note: All pistons produced after Feb 2007 will also come standard with "peripheral gas porting" to aid on top ring sealing.

It is possible to use a 2-ring configuration. This would be purely a "competition only" setup. The overall size of the ring pack would be reduced by about 3mm, allowing the piston pin to be moved up an equal amount. This would be particularly helpful for 74mm stroke motors. The following results would be obtained.

- 843 block motor - 74mm stroke 113mm long rod for a rod angle of 1.527
- 903/1050 block motor - 74mm stroke 121mm long rod for a rod angle of 1.64

See explanation of rod angle issues under the connecting rod section.

Piston Layout Combinations – Scuderia Topolino pistons are now in their 3rd design generation. The current piston has a reduced skirt length, thinner skirt sections, tapered forged piston pin, and the top ring groove is now gas-ported for improved top ring sealing. Weights of all of these new pistons have been reduced by an average of 15%. As an example, the 3rd generation piston with pin now weights 218 gram (compared to 279 gram for the 2nd generation piston).

Scuderia Topolino now stock three different piston dome configurations. There are:

1. Flat Top. These pistons have approx. 10.5:1 compression with a standard Fiat/A112 cylinder head. These is perhaps 0.5 point higher compression than the standard cast A112 piston, which is slightly dished.

2. Small Dome with Valve Reliefs. These pistons have approx. 12:1 compression with a standard Fiat/A112 cylinder head.

3. Large Dome with NO Valve Reliefs. These pistons have more than 13:1 compression with a standard Fiat/A112 cylinder head. Caution: Depending on the type of camshaft used, these

pistons may produce more higher dynamic compression than desired.

I am sure that someone will ask why we did not make it a "full slipper style" piston. The overall goal was to reduce the weight of the pistons with pins to 250 grams or less. We have more than achieved this without resorting to a full slipper design. These were the considerations.

1. Because of the small diameter of the piston, in order to achieve a full slipper design, the engineers felt that the pin bosses would become too small to support the RPM levels expected from the pistons. At 8500 RPM the piston speed is around 68 ft/sec (21 m/sec), with 3900 G’s of force at TDC at the same RPM.

2. It was determined that not much additional weight advantage could be gained from a full slipper design, as the amount of aluminum material saved by repositioning the pin webs would me minimal.

3. The piston would retain better dimensional stability with a semi-slipper design with more predictable expansion characteristics. This would promote better ring stability and cylinder sealing.

Here are the weight comparisons between 2nd and 3rd generation pistons.

Abarth/Fiat 10.5:1, 3.5mm dome with valve pockets - 2nd Gen. 279 gram - ***3rd Gen. 218 gram***

Abarth/Fiat 13:1, 6mm dome, no valve pockets - 2nd Gen. 275 gram - ***3rd Gen. 226 gram***

Abarth TCR 10.5:1, 2nd Gen. 264 gram - ***3rd Gen. 239 gram***

Abarth TCR 12.5:1, 2nd Gen. 274 gram - ***3rd Gen. 249 gram***

Anti-Friction Coatings - Developments in anti-friction coatings have come a long way in just a few years. The coatings available today certainly provide a measurable advantage when used in a race motor. At the moment I specify coating applied by Calico Coatings. This plasma sprayed coating can be used on piston skirts, main and rod bearings, rocker shafts, lifters etc. They also have oil shedding coatings for use on reciprocating parts such as crankshafts and connecting rods.

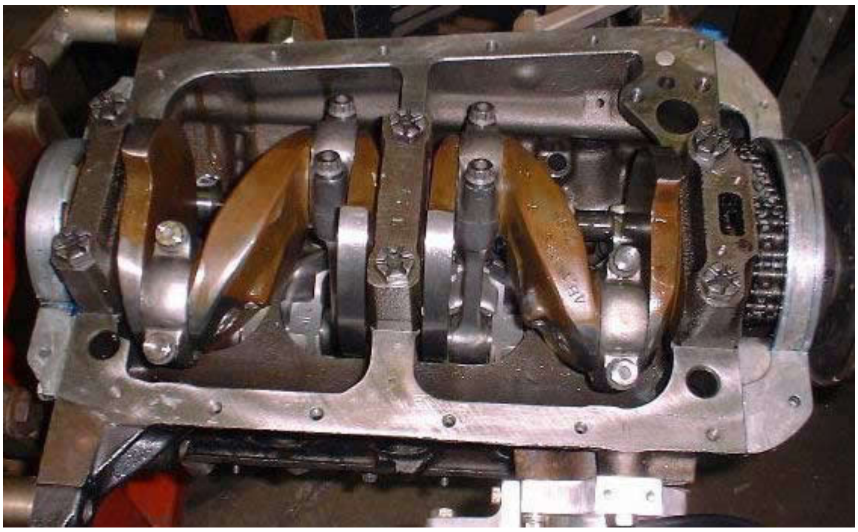
In January 2007 we started testing with DLC type coating by Sorevi (Bakaert). This company has developed several proprietary coatings by the trade name CAVIDUR. These can be applied to various engine "friction interfaces" such as rocker shafts, camshafts, lifters, finger followers and quite interestingly pistons skirts. I have also done a complete gearbox to see how the material stands up high levels of both friction and shock loading. Certainly on rotating shafts, collars and plain bearings, the material would appear to have distinct anti-friction advantages. We examined a set of lifters, camshaft, pistons and piston pins, after a full season of races and found virtually nil wear to these items.

Flame Propagation – In order to get the maximum combustion pressure from the engine, measures must be taken to insure that there is complete burning of fuel. Just because an engine has 13.5:1 computed compression does note mean that if will produce the most power, unless it can burn the fuel efficiently. A good general rule of thumb is that the higher the dome on the piston, the more difficult it is going to be to burn all of the fuel in each combustion event. In essence, the fuel charge will be distributed on either side of the dome and the spark plug is on one side of the dome. Therefore an engine running with 12.5:1 compression, and completely burning all of the fuel in the cylinder in each combustion event, may in fact produce better horsepower. About the only way to tell is to run two engines, back to back, on an accurate engine dynamometer.

3.3 Balancing Considerations

Rotational Assembly Imbalance Tolerance - and its affect on engine reliability

No one involved in the preparation and building of race engines would argue the importance of balancing. Improper, or better yet inaccurate, balancing may have far reaching implications on both the performance and the useful life expectancy of the engine in question. Balance, or the lack of it, is more than simply a matter of matching individual components. Perhaps more importantly, it is the balance tolerance limit of the entire assembly that is of vital importance. As I will explain later, a small static weight difference can have a profound negative effect at higher RPMs.



In dealing with historic engines, like the Fiat 600 engine and it many derivatives, one has to come to grips with the fact that this engine was designed well over 50 years ago, and probably to a much different design and performance criteria than used in historic motor sport competition today. After all, it is a fairly plain, garden variety 3 main bearing, OHV motor. For this very reason balance tolerance limits may be much more critical here than in say a modern, five main bearing engine such as the Duratec 2.0 litre 4-cylinder. Yet amazingly, at high levels of tune, it is not impossible to get power outputs of 100 HP/Litre from these early Fiat block designs. Over the years I have seen Fiat blocks fail in two general ways that could be attributed to "excessive balance tolerance".

- Center Main Bearing Support Failure - Here the center main bearing portion of the cast iron block literally breaks away from the block. It is sometimes difficult to determine cause/effect with this failure if it is determined that the block is broken after the engine has had a major mechanical failure (thrown connecting rod). One might argue that the rod broke first, and then the block was damaged when the block was struck by the rod. I believe the cause/effect sequence may be different. I believe that due to excessive imbalance tolerance the center main was pulled from the block and then due to crankshaft flexing further damage is inevitable.
- Cracked Front Block Surface - The failure I have seen on several A112 blocks where there is a fracture between the front main bearing bore and the front cam bearing bore. I believe this to be entirely an excessive balance tolerance problem

So, what is involved in achieving a minimum balance tolerance. First we have to understand the nature of imbalance, and how the affects of this imbalance are manifested, and then work backwards from there. Imbalance is generally caused by some type of "uneven distribution of weight". In the case of a race engine, imbalance has to deal with rotational as well as reciprocating elements. This uneven distribution of weight can be cause by several factors including

- Improper manufacturing and installation tolerances
- Metal inconsistencies (forgings or castings)
- Fasteners
- Trapped oil
- Overall component and assembly weight
- Damping (or lack thereof)

First it is important to understand the balancing procedure. Almost all dynamic balancing is done at rotational speeds between 200 and 1200 RPM. This is far from the 7000-9000 RPM that these engines are likely to see in competition. Therefore, any small imbalance at 1200 RPM will be much more pronounced at 9000 RPM.

Second, many "assume" that the crankshaft, the major engine rotational component, is stiff enough to resist bending caused by compression loads imposed on it, given that it is adequately supported in a crankcase of sufficient "beam stiffness". These are very large assumptions when it comes the 3-main bearing Fiat blocks, so it may be prudent to keep the balance tolerance as low as possible.

Let’s start by defining the types of mechanical issues that can affect balance. Most engine builders would concern themselves with the rotational balancing of the crankshaft. However, this process is more complicated that first meets the eye. The crankshaft is made either from cast iron, nodular iron, steel casting, forging or billet, in corresponding order of resistance to bending. Given the irregular shape of the crankshaft, as well as the other assembled components that make up a complete crankshaft assembly, the job of balancing is made all the more difficult. Certainly fully machined forgings or billets are the most preferable, as they should have equal dimensions for the various webs and counterweights associated with the crankshaft itself. Cast crankshafts, other than being more flexible than their forged or billet steel counterparts, may also have casting inconsistencies, causing differing dimensional characteristics and associated imbalance. Certainly it is helpful if these rotational dimensions are standardized, before any balancing is done.

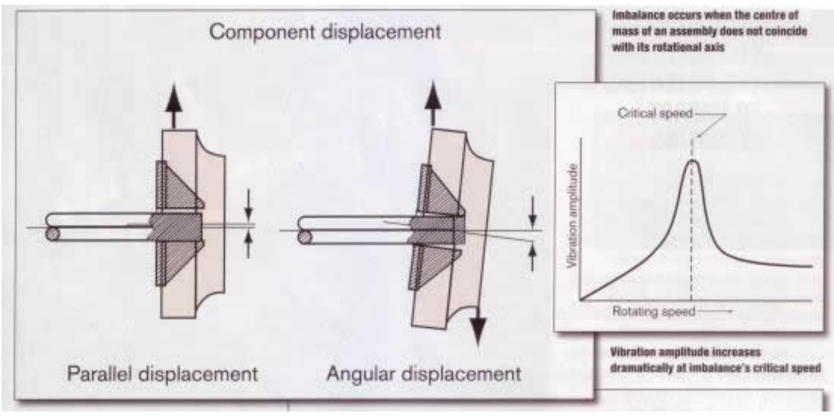
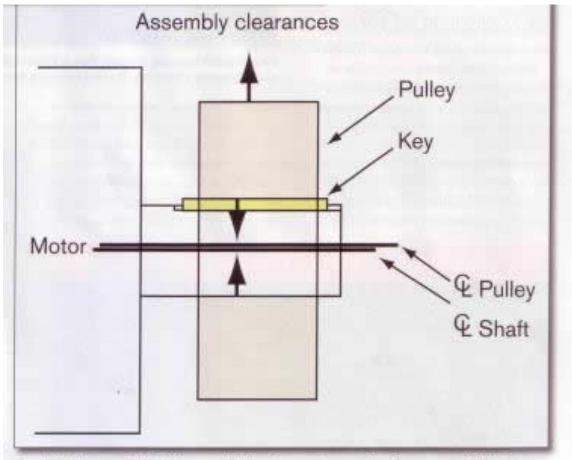


Illustration attributed to Steve Smith from Racecar Engineering article "Vibration Free"

It is also important to inspect all of the ancillary attachments to the crankshaft to make sure that they are concentric. The tin front pulley of the Fiat 600, if bent or out of round should be discarded, as it may have both parallel and/or angular displacement, contributing to overall crankshaft assembly imbalance. Better yet, it should be replaced with either a light alloy or steel machined pulley where the outside diameter is concentric with the central bore, and the front surface of the pulley at 90 degree to the bore centerline. With the pulley centerline the same as the crankshaft snout centerline, this will insure the minimum of parallel and angular displacement. The keyway bore should not have any excessive wear or clearance, as this will allow the pulley to rotate in relation to the keyway. The same argument would also hold true for the flywheel and pressure plate assembly.

There is an "order" of balancing that I recommend, but first we need to look at what type of imbalance tolerance we are willing to accept. It is simply not enough to tell the machine shop to "balance the assembly".



Acceptable tolerances of fit or balance on indvidual components can add up to an unacceptable imbalance on the complete assembly if all biases are orientated in the same direction

We can illustrate this by first balancing the four associated connecting rods (make sure the bearings are installed and rod bolts seated). This requires a scale, accurate to 0.1 gram or better, and a connecting rod weighing fixture capable of supporting the small end of the connecting rod so that the rod centerline is level and square to the top of the scale platform. The large end of the rod should be placed in the center of the scale platform, and as the rod is a fixed length, the small end will be the same distance away. In this way we can match the connecting rods so that the big end weights are within 0.1 gram of each other. How important is this measurement? According to Steve Smith, of Vibration Free" in England, "If one rod was 0.1 gram heavier than the other three, this would produce a force of 5.5 lbs (2.5 kg) at 6000 RPM for an engine with a 70mm stroke". As Abarth motors of 982cc displacement have an even large stroke (74mm), and will achieve engine revolutions of up to 9000 RPM, for a 0.1 gram imbalance the amount of force would be more than 13 lbs (5.9 kg). An assembly that was out by 5 grams would generate a force of 659 lbs (299 kg) at 9000 RPM.

Engine designers calculate the engine bearing surface to be able to support a certain load force at a given oil pressure. If this "loading limit" is exceeded, then meta-to-metal contact occurs and a catastrophic failure is inevitable!!

Obviously this is just one component out of many that may affect the balance of a complete crankshaft assembly. Once the big ends of the rods have been balanced (probably to 0.1gram or less) then the overall weight of the rods must be measured and recorded. It is helpful to temporarily number the four rods with a marking pen. Next weigh each piston assembly (complete with rings pins and clips) at match them to the respective rods so that the combinations are as close to the same as possible. Only them can you remove weight from the small end of the rods that are heavy, as compared to the lightest combination of rod/piston. Once you have them all the same, then you can permanently mark the rod/piston combinations for the cylinder where they will be installed.

Every component that rotates or oscillates has an inbuilt resonant, or critical frequency. Crankshafts may in fact operate at higher critical speed than their natural frequency and will need to be balanced in several planes along their length. As any imbalance is amplified at/near the critical speed of any component, balancing at multiple planes is essential. It is this amplified imbalance at different locations along the crankshaft that causes bending and flex, resulting increased parasitic losses and if too great engine failure.

The order of things should be as follows:

- Crankshaft only
- Add front pulley and nut
- Add flywheel
- Add pressure plate

Now you can replace any one of the ancilliary items attached to the crankshaft without having to rebalance the crankshaft itself.

There is one more factor that can be lessened, in terms of imbalance tolerance, and that is the effect of any oil clinging to the rotating and reciprocating assemblies. Engines running at high RPMs create what an "internal whirlwind" of air and oil, around the crankshaft. This is referred as "oil roping". This may add to any other imbalance that exists. This can be minimized by the following methods.

1. Installing a windage tray - This is a full separation between the rotating crankshaft and the oil in the pan. This will keep the oil from contacting the crankshaft during cornering, acceleration and deceleration.

<u>Are oil pan windage trays important in preventing oil aeration and why?</u> The answer is both yes and no. All engine oils have a level of dissipated air, as part of their makeup. At one atmosphere (14.7PSI) it is generally accepted that there is 9% by volume of dissolved air in mineral oil (Bunsen's Coefficient). Therehave been several papers written about the behavior of oil within the sump of a wet-sump lubricated engine (dry sump engines have a different set of circumstances). From the various studies, on the effect of windage tray design on surface aeration of oil, it would appear that the even at 50 PSI (3.4 times higher than atmospheric) the percentage of allowable air entrainment in oil may be on the order of 50% for rotating assemblies such as

crankshafts, camshafts and counterbalance shafts. That is not to say that 50% should be the design goal. As the percentage of entrained air in oil tracks both RPM and oil temperature, it is important to control free air in oil to a workable amount, as concentration higher than 50% will cause main failures and concentrations of 30% is considered by many to be the upper limit for connecting rod bearings due to interrupted nature of the oil feed.

From empirical testing at Ford Motor Co. the effects of oil droplets, flung from the rotating assemblies, on the free air percentage of oil in the sump (prior to entering the pump) is inconclusive. Yes, at higher RPM these high speed droplets did cause the oil to foam on the surface of the oil in the sump, but this appeared to have little effect on the entrained air percentage of oil entering the pump. Adding a windage tray to a wet sump appeared, according to cited evidence, to have little positive effect. Once oil enters the pump no additional air is added to the oil as it traverses the system. With changes in pressure there may be a conversion of some free air to dissolved air and vice-versa. A windage tray, or some other means of controlling the movement of the mass of oil in the pan, does have considerable positive value in terms of ensuring that the oil pump pickup does not become uncovered during high G force situation (Braking, cornering and acceleration - in order of magnitude), or preventing the crankshaft from causing additional aeration from physical contact with the oil in the sump.

So it would appear that the principle benefit of a windage tray is to control the "gross" movement of oil in the sump. However, while we are examining this situation, lets take it one step further.

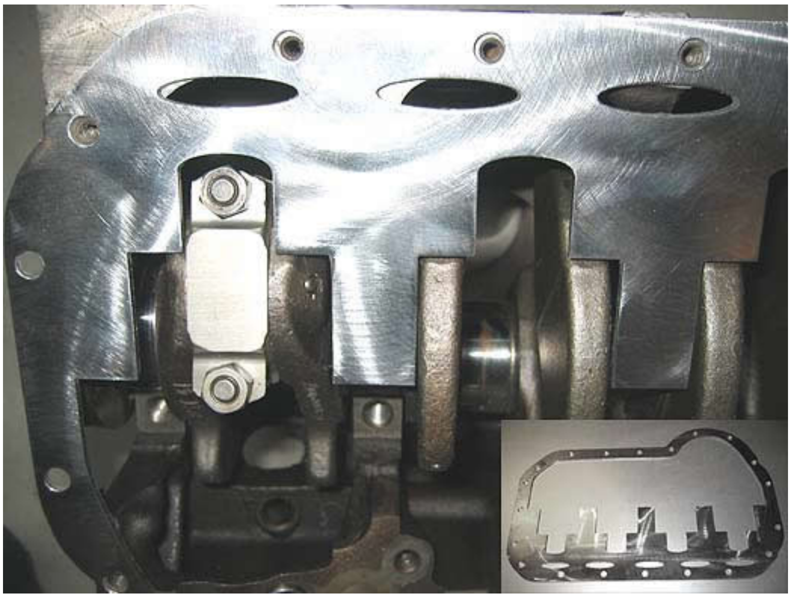
Because there is both dissolved and free air in the oil, the two must be considered together. Most test specifications dealing with oil aeration assume an 18% air-to-oil volume for dissolved and free air combined. This number also appears to be a good average to shoot for in an oil system design. As I understand it, using Bunsen's coefficient for engine oil, "for every one Bar (14.4 PSI) increase of oil pressure, the oil can take up an extra volume of air equal to 9% of the oil volume". As such at 3.5 Bar (50 PSI) the volume of air-to-oil could be as high as 31.5%. Using the old "hot-rod" formula of 10 PSI oil pressure per 1000 RPM the allowable percentage of air-to oil would equate to 49.6%.

So is this high percentage of air-to-oil what we want? **The answer is patently NO.** Quite the opposite is what is required, as there is a mechanism within the oil delivery circuit that will negatively impact the percentage of free air in oil to certain parts of the engine, particularly the upper end. Generally oil routed to the cylinder head will travel via a "sharp edged restrictor". The effect of this type of restrictor is to cause an oil pressure drop in the oil traveling to the upper end. Typically, the pressure drop will be in the order of 40%. Thus if the pressure at the main bearings were 3.5 bar, then the pressure at the rocker shaft (in the case of the Abarth motors that I work with) would be on the order of 2.1 Bar. Whereas at the oil pump, due to the increase in pressure over atmospheric, the percentage of allowable entrained "could" be more due to the higher pressure, this might place the upper end lubrication at risk. The lowering of pressure at the orifice feeding the upper end means that dissolved air may reverse from dissolved to free air. Combine the additional free air with reduced oil flow (a function of pressure and orifice size), then it is quite feasible for rocker/shaft boundary interface in an OHV engine design to be marginalized. In competition engines with more aggressive camshafts and higher rate valve springs this would give serious cause for further analysis.

One could argue that it would be beneficial to reduce the size of the restriction, thus increasing the pressure/flow to the upper end of the motor. This would have a beneficial effect on reducing the pressure drop, thus lowering the possibility of reconstituting free air from dissolved air in the oil. In addition, the additional oil to the upper end would enhance cooling due to the increased flow. However, this may decrease overall engine oil pressure and yet other measures may have to be taken to return upper end oil to the sump in an efficient manner.

So there is a fine balance to be struck if one were to contemplate running lower oil pressures to reduce parasitic losses. Certainly the use of a windage tray to control gross oil movement in the oil sump and keep the oil from coming in contact with the rotating assembly is a good idea, as the lower the entrained air percentage per volume of oil, the better. Further, the use of an effective "oil scraper" to strip oil from the rotating assembly and return it to the sump would also be recommend.

2. Installing a crankshaft scraper - A crankshaft scraper is a device attached to the main caps of the engine that is contoured to very close tolerance of the counterweights and cap areas of the connecting rods. The scraper assembly may be as close as 0.015 inch (0..4mm) of these surfaces and is intended to scrape any remaining oil from the crankshaft counterweights and rods.



The combination of an effective windage tray and crankshaft scraper will greatly reduce the impact of oil on the balance of the crankshaft assembly. Perhaps even more importantly, by reducfing the amount of oil clinging to the crankshaft and connecting rods dyno tests have shown an increase in horsepower of up to 5%.

This is of course for wet sump situations. If a dry sump is used a windage tray is not necessary, but a good crankshaft scraper my still reduce parasitic roping losses.

The instances of block failures are, in all likelihood, due to larger than allowable imbalance tolerances. A large out-of-balance condition may cause the beam stiffness, available from the standard Fiat block, to be exceeded. This will cause the engine to "pound the bearings", causing eventual main bearing and/ or block failure. Plus, the effect of maintaining strict balance tolerances will pay off in added reliability and performance due to the reduction of parasitic losses.

My thanks to Steve Smith of Vibration Free in Oxon UK for some of the material used in this discussion.

3.4 - Compression Ratio Fundamentals

There is a basic difference between “Calculated Compression” and “Dynamic Compression”. Both numbers are important, but the one that will make difference in the reliability of a race motor will be the dynamic compression.

Calculated compression is basically the ratio between the volume of the cylinder and combustions chamber, divided by the combustion chamber volume. In the case of a A112 motor with a standard cylinder bore, this could be as follows:

Cylinder volume (per cylinder) 259.75cc

Plus, Combustion chamber volume 28.00cc

Equals 287.75cc

Divided by, Combustion chamber vol. 28.00cc

Equals 10.28

So this motor would have a Calculated Compression of 10.28:1 and would run quite happily on high octane pump fuel.

For most competition engines we will want to raise the compression ratio to around 12.25:1, and in some cases we may even go as high as 13.5:1.

This calculated compression ratio is the beginning of a much more complicated calculation to determine the “knock resistance” of the engine. This will involve both the camshaft intake valve closing point, the octane rating of the fuel to be used and, the Dynamic Compression ratio.

First, we have to compute Dynamic Compression ratio. I am not going to list the actual formula for computing it here, but this information can be found in the internet for those who are really interested.

Here is a comparison of different camshafts in a 1046cc motor with 110mm rods, 12.25:1 static (computed) compression ratio.

Camshaft	**Duration/Lift**	**L/C**	**Overlap In.**	**Int. Closing Deg**	**Dynamic Displ (DD)/DCR**
SLR300S	290/300/12.4mm	107	81 deg.	69 deg.	804cc/9.49:1

		deg		ABDC	
SLR300	300/12.4mm	108 deg	84 deg	74 deg. ABDC	765cc/9.07:1
Kent FT 6	304/10.8mm	106 deg	92 deg	78 deg. ABDC	732cc/8.73:1
PBS A8	305/10.6mm	108 deg	89 deg	76.5 deg. ABDC	745cc/8.86:1
CatCams	305/11.45mm	108 deg	89 deg	80.5 deg. ABDC	711cc/8.50:1
CatCams	310/11.45mm	108 deg	94 deg	83 deg. ABDC	689cc/8.27:1
Abarth 316	316/10.4mm	105 deg	96 deg	88 deg. ABDC	644cc/7.80:1
Laur 319	319/10.5mm	108 deg	103 deg	87.5 deg. ABDC	649cc/7.84:1
Abarth 336	336/11.7mm	105 deg	126 deg	93 deg. ABDC	634cc/7.70:1

From the above, it is obvious that if someone asks you what compression ratio you are running, the answer will be meaningless, without telling the person asking the question, additional details.

The formula for computing horsepower is: **Horsepower = rpm x torque / 5252**

From this you can deduce that if all of these camshaft combinations were to produce the same "peak" horsepower, then those camshafts with smaller valve overlaps, and early intake valve closings, will produce higher torque/horsepower at lower RPM and taper off before 8000+ RPM, whereas the cams with large valve overlaps, and later intake valve closings, will produce lower torque numbers and will rely on higher RPM levels (perhaps as high as 9000+) to make the same "peak power".

Peak horsepower is great for bragging rights, but we don't actually spend a great percentage of our time running at 8000 RPM or more. Much more time is spent between 5500 and 7500 RPM, so this is where we should aim for the best engine efficiency. The better indication is to plot the torque/horsepower numbers 5000-7500 RPM and to find the highest "average" horsepower and torque in that range. Then choose a camshaft that will best deliver this. This will, in most cases, provide the best overall performance in a road racing vehicle.

There is a direct relationship between the "Dynamic Displacement (DD)" of an engine and the closing degree of the intake valve. This, then in turn, determines the DCR of a motor. This is one of the critical design criteria in building any race motor. The earlier that you can close the valve, without creating negative pumping effect, the greater the DD and the resultant DCR. You can also vary the DCR by advancing/retarding the camshaft up to 4 degrees. The higher the DCR, the greater the knock sensitivity, and therefore the greater the fuel octane requirement.

You will notice that this one engine can in fact produce a Dynamic Compression (last column in the chart) ranging from 7.70:1 to 9.49:1. It may not seem immediately obvious, but it is unlikely that all of these engines will survive using fuel of the same octane rating. From my own experience I know that a dynamic compression of 7.99:1 will have a "knock rating" of approx. 4.4 @ 6000 RPM. This is very close to being marginal for 100 octane fuel, but with no more than 28 deg. of total distributor advance. However, changing the camshaft to one that produces a dynamic compression of 8.99 will require a fuel with an octane rating of 105 in order to maintain a knock rating of 4.4 or less. Now if we jump up to the most aggressive camshaft, with a dynamic compression of 9.45:1, nothing less than 112 octane racing fuel will do.

As you have probably noticed by now, compression ratio cannot be viewed alone, without taking into account ignition timing and fuel octane. I suggest that all who are interested knowing more about "fuel octane" numbers go to the following link and read the information carefully. Here in 3 pages is a very good explanation of the different rating measurements and how to compare different rating systems.

http://www.btinternet.com/~madmole/Reference/RONMONPON.html

After you have read and understood this information, the following will make more sense.

Note: I am running my own engine at a computed compression of 13.5:1, The camshaft that I an using computes to a DCR 10:57:1. With DCR levels this high you MUST use very high octane. For this motor I use Sunoco 112 octane leaded racing petrol. In the interest of "longevity" I have reduced the static compression ratio to 12.5:1, which has lowered the DCR to approx. 9.67:1

Valve Train and Camshafts

4.1 Valve Train Friction

The valve train is one of those almost forgotten items in a race motor. Like most other things, there is lots of folklore about what is good and what is not. Over the years I can count the amount of time that I have spent playing around with valve train components not in hours or days, but more likely in months.

- First, it is important to understand that the valve train assembly consists of everything from the cam bearings to the valve seat, and everything in between.
- Second, the valve train also accounts for the highest percentage of total parasitic loss within an engine. Whatever can be done to reduce parasitic loss WILL make a difference, albeit small at times.

Cam Bearings - Letâ€™s start with the cam bearings. As Scuderia Topolino does on great deal of work on A112 type motors, I will focus on this type of installation, but the general ideas are applicable to almost any internal combustion engine. There are basically three cam bearings. The front one is pre-sized, but the middle and rear must be bored-to-fit in the block. This is a very important step, as the specifications for the cam bearing clearance are 0.001-0.0015 inch (0.025-0.037mm). The first task in optimizing the valve train, is to check the straightness of the camshaft. Knowing the bearing tolerance, the camshaft cannot be out-of-true by more than 0.0005 inch (0.012mm) or it will bind in the center bearing or have insufficient clearance. Next, the camshaft lobes should be in good condition and/or polished lobe surfaces . Ultimate friction reduction would involve DLC coating the bearing races and the lobes, to where the Ra value of the surface approaches 0.1.

Lifters and Lifter Bores - Next we need to look at the lifters and lifter bores. The foot of the lifter should have a slight convexity. This means a new lifter will have a slight crown to the foot. If you want to visualize this, take two new lifters and place the feet against each other. You will notice the curvature more readily when you do this. This curvature insured that the lifter turns a small amount each time the lobe lifts the lifter, to distribute the wear. If you ever find a lifter with a distinct wear pattern, then it likely that it has lost its convexity and is not rotating. In the race engines that I build in generally hone the lifter bores with a special small cylinder hone that has cork contact areas. I use a special compound, deposited on the cork and use this to provide a very light surface polish to the lifter bores. Again, if the ultimate in friction reduction is desired, then you could DLC coat the stem of the lifter. I do not recommend drilling any holes in the lifter. This significantly weakens the lifter and failure of the lifter can be the result. Be sure to inspect the push rod seat in the bottom of the lifter. This should have a shiny appearance and should not have any defects.

The Push Rod - The next item to consider is the push rod. I know that all of you will have seen advertisements for all types of alternative push rods, ranging from one aluminum ones to carbon fiber to metal-matrix-composite. Just keep the following in mind. The push rod should be as stiff as possible, in order to maintain proper cam lobe to lifter tracking, at a price you can afford. That said, yes, a push rod made from 3M aluminum matrix composite (not aluminum tubing) is lighter and stiffer than an equivalent 7mm 4130 steel push rod. However for most competitors it looks less appealing when the price of over $200 per push rod is realized. Because of the nature of a tube, as compared to a solid rod, a tubular steel push rod provides a good balance between performance, durability and cost. The last thing we want is for the push rod to flex. This produces unwanted harmomics in the valve train (valve spring to be exact) and will cause valve train failure.

If we divide the valve train into the components that are on the lifter side of the rocker arm fulcrum and those that are on the valve side of the rocker arm fulcrum, then it is more important to reduce the weight of all of the components on the valve side.

The Rocker Arm - Next is the rocker arm itself. Its weight is divided on either side of the fulcrum, and for most instances it is a bout a 50/50 proposition. Taking weight out of the tip side (where it contacts the valve stem) will help. Be careful how far you go, as you do not want to lighten it so much that the rocker becomes unreliable. In the rocker arm alone are three distinct boundary layer friction interfaces. There is the adjuster-to-push rod cup, the rocker-to-shaft, and the tip-to-valve stem.

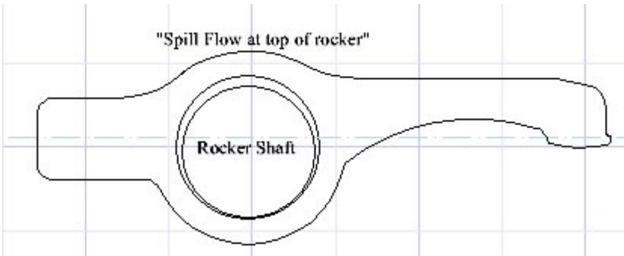
To recap, we had a cam-lifter friction interface, a lifter-bore interface, a lifter cup-push rod interface, plus the three associated with the rocker arm. Not hard to see why this is such an important area, as none of these interfaces are hydrodynamic film interfaces* and have some sliding friction action associated with them.

The push rod cup-adjuster interface is one where little improvement can be made. I have considered making a very small orifice in the rocker arm which would exit on the push rod side of the rocker arm, behind the adjuster. This would allow some oil to squirt onto the adjuster and run down into the push rod cup. This would provide a level of oil cooling to this interface, as otherwise it simply replies on whatever random oil happens to splash into the cup. If you have a situation where the top of the push rod is discolored (brown or blue), then it is getting much too hot and additional cooling is required. This is generally a result of excess valve spring pressures and or lack of lubrication.

The actual rocker arm is the next item that we should take a look at. As you know, many used Fiat rocker arms have excessive play, even when installed on a new shaft. Even worse, as the metallurgy of the rocker and the shaft is similar, they both wear equally, making the problem more pronounced. I recently measured about 40 rockers and NONE had the required 0.002-0.0025 clearance, if used with a new shaft. New rockers are available (at about $40 per rocker) but are in limited supply. Of course a new rocker arm, without any metallurgical improvements, would suffer the same fate.

The problem of rocker longevity, and damage, is one that rarely affects standard road going Fiats, as these are generally not subjected to the extra stresses imposed by high lift camshafts and the associated parts. Yes, a street engine, poorly maintained with 50,000 miles on the odometer will have worn rocker arms for sure. In racing applications, where we run with stronger valve springs, much more aggressive lobe designs and much higher valve lifts, the dynamics are totally different. ALL of these put additional stress on the rocker/shaft boundary layer interface. As I mentioned before, even standard rockers from road cars will show signs of damage if the oil to the shaft has been less than adequate for some reason.

If we look at the dynamics of the rocker-to-shaft interface, the first rule of thumb to remember is that oil will take the <u>course of least resistance</u>. Therefore, if there is an excess of clearance, the likelihood of high spill flow is very likely. If there is no pressure (only flow) then, <u>when spring pressure is applied to the rocker and all of the clearance moves to the top of the rocker, the oil flow will go to where the clearance is</u> . Thus the very area that needs the oil, THE BOTTOM OF THE ROCKER, will not get any.

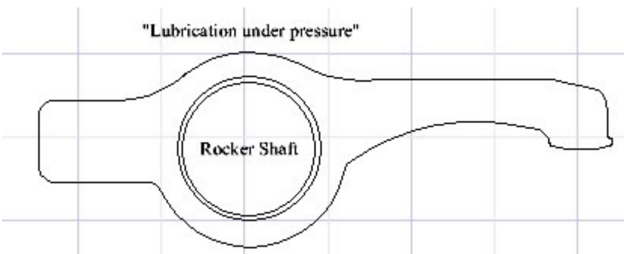


The second issue with the standard shaft, in a racing environment, is that the supply hole is too small to allow sufficient flow to service the 8 rocker oil delivery holes. As such, there is a pressure drop at this small inlet orifice and this will restrict the flow of oil. If, as I described above, there is excess clearance and the flow to the rocker is reduced, then all of the available flow will escape to the top of the rockers (where the excess clearance is) of the rockers immediately adjacent to the rocker shaft supply inlet hole. In this case, when the oil is hot, it is quite possible to have NO OIL PRESSURE build-up at the rocker arm/shaft boundary interface. This will then mean that surface friction will go up dramatically and heat will not be carried away as there is no oil flow at the rocker/shaft boundary interface. The end result is a seized rocker arm. (Note: Please read the section in blue later on for more clarification)

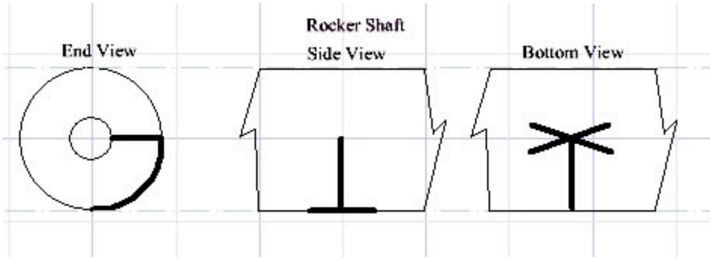
The solution is therefore to be found in four modifications.

1. Increase the supply hole to the shaft (remember there are two, even though one is used at a time) to 0.150mm (3mm) diameter. This is equal to the total area of the rocker supply holes** (0.027 inch [0.7mm] each) in the shaft. In this manner the input and output flow capabilities of the shaft will be balanced and insure that adequate flow and pressure are available to the rocker/shaft boundary interface.

2. Insure that the rocker-to-shaft clearance is approx. 0.002 inch(0.05mm), but no larger than 0.003 inch (0.075mm). This will insure that the supply flow to the rocker is greater than the "spill flow", and allow pressure to build up at the rocker/shaft boundary interface. This will promote better lubrication and cooling of this interface. As a secondary issue, the spill flow must be under sufficient pressure to cause oil to "splash" into the push rod cup. This is the only method for both lubrication and cooling this vital pressure interface. The surface area of the adjuster, where it rides in the push rod cup, takes the full force of the lobe opening pressure and must be adequately lubricated and cooled.



3. Modify the shaft, by providing a partial groove from the rocker oil delivery hole to the bottom of the rocker shaft, and then scoring an "X" at this point to spread out the oil. If the correct rocker/shaft clearance is maintained, then this will provide a wide cushion of oil for the rocker to work against.



4. Finally, where rules allow, the engine preparer may opt to install a small oil spray bar inside the valve cover, fed from a small line external to the motor. This will help insure that both the push rods and the valve springs are adequately oil cooled by high pressure oil. This is a common modification in many OHV motors, particularly in NASCAR, where flat tappet designs with very high valve lift are commonplace.

Further Information and Calculations

When a rocker is under tension, it is up against the bottom of the shaft and, there will be no oil flowing into the rocker/shaft boundary interface pressure point (presuming that the feed hole is on the bottom of the shaft). Considering that each rocker is under tension approx. 250 degrees out of every 720 degrees of crankshaft rotation, it can be assumed that, for the remaining 470 degrees of engine rotation, oil will flow through the rocker/shaft boundary interface clearance (approx 0.002 inch). This flow would re-establish a lubrication supply to the rocker/shaft boundary interface in preparation for the next 720 degree engine rotation cycle for that particular rocker arm, and, most importantly, carry away heat generated during the previous cycle at this boundary interface.

According to my observations there are 4 rocker arms in tension (to some extent anyway), at the rocker/shaft boundary interface, at any given point in time. During this "tension period" all the clearance is at the top of the shaft, with virtually nil clearance at the boundary layer pressure point (also the oil feed hole) on the bottom of the shaft. As 4 of the eight rocker feed holes are occluded, either partially or completely, any oil flow would be diverted to the remaining four rockers not under tension, with equidistant circumferential clearance.

Based on my earlier computations of cross-sectional flow area, each rocker's 0.070 inch (1.75mm) feed hole will flow about 10% greater volume than the spill volume of the 0.002 clearance between the rocker and shaft. The spill volume computes to an area of .004 sq. inch (2.58 Sq mm) per rocker arm. As such, the aggregate spill volume for 4 rocker arms would be 0.016 sq. inch. The single oil supply hole to the shaft must be 0.145-0.150 inch in diameter ,to supply sufficient flow, to service any **four** non-occluded rocker arms at any given point in time.

The metallurgical inconsistencies associated with a steel rocker arm against a steel shaft are not acceptable to for a high performance application. A better alternative would be a harder rocker shaft, made of 4130 steel and hard chrome finished for a surface hardness of RC65, and then an associated rocker arm with a pressed in bronze bushing. This combination of differing metals will prevent rocker arm seizures due to microwelding.

Scuderia Topolino has already undertaken steps to produce this new type of rocker shaft and rocker design.

With a conventional Fiat rocker arm the pad on the end of the rocker arm is larger than the stem of the valve that it contacts. Many times I have see this pad marked with a small â€œhalf moonâ€? as indication that there is a great deal of pressure concentrated in a small area in this interface. I will talk some more about the causes for this pad damage in the camshaft dynamics later on. It has always been my contention that it would be better if we could make maximum use of the total surface area of the contact pad, and hence many of the motors that leave Scuderia Topolino have â€œlash pads or lash capsâ€? over the end of the valve stems. This serves two purposed, both to protect the end of the valve stem and the pad on the rocker arm. Yes, it does add a minute amount of weight to the valve train on the â€œsensitiveâ€? side of the rocker arm fulcrum, but the benefits far outweigh the weight penalty. The pad and the associated lash cap should be polished to a fine finish to reduce parasitic losses.

One other possibility is to use an alternative rocker arm. Scuderia Topolino has available an aluminum rocker arm with a rollerized tip. The aluminum rocker arm body is made from 2024-T6 aluminum and is then hard anodized. This provides a surface hardness of RC62, and combined with the ductility of 2024 aluminum, provides a usefull wear surface against a steel shaft.

* *There is some evidence that at higher RPM the lifter-cam lobe interface converts from a boundary layer interface to a hydrodynamic interface.*

Valve Spring Retainer â€" Here is the first area where we can make a real weight savings. By changing to a titanium retainer we can cut the weight in half. It may not seem like a great deal, but it will make a decided difference in the overall dynamics. Scuderia Topolino uses special retainers that use special collets, with an included angle of 6 degrees, as opposed to the standed Fiat ones which are 5 degrees. We can also provide these retainers with 7 degrees collets, and then the collets can be provided in titanium as well.

Valve Springs - We finally pay attention to these when we install a new camshaft, to check that we do not have coil bind. There are however a number of other considerations that must be examined in terms of valve springs.

Most standard valve springs are selected on the basis of basic performance and longevity. Obviously when the 4 cylinder Fiat engine was first developed, what with a whopping 27 horsepower, the requirements of valve spring performance was not very comprehensive. After all, with a camshaft with a total lift, at the valve, of less than .300 thousands of an inch (7.6mm), the real criteria was to use a spring that would last a long time. These springs had a sufficient number of coils so as to not be very highly stressed.

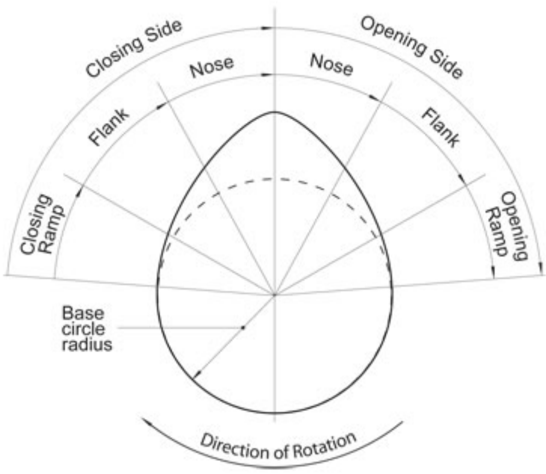
In a racing situation the requirements are just the opposite of that of the standard road car. First, whereas the standard engine uses relatively low RPMs, therefore spring harmonics play a minor role. Not so in a competition engine. Modern racing camshafts not only open the valve more and for a longer period, they also open the valve quicker as well. This means that the modern valve spring must be able to control rapid valve train acceleration. We need to find a good balance between spring pressure, harmonic control and minimum parasitic loss. Without a great deal of trial and error, the only other alternatives is to computer model the valve train, or to follow the recommendation of the camshaft supplier. The cam grinder will almost ALWAYS be very conservative in choosing a valve spring. In most cases he would rather err on the side of a too heavy spring, than one that is too light.

Valves â€" Here again, we are confronted with a choice between ultimate weight savings and cost. One way to reduce the weight of the valves is to use a small diameter stem, say from 7mm to 6mm. Also, many racing oriented valves, because the way they are formed, are inherently lighter than the standard Fiat valves. Finally, if the pocketbook will allow, you could go for titanium valves. This will greatly reduce the weight. I am fairly comfortable with using titanium intake valves, but titanium exhaust valves have a very short lifespan, and should be replaced at least once each season. This of course adds to the cost.

Valve Seats â€" While technically there is nothing that can be done to lighten valve train components with the valve seat, if you plan on using titanium valves, then there will be a cost impact that is not insubstantial. Unfortunately you cannot use steel valve seats, as the titanium valves are subject to a phenomenon call â€œmicroweldingâ€?. Tiny amounts of steel are transferred to the valve during operation and eventually the valve no longer seals. This means that either special copper or berrylium valve seats must be used. Unfortunately these are from 6-15 times more expensive than a steel seat.

4.2 Camshaft Dynamics

Camshaft and Induction Dynamics



In an effort to better understand camshaft technology, I asked a number of fellow Abarth competitors what type of camshaft they were using. I wanted to take these various camshaft designs and put them through a computer based engine simulator to compare the various grinds. Below you will find a spreadsheet of the information that was provided.

Manuf	Adv	Lobe	Cam	Intake	Overlap	Open/Close

	Duration	Center	lift (mm)	L/C		Adv.
CatCams	305	108	7.21	108	78	39/79 79/39
PBS A8	305	108	6.98	108		
PBS A6	292	106	6.85	104	72	40/72 80/32
SLR286	286	106	8.43	106	74	37/69 69/37
SLR300-106	300	106	8.43	104	88	46/74 78/42
SLR300-110	300	110	8.43	105	80	45/75 85/35
Kent FT6	304	106	7.11	106	92	46/78 78/46
Alquati	316	110	6.85	108	96	48/88 88/48
Abarth	316/304	105	7.21	102	100	53/83 77/47
Abarth	336	105	7.72	105	120	60/96 96/60

As you can see, almost every competitor is using a different camshaft. I did find that two people were using the same Kent FT6 camshaft. Duration ranged every where from 286 to 336 degrees. Certainly there were some interesting surprises when I ran some of these profiles through my analysis program. My first reaction was that the larger the duration, the more horsepower. Not quite correct. At the end of this study I will rate each of the above cams that I was given "advertised" duration information for.

First I have to state the assumptions that I used to make all of the comparisons that follow.

Bore	68m
Stroke	74mm
Rod length	110mm
Compression	13.8:1
Cylinder head	PBS 8P
Inlet Valve	31mm
Exhaust Valve	27mm
Carburetion	2 x DCOE40

There is a complex "ballet" of inter-acting numbers that define a particular camshaft and what it is capable of delivering. As with anything in life, a good "plan" is always worth the time it took to develop. Such a plan is also required when deciding upon a camshaft and the other components it will be required to interact with. In my mind there are three principal elements. It is very much like a 3-legged stool. It cannot stand unless all three parameters are well thought out and developed.

A) The maximum RPM that will be used. - It is of little value to specify a camshaft where horsepower is likely to be produced above the RPM range where it will be required.

B) The RPM where maximum torque will be expected. - This may have more to do with shift points and such, but it is something that can be adjusted for to some extent.

C) Best Available Octane Fuel â€" This will affect which camshaft can actually be used without encountering detonation.

From the chart above, it is obvious that different competitors had different things in mind when choosing a camshaft. I am of the firm belief that, given the design of the small Fiat motors with only three main bearings, that a top RPM around 8500 is not only prudent, but the only way of assuring reasonable reliability. Therefore, I will go out on a limb and say that anything over 304 degrees duration is may too great, as it puts the power production too high in the RPM range. Now there will be people who disagree, but don't dismiss the idea just yet.

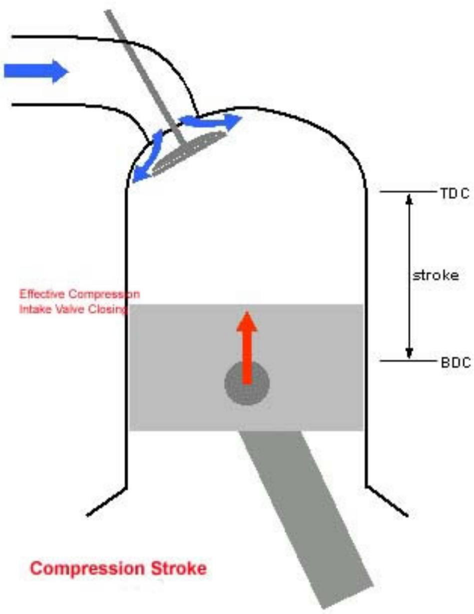
The PBS head and intake manifold/Weber carburetor combination has an overall inlet tract length of around 330mm. Generally the carburetors would have 30mm chokes. We want to maintain a mean a peak torque number somewhere between 5500-5800 RPM. Below is a chart of the calculations that I did for a 1050cc motor (68mm bore 74mm stroke, volumetric efficiency of 85%) for the diameter of the inlet port at the head/intake manifold interface.

Peak torque RPM	Inlet Diam.(mm)
5000	21.8
5250	22.8
5500	24.1
5750	25.1
6000	26.2
6500	28.4
7000	30.7

All PBS heads are machined for a 25.4mm port, so it would appear that there is a good match. This size port assumes an air velocity of just under .6 mach, or 650 ft/sec (196m/sec). So to match these head characteristics we will be looking for a camshaft that will deliver peak torque between 5000 and 6000 RPM.

Finally we need to look at third leg of our three legged stool, namely Fuel Octane Rating, and this will then tie directly to Dynamic Compression Ratio (DCR), which is quite different from the Computed Compression Ratio (CCR). We should all be familiar with how CCR is calculated. Basically it is the displacement of the cylinder, plus the displacement of the combustion chamber, then divided by the volume of the combustion chamber.

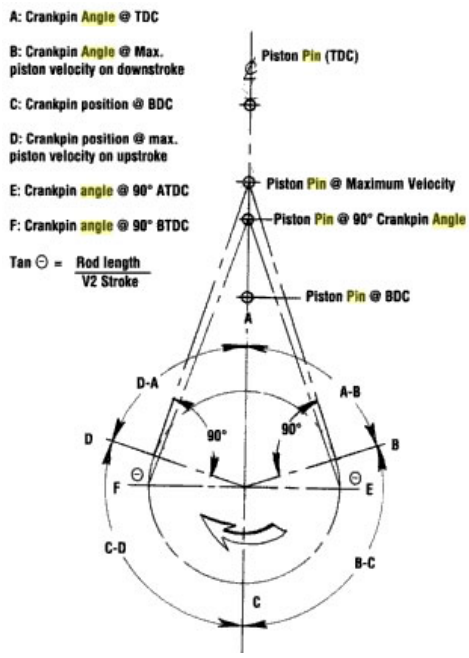
For our "standard" engine I stated that the compression ratio would be 13.8:1. Of course the intake valve is NOT closed for the entire compression stroke (BDC to TDC). In fact, even though the piston has already passed BDC, air is still flowing into the cylinder due to scavenging effect. The key is adjusting the intake valve closing position so that it coincides with the point where intake flow ceases.



According to the camshaft specifications in the chart the intake closing is anything from 65 to 96 degrees. With such a wide variance, it will be interesting to compare.

A small story *- At a recent race at Hockenheim I ran across Bram Paardenkoper. During our conversation he mentioned that he had installed an Alquati 316 degree cam in his car for the weekend, as he was trying to find some extra pace. He indicated that over the years he had done quite well, but now the competition was starting to reel him in. He was however disappointed as the car, with a cam with 12 degrees more duration, was actually slower than before!! The Alquati cam is very similar to the 316 can in the list, and the cam Bram used previously is very close to the Kent FT6 grind. This led me to do some further analysis.*

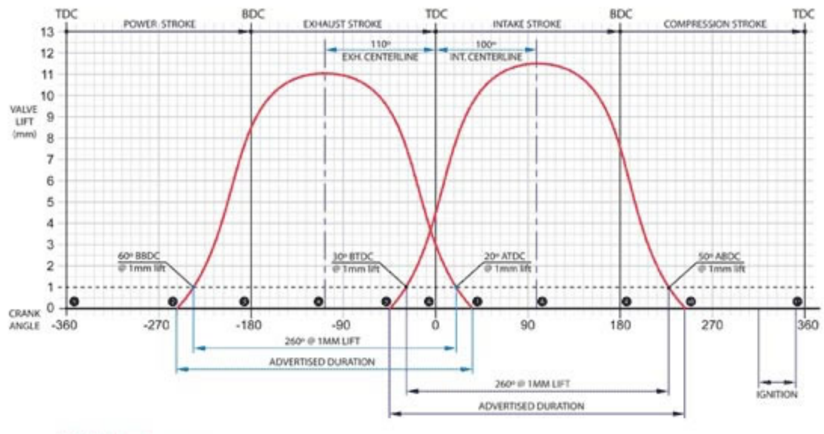
I started out by looking at where the rod and crank throw were at 90 degrees to each other on the compression stroke, in terms of degrees of crankshaft rotation, and how many degrees this was from the point of maximum piston acceleration. Perhaps this diagram will help in understanding the the reason why the rate of piston acceleration is not uniform over the entire stroke.



It turns out that for the A112 engine this is at 71.4 degrees before TDC. Conversely, this means that on the compression stroke the point of maximum acceleration for the piston is 108.6 degrees from BDC.

The most important consideration for any engine is the timing of the closing of the intake valve. The valve must be closed before this point of maximum piston acceleration to minimize the effects of pressure reversal in the cylinder. Of course the earlier that we can close the valve, and still meet our design objective with regard to power generation, the better the performance will be. After modeling many camshafts for use in the A112 motor, with its particular bore and stroke characteristics, it would appear that a valve closing period between 65 and 78 degrees ABDC produces the best results, at least for engines like the A112. Coincidentally this also places the closing event prior to the crankpin achieving a 90 degree angle with the connecting rod centerline, so that the intake valve is closed prior to the fastest portion of the piston acceleration.

The challenge is then to find a combination of lobe center, overlap and duration that will maximize DCR. Remember that DCR can only be the amount of the stroke from the time the intake valve is closed to TDC.



4 CYCLE Cam Diagram

Suppose we have a cylinder volume of 268.75cc per cylinder (68mm stroke x 74mm bore), with a combustion chamber volume of 21cc. This would provide a CCR of 13.8:1. Now if we use the Alquati 316 degree cam (110 deg L/C, 6.86mm lift at the cam), with an intake closing of 83 ABDC, we can compute the position of the piston above BDC and determine the actual cylinder volume remaining at that point. In this case, at 83 degrees the cylinder volume (VE) is 170cc. So according to the formula "(Cylinder Volume + Combustion Chamber Volume)/Combustion Chamber Volume", this makes for an DCR of 9.2:1. The DCR is always lower than the CCR, it is only important how much lower.

Now if we take the Kent FT6 304 degree camshaft (106 deg. L/C, 7.11mm lift at the cam) with a intake closing of 74 degrees (installed 4 degrees advanced) this results in a VE of 192cc and a DCR of 10.2:1. This is a full compression point higher. In addition the overlap on the Kent camshaft is also 4 degrees less.

Small story continued *- Seeing the above results I could easily see that the Kent camshaft was a much better cam for the A112 motor.*

Most camshafts are installed 2-4 degrees advanced, particularly if they use a cam chain which over time will stretch. This will change the Effective Cylinder Volume and also the Effective Compression Ratio. Below find a spreadsheet of the results of the computations for the various camshafts. Remember that Actual Cylinder Volume is 268.75cc per cylinder.

Manuf	Adv	Lobe	Cam	Intake	Overlap	Open/Close	VE in	CRE

	Duration	Center	lift (mm)	L/C		Adv.	cc	
CatCams	305	108	7.21	108	78	39/79 79/39	182.4	9.68
PBS A8	305	108	6.98	108				
PBS A6	292	106	6.85	104	72	40/72 80/32	196.88	10.37
SLR286	286	106	8.43	106	74	37/69 69/37	202.73	10.65
SLR300-106	300	106	8.43	104	88	46/74 78/42	192.86	10.18
SLR300-110	300	110	8.43	105	80	45//75 85/35	190.82	10.09
Kent FT6	304	106	7.11	102	92	50/74 82/42	192.86	10.18
Alquati	316	110	6.85	108	96	48/88 88/48	162.31	8.73
Abarth	316/304	105	7.21	102	100	53/83 77/47	173.67	9,27
Abarth	336	105	7.72	105	120	60/96 96/60	143.34	7.83

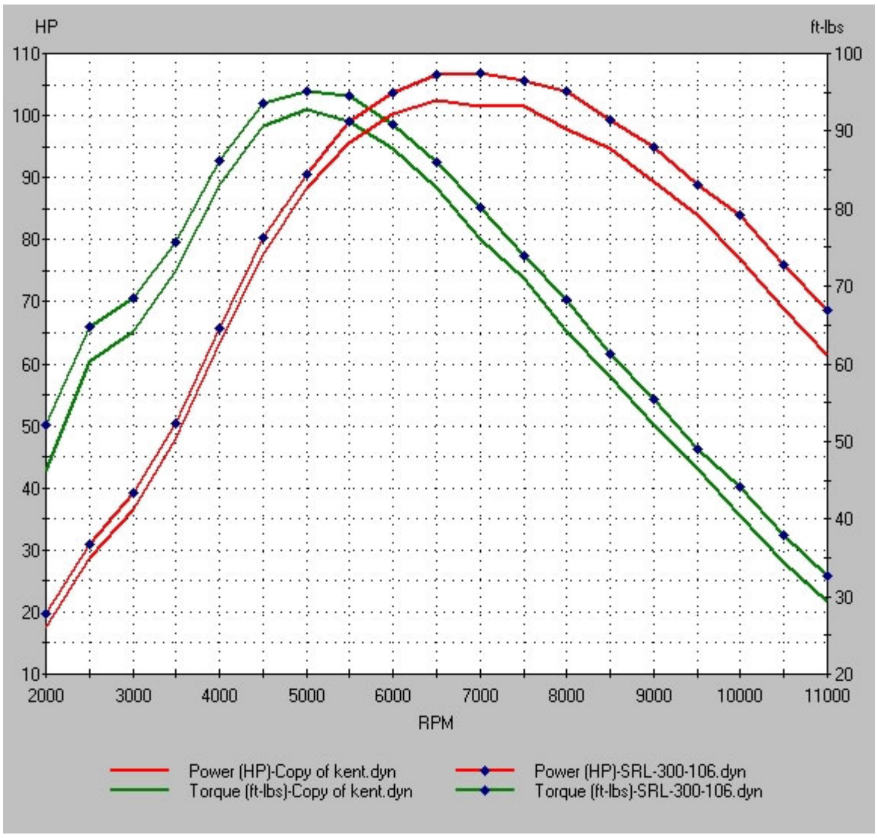
By looking at the above chart it becomes obvious that BIG duration, with small lobe centers, equates to a severe lowering of VE and the directly interrelated DCR. So a compromise must be reached. We can choose a larger lobe center, making sure the power band stays where we want, and also advance the cam to get back the VE and DCR numbers that we want.

This is what has been done with the SLR cams, Kent FT6 and the PBS cam and it appears that the "sweet spot" is between 71.4 and 78 degrees for intake valve closing. This produces a DCR above 10 in all cases. Noting is free however, and DCR numbers over 10:1 may give cause for alarm, as you would definitively have potential for detonation. It may be that slightly lower CCR may have to used, so as to lower the knock index number.

Note: I went back to an article that I had read some years ago about the short lived Pontiac GTO Trams-Am project. Pontiac's project engineer Tom Neil explained how they had gone about determining what their "road-race" engine required in the way of a camshaft. At the end of the day they also determined that the "secret" lay in the closing times of the first the intake valve and secondarily the exhaust valve. As it turned out they settled for a 300/310 camgrind on 105 centers with ,500 inch (12.5mm) lift. Intake closing was slightly different, because of the short stroke and long connecting rod. However when computed backward, it falls right into the range that I found to be effective for the A112 motor.

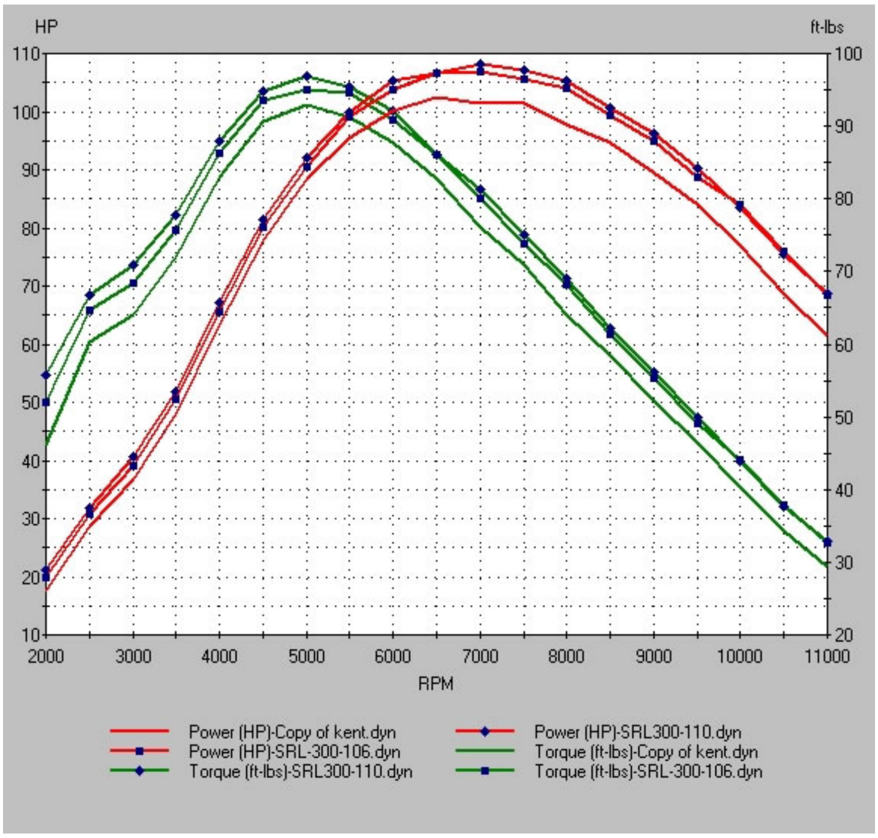
There is one additional difference and that is that the SLR cams all have smaller exhaust/intake overlap and appreciably greater valve lift than the other cams in the list. This means the lobes on these cams will be more aggressive as far as lift per degree of rotation and so exhibit greater lift "earlier", providing a greater area "under the curve". I used one of my engine design packages to illustrate this in graphical form, as in number form it becomes too cumbersome.

First, let's compare the two most directly comparable camshafts on the list for which I have data. This would be the Kent FT6 and the SLR300-106. These are 304 and 300 degrees respectively (both on 106 L/C), with the Kent FT6 winning out in duration and the SLR on valve lift and less overlap. Lets see how they compare.



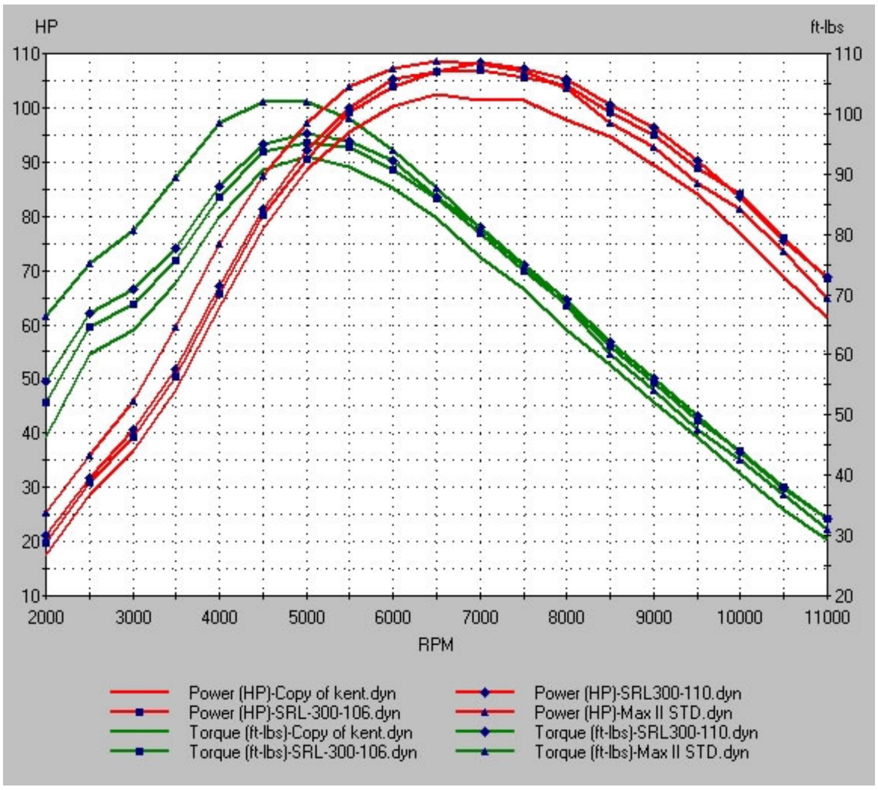
Interesting !! Here you can see that cams with the same VE and DCR, can have different power curves. Obviously the higher, and inherently earlier, cam lift has increased the area under the curve for the SLR300-106 cam. The SLR300-106 cam also has 4 degrees less overlap. Both the horsepower and torque show an reasonable increase, about 5 HP and 2 lb/ft of torque.

What would happen if we spread the lobes apart to 110 degrees, thereby reducing the overlap by another 8 degrees? Would this reduction in overlap, theoretically reducing pumping losses, have beneficial effects?



From the results you would have to say that the SLR300-110 cam does appear to do better. While the horsepower increase is relatively small (maybe 1-2), it has moved the horsepower peak higher in RPM, and there is also a noticable increase in peak torque, even though the VE and DCR numbers are marginally less than the SLR300-106.

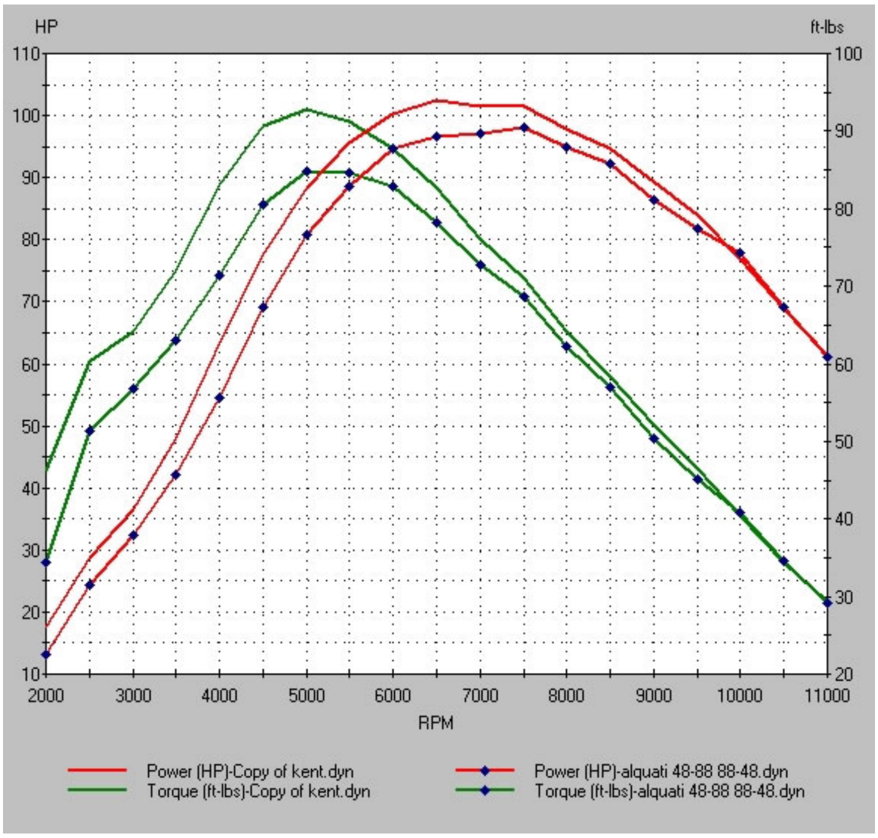
What about the SLR 286 cam (also known as the Max II). It had higher VE and DCR numbers than the other two SLR300 cams and the Kent FT6, yet it is another 4 degrees less in duration. How does it compare?



Did you expect this result? Let's see if we can judge why. The torque is considerably higher, based on having the lowest overlap of any of the cams we tested. Likewise, because of its shorter duration, it does achieve peak horsepower as early as 6500 RPM, and by 8500 has dropped off considerably.

As far as the original design criteria of a cam that had peak torque between 5000 and 6000 and carried power to between 8000-8500, this may not be the best cam, although it is what I used in 2001 when I ran in the Coppa MIlle. On the other hand if you were doing fast slaloms or perhaps even hill climbs, where good torque response at lower RPMs are required, then the SLR286 would get my vote.

Small Story Final Episode - *Just to finish the story, let compare the Kent FT6 cam that Bram would regularly run with the 316 degree Alquati camshaft that he decided to try at Hockenheim.*



Well as you can see the Alquati cam comes up well short of the original Kent FT6 and all of the SLR designs would probably outperform the Alquati as well.

So to summarize, it is probably more important to maximize the "area under the cam curve", by increasing both the aggressiveness and lift of the cam lobe and making sure that the valve closing is somewhere in the range of 65-78 degrees ABDC. Likewise, using slightly larger lobe seperation (108-110 degrees) will also reduce overlap and minimize possible pumping losses.

4.3 Optimizing Valve Train Reliability

1. **Valve Spring Forces** - The answer is both simple and complex. The simple one is "enough to keep the valve from floating or bouncing off the seat". The more complex answer takes into account the weight of various components and the aggressiveness of the opening and closing ramps.

   In order to answer this fully you would have to run the engine on a Spintron machine. Then with a high speed camera and a strobe you could isolate each of the movements of the camshaft action and the impact on the valve and spring. Since almost none of us have access to this type of equipment, we have to make some educated assumptions. These are that we need sufficient spring pressure to make sure that the lifter accurately follows the cam lobe and that the valve is not lofted off the lobe. *(There are some cam designs where the valve is PURPOSELY lofted off the valve lobe in order to achieve higher opening lifts, however this may have other consequential effects which under normal circumstances would be catastrophic.)*

   We already know that with medium lift camshafts (6-7mm at the lobe) that standard valve springs are quite adequate. Abarth made some springs that dealt with cam lobe lifts of 7-7.4mm. Finally, with aggressive camshafts with 7.5-8.2mm lift we need springs with both more preload, and with more free travel. The more aggressive cams will have valve openings between 11.5 and 12.3mm.

   I have seen some valve installations where the seat pressure was less than 30 lbs and the nose pressure only 120 lbs. The Abarth springs are rated at about 45 lbs seat pressure with about 160 lbs across the nose. For very aggressive camshafts seat pressures go up to about 60-70 lbs and nose pressures may be in excess of 230 lbs. This clearly illustrates that the rocker/rocker arm lubrication boundary interface is being stressed much harder with aggressive camshafts and springs.

   We can do things to minimize the spring pressure however. First and foremost would be to reduce the weight of those items on the valve side of the rocker arm. This includes the valve, retainer, spring and the rocker arm itself. In a secondary fashion the valve spring is also responsible for controlling the lifter contact with the camshaft. It is in this area where there may be significant gains that can be made.

   If we were to make a spring holder that would sit on top of the lifter, with an orifice for the push rod to go through, we could put a compression spring there to control the action of the lifter, and a portion of the push rod. The other end of the lifter spring would be held captive by a notched plate on the underside of the cylinder head, and positioned in the lifter galley. *In "hot-rod" circles this would be referred to as a "rev-kit".* This would relieve valve spring of this responsibility and the spring tension of the valve spring could be reduced. This would also mean reduced pressure on the adjuster/pushrod and rocker/rocker shaft interfaces, thus reducing the parasitic losses associated with these interfaces.

   The resultant redistribution of spring pressures would either allow more RPMs before valve float would occur, or would allow lighter valve springs to be used with the current limit on RPMs being observed.

   My current test have indicated that if a spring is mounted on the lifter, then the valve spring pressure could be reduced by as much as 25% or more.

   As with most things in life, there are really no "free lunches", and the same goes for this idea. It is true that the redistribution of spring pressures would have a beneficial effect on the rocker arm/shaft boundary interface. However there are some trade-offs.

   1. 1. The spring in the lifter idea will add three extra components. All of these items will add to the weight of the total valve train.
   2. 2. The addition of a third spring will also add a third harmonic element to the valve train. So we have a dampened pair of springs on the valve and an undampened spring on the lifter.
   3. 3. The chilled iron lifter will now be constantly spring loaded on the cam lobe. Principally this will have an effect on the heel, or base circle, element of the lobe. This means that additional lubrication may be required to deal with this. As the standard lifter tends to collect oil, it would be possible EDM a small hole in the bottom of the lifter to provide a "drip" oiling system for the cam lobe. It is not known if the metallurgical structure of the chilled iron lifter would take well to any "interruption" in the lifter foot surface, no matter how small. It could be that this "defect" could cause be the beginning of catastrophic lifter failure, and also perhaps camshaft failure.

   Having looked at all of the "pros and cons", I have come to the conclusion that this is not an avenue that is worthwhile pursuing, at least at this time, simply because the attendant risks outweigh the possible rewards.

   However, what it did reinforce was that a better understanding of the spring forces required for the proper operation of a camshaft is required. To that end I went back to one of my engineering programs that allows me to model for valve train dynamics. This involves recording the lift of the camshaft at 1 degree intervals (Ideally you would want to record the data at much smaller increments with a Cam Doctor or similar device, but for this exercise this level of accuracy will be sufficient) and recording this information in a "camshaft file". This information, along with the weight of each individual component in the valve train (valve, spring, retainer, collets, lash cap, rocker arm, push rod and lifter) can then be modeled to determine what the minimum seat pressure and full lift pressure that are required to have accurate tracking of the lifter to the cam lobe.

   At this point I have to again indicate that all of this came about because of a lubrication problem at the rocker arm/shaft boundary interface, but it also important in terms of the overall performance of the engine.

   For examination purposes I used our SLR300 camshaft as our test sample, principally because this is the camshaft where the rocker arm/shaft lubrication problem first emerged and it would appear to be linked to the increased spring pressures exerted by the uprated springs, supplied by Scuderia Topolino, for use with the SLR 300 camshaft. Remember, the primary valve spring criteria is that the cam must not float the valves within the operating range of the engine.

   So just how much spring pressure is required to control the valve action with a SRL300 camshaft, yet NOT overstress the rocker arm/shaft boundary interface? I decided to draw on some empirical data, provided by customers, and to model all of this to try an determine an answer. To this end II had to set some standard weights for the various valve system components. These are within one gram +/-.

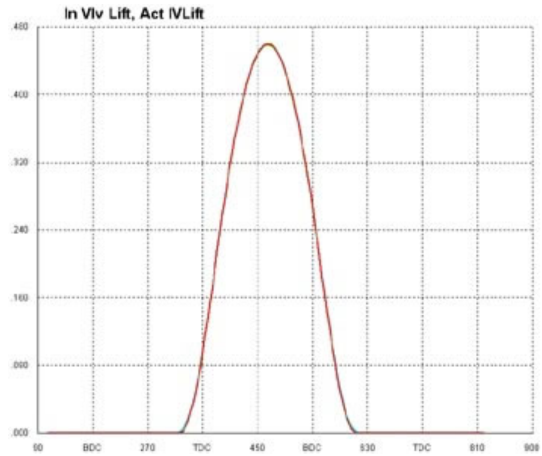
Push Rod 48gm, Lifter 29gm, Intake valve 44 gm, Exhaust valve 42gm, Titanium retainer and collets 8 gm, Spring 50gm, Lash cap 3gm, Rocker arm 52 gm.

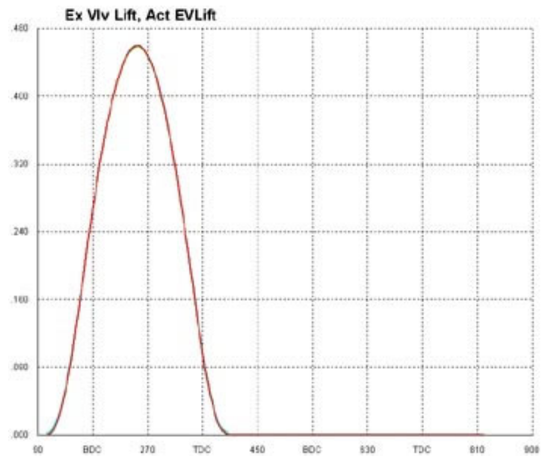
One of my customers (Customer A) used the SRL300 camshaft with a set of the valve springs we supplied, rated at 90 lb (41 Kg) seat pressure and 245 lb (111 Kg) full lift pressure. This engine suffered from a rocker arm/shaft lubrication problem, with the far rear rocker seizing to the shaft. Without knowing the condition of the rocker and shaft in terms of wear and clearance when the engine was assembled, it is difficult to estimate just how much the limits had been exceeded. This is of course always a problem with high performance camshafts, as the cam designer has no knowledge of the condition of the remainder of the components that must work together to make an effective camshaft installation. Obviously, this amount of pressure, given the Fiat lubrication system, was too much and the rocker arm failed.

Another customer (Customer B) used the SRL300 camshaft with a set of Schrick springs rated at 50 lb (22.7 Kg) seat pressure and 151 lb (68.6 Kg) full lift pressure. This engine did not suffer from a rocker arm/shaft lubrication problem. From this empirical result it would be safe to assume that if the rocker arm/shaft boundary interface were to factory clearance specifications, that a rocker arm failure would not occur. *

Obviously, the answer must lie somewhere in between.

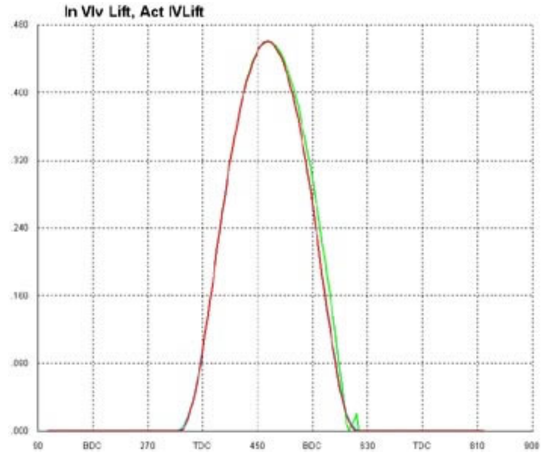
I modeled both the intake and exhaust valve valve lift and actual dynamic valve lift characteristics for Customer A first. This was a SLR300 camshaft with the Scuderia Topolino supplied valve springs. Test RPM was 7500 RPM.

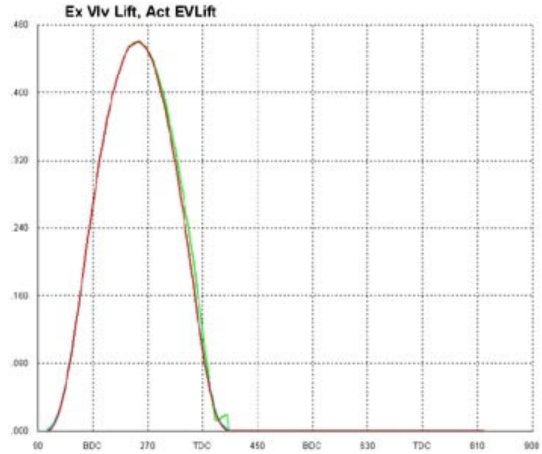




As you can see in both graphs the two traces for (red and green) are perfectly superimposed. This would seem to indicate that the valve spring supplied by Scuderia Topolinbo has sufficient pressure to control valve. **What it does not say is whether the spring pressure could be less, and still do an adequate job of controlling the valve train motion.**

Next I modeled the same SLR300 camshaft, as used by Customer B, using the spring specifications for the Schrick valve spring that he used, again at the same test RPM of 7500 RPM.

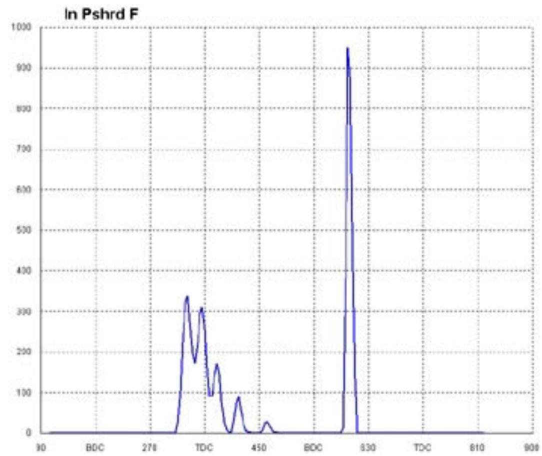


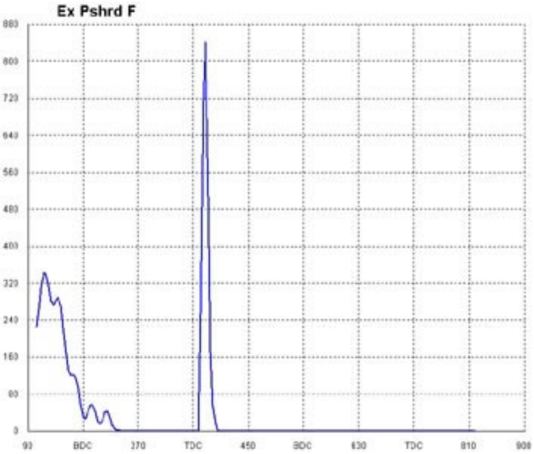


Here the result is quite the opposite. The two traces are easily distinguished. Both the intake and exhaust valve are being "lofted", indicating that the lifter is not properly tracking the cam lobe. This is not necessarily "valve float" in the traditional sense when the spring goes into a harmonic condition, but it may in turn cause some unwanted spring oscillation to occur. Obviously this is spring is not sufficient to control the valve train motion of the SLR300 camshaft. The lifter is for all intents off the closing side of the lobe altogether and comes back down with the lobe at less than 0.050 lift. You can also see that it is predicted that the valve will bounce when it does come down with considerable destructive force.

** The engine was assembled by a knowledgeable mechanic. The cam was degreed in and found to be installed to specification, with minimum valve-to-piston clearance of 0.100 inch (2.5mm). He did set the valve clearance at 0.008/0.010 inch (0.2-0.25 mm) instead of the recommended 0.020/0.022 inch (0.5-0.55mm), as he thought the clearance was too great and "could not be right". The closer clearance of course has two effects. 1) It effectivecly increases the duration of the camshaft about 20 degrees. 2) It compromises the effectiveness of the "lobe clearance take up ramps". 3) This combined with the lower pressure Schrick valve springs led to the lifter crashing on the end of the closing ramp and bouncing off the seat.*

This engine did lose compression in 3 of 4 cylinders after about 15 minutes of running. What can we deduce from the above information? First it is obvious that both the intake and exhaust valve are not in contact with the cam lobe on the closing ramp, and totally overshooting the clearance take-up ramp on the camshaft. This would cause a very high spike in pushrod pressures. (Note: On engine disassembly the camshaft had at least one damaged cam lobe) See the following graphs.





These graphs illustrate the effect of valve "lofting". The closing ramp forces are more than three times higher than the opening ramp forces. This camshaft was going to fail, it was only a matter of time.

Valve lofting has other unintended consequences. First, the inertia contained in valve lofting could cause the associated valve spring to go into coil bind, because it does not have sufficient tension. This will place severe stresses on the retainer and collets. Further, if the amount of lofting is in excess of the assembled piston-to-valve clearance, then almost certainly the exhaust valve will come into contact with the piston. From the spring technical data it appears that the spring can compress an additional 3mm before it is in coil bind. If the valve is lofted into spring bind (.120 inch [3mm] more than the actual lift of the cam lobe), then there would be interference of about 0.020 inch [0.5mm]) between the piston and valve. This situation is more acute for exhaust valve, due to the motion of the valve relative to the piston and would be worse if the cam were installed advanced 3-4 degrees to accommodate any chain stretch. The test data suggests that the valve lofting will occur between 7500 - 8000 RPM if the Schrick spring is used with the SLR300 camshaft.

Conclusion - If the engine was assembled with a valve-to-piston clearance of 0.100 inch (2.5mm), then certainly at 8000 RPM their is a very good likelihood that the pistons will clash with the exhaust valves. The problems associated with the engine of Customer B seem to bear this out.

Note: Under no circumstances would I imply that there is physically anything wrong with the Schrick valve spring. This company makes very good products. It simply means that it is not the correct valve spring for a SLR300 camshaft. In fact, in other tests that I conducted with different camshafts with less duration, there appeared not to be a problem at all with this spring. The margin of difference is very small.

So what is the answer.

After further modeling, if a valve spring with a rate of 220 lb/inch (3.9 Kg/mm) were used, then lofting is not longer an issue. Please note that there is a difference between the â€œspring rateâ€? and the actual seat spring pressure and the over the nose pressure. As a matter of safety, I would probably opt for a spring with a rate between 250-260 lb/inch (4.47-4.65 Kg/mm) to provide a little extra safety margin. This valve spring would have 52 lbs (23.6 Kg) of seat pressure (at 1.250inch [34.29mm] installed height) and 172 lbs (78 Kg) of pressure at 12.18mm valve lift.

This spring combination would be a 24% reduction in valve spring force, as compared to the spring supplied by Scuderia Topolino, and recommended by our cam grinder. This is not the complete story however. As the rocker arm ratio is 1.45:1 the reduction in pressure on the valve adjuster is close to 35%.

It goes with out saying that a reduction in spring pressure, as well as the attendant reduction in other parasitic friction losses associated with using a spring with less pressure, will mean an incremental increase in horsepower. Between 7500 and 8000 RPM this may very well mean 2-3 more horsepower is available to drive the wheels.

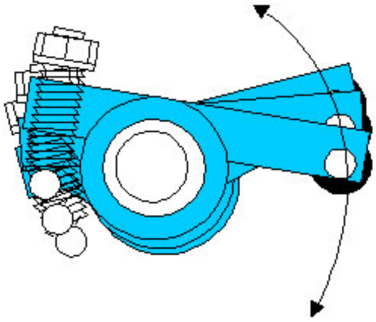
2. **Rocker Arm Geometry** â€“ One of the misunderstood and mostly forgotten functions of the push rod is to adjust rocker arm geometry. After the cam has been reground, the head and block decked, new valves installed in the head, why would you assume the standard push rods are still the correct length? After all regrinding the cam lowers the base circle of the cam, thus lowering the position of the push rod to the rocker arm. Now, if you also machined the head 0.040 inch (1mm), and perhaps the block also 0.020 (0.5mm), then you will have compensated for some of the material ground from the base circle of the cam. The point is, you never know how much of each, so when the engine is first test assembled is when you find this out.

   As a good rule of thumb, when the cam is at 33% of its total lift (so for the SRL300 that would be 0.110 inch [2.8mm]) the adjuster should be in a straight line with the pushrod, with the adjuster showing no more than three threads below the rocker arm. So if 5 threads are showing, and each thread is .8mm, then you would require a push rod that is approx. 1.6mm longer than the test push rod. If however the cup of the push rod is right up against the rocker arm, then you will need a push rod that is 2.5mm shorter than the one you are using for the test.

   Note: The easiest way to measure the effective length of a push rod is to put a small ball bearing in the cup and then measure the overall length with the ball bearing. Now measure the diameter of the ball bearing and subtract this amount to get the effective length.

   You are not finished however. Next you must look at the rocker arm pad where it contacts the valve stem. Rocker arm geometry is generally optimal when the travel or movement of the rocker arm tip on the valve stem is minimized. To understand how to achieve correct geometry, it must be understood that the rocker arm tip itself travels in an arc. At zero lift, the rocker arm tip is expected to be closer (or inboard) to the plane of the pivot point and as the valve starts moving down, the rocker arm tip starts moving outboard. If the geometry is close to ideal, then the rocker tip will be at its most outboard position at half or mid lift at which point the rocker tip starts moving inboard again as the valve reaches full lift. Simply put, ideal rocker arm geometry is achieved when the rocker tip is sitting on the valve stem tip at the same position at both zero lift and full lift.

In a perfect world, where the rocker shaft pedestal stand locations, the valve guide, and the rocker itself are all machined to exact specifications, the rocker tip is expected to be sitting slightly inboard of the valve stem center at both zero and full lift while the rocker tip will be sitting the same distance outboard of the center of the valve stem at exactly mid-lift. As this is not a perfect world, this sometimes does not happen.



It may be that the valve is installed at a slightly lower installed height. If so a lash cap may be required. Alternatively, if the rocker pad does not sit in the correct location on the valve stem and the rocker fulcrum point has to be moved closer to the center of the stem, and so a shim will have to go under the rocker arms stands. If the opposite is true then the rocker stands may have to be reduced in height a small amount.

All of these actions, will affect the length of the push rod required. Take the trouble to do it, as it is worth the effort to get the geometry correct.

Note: The condition of having the contact pad too far extended over the valve stem, is the cause for the little half moon wear marks, that I discussed earlier. This causes the rocker arm to extend â€œoverâ€? the valve stem at full lift, and instead of depressing the valve stem it pulls the valve stem sideways toward the rocker arm fulcrum. This increases parasitic friction losses and causes premature valve guide wear.

SCUDERIA TOPOLINO when only the best will do

[Technical Advices](#) > Cylinder Heads

Cylinder Heads

5.1 Types of cylinder heads

There are both OEM type and aftermarket cylinder heads that can be used in combination with the various types of Fiat/A112 blocks. The head bolt pattern is the same, with the exception that some models used 10 x 1.25mm head bolts, while the majority used 9 x 1.25mm head bolts. It should be noted that Fiat is probably the ONLY car company to use 9 x 1.25mm hardware extensively.

OEM Heads

Fiat 600 and 600D - The early Fiat 600 head is distinquished by 8mm rocker stand studs and 6mm rocker arm adjusting screws. The later 600D heads had 10mm rocker arm stand studs and 7mm adjusters, as the early 6mm ones did not stand up well to the rigors of competition.

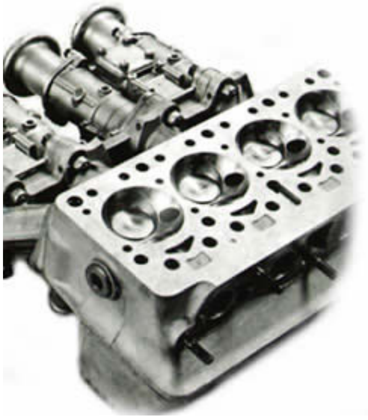
Photo – 600D head combustion chamber

Both 600 and 600D heads were used for most of the early Abarth coachbuilt automobiles. The head featured a deep "bathtub" type combustion chamber that required a piston with a "kidney" shaped dome on the piston to get the compression up. The valve sizes that would fit into the head were also limited, although at the time this was dictated by FIA rules and homologation papers. As engine displacements grew to 982cc, as for the 1000TC motor, this proved a real challenge.

Fiat 850 - The Fiat 850 head configuration was quite different. Still using a 2-valve side-by-side arrangement, the chamber was now much more open, with a good squish area to increase the combustion chamber turbulence as the piston came up to TDC. This same chamber configuration continued on and was eventually used for the A112 motor, with only minor modifications around the intake valve to assist flow.

850/A112 Combustion Chamber – Note that the chamber is not yet completely finished.

Abarth TCR and OTR - While considered and OEM head, the TCR and OTR heads did not look anything like the Fiat 600/600D or 850 heads. While the head bolt pattern is still the same, that is about where the similarity ends.



The general design of the head is a scaled down version of the head designed by Aurelia Lampredi for the Fiat 2300 saloon, and later adopted for the Abarth 2400 couple. It is still push rod operated, but with a hemispherical combustion chamber. Valves are operated by rocker arms, however intake and exhaust valves have separate rocker arm shafts. The TCR/OTR head is usually equipped with either dual Solex or Weber carburetors. The exhaust manifold is also different, as the port spacing and location does not coincide with either the Fiat 600/600D or Fiat 850 heads.

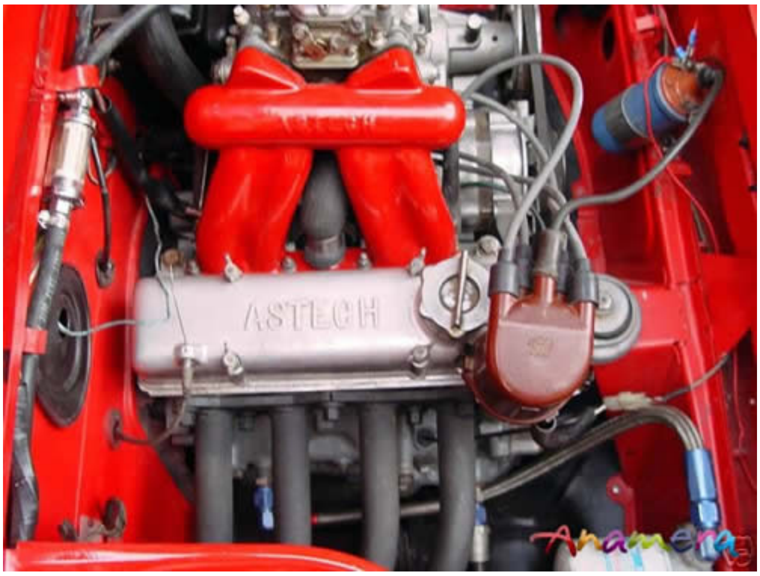
Aftermarket Heads

Vizza Motorsports

Vizza Motorsports Cylinder head

This head is a modified Fiat or Autobianchi standard head, and is sold by several people including Vizza Motorsports. The head has been welded up and then individual ports machined for each cylinder. As the ports enter the top of the head, the intake has to make a rather abrupt turn so that the dual Weber carburetors sit at the correct angle. I have also seen this head with two 40mm DCNF downdraft carburetors, and in this case the entry into the head is pretty much straight down. I have not had any experience with this head, but I would imagine the downdraft version would be more effective that the side draft version.

Astech

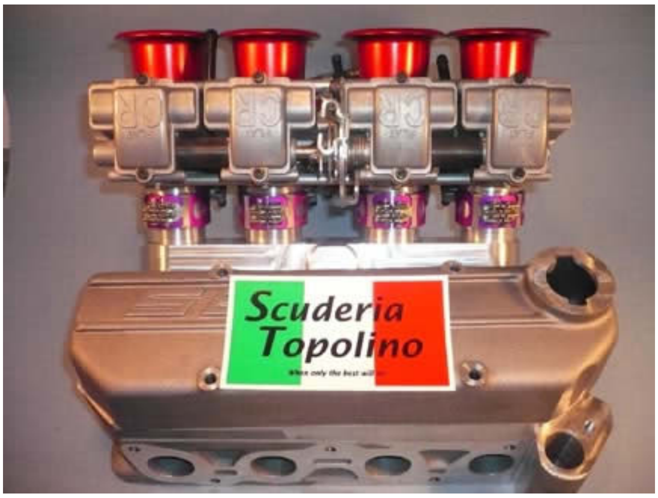


Astech Cylinder Head

This head was designed in the USA and produced for a number of years. It uses a log-runner intake manifold design for a single 40 or 45 Weber DCOE carburetor. The head is no longer in production. Given the design of the intake manifold, this type of installation would provide good torque and horsepower in the range from 4000-7000 RPM.

PBS 8P

Designed by Paul Swenson, this head has been in constant production since 1968. There was one revision of the casting models in the 80s to slightly change the head so that it would work on the A112 blocks,which had a different bore spacing. Scuderia Topolino is pleased to carry on the production of this head. The chief characteristics of this head are improved squish area, a "high angle intake port and revised cooling layout.



Here are two implementations of the 8P head. The first is a more conventional version with twin 40DCOE Weber carburetors. The second is a racing implementation using 4 Keihin FCR single barrel carburetors. In this case the carburetors sit at a 50 degree angle and the approach to the intake valve is very direct. Even with the Weber carburetors the curvature of the intake runner is very gradual, which is the reason for its good performance.

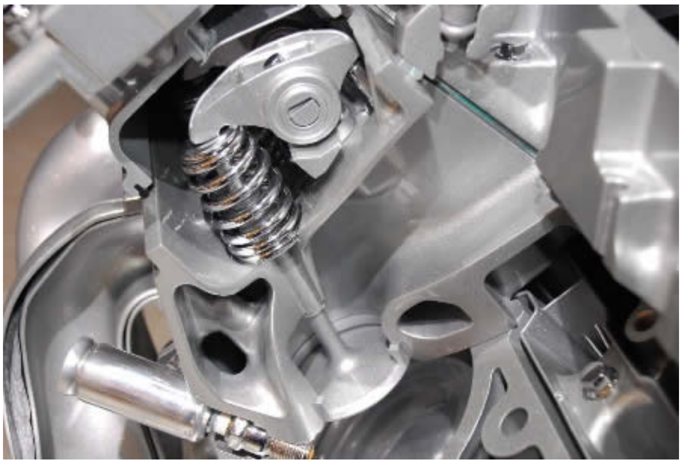
5.2 Head Layout and Modification

Valve size - It is possible to put larger valves in 850, A112, PBS and Vizza heads. I am not sure about the Astech head, as I have not worked on one. The maximum valve size is 32mm intake and 28mm exhaust. Using these sizes does pose some interesting problems. First, the intake and exhaust seats must be "nested" in one another (the exhaust is cut into the intake) and this can significantly weaken the seat area of the head. Second, by using such a large intake valve, the edge of the valve is shrouded by the edge of the combustion chamber and does not unshroud itself until the valves is nearly completely open. This causes a disruption in the flow of the valve. Third, using valve this large means that they "nearly" touch. If anything goes remotely out of sync, the valves could end up touching and this could be disastrous. The only way to rectify this is to move both the intake and the exhaust valve in the head. This would mean installing off-center valve guides, welding up the seat area, and cutting new seat pockets. This is a large, and exacting, job and not for the faint of heart. At Scuderia Topolino we consider 31 and 27mm valves for intake and exhaust respectively, to be the maximum size that we feel comfortable installing. Even then we go to some extra effort, on fully race ported heads, to take the combustion chamber wall out as far as possible. The shrouding of the valve is thereby kept to a minimum.

Intake port short side radius – Probably the most important area of the any of the heads, more so for the Fiat/A112 heads than the PBS 8P, is how you treat the "short side radius". As the intake charge enters the port, and subsequently flows around the valve head, the general idea is to induce a high helix swirl pattern. In fact there are probably several of these patterns being generated.

All too often we get carried away with making the hole in the head bigger without considering the short side radius. You want to leave the radius as large as possible, and make it smooth while removing the minimum of material. It is already a very short radius, so the last thing that you wish to do is lower the floor of the common chamber that feeds all four inlet valves. It would take a great deal of work, but it could be that there is advantage to be gained from actually raising the floor of the chamber slightly to that a larger short side radius is maintained.

Here is a good cross-section of a typical intake port. The one on the right has a larger physical port, but the one on the left will actually flow air more efficiently. The secret lies in the port floor and the short side radius. In the port on the left the floor is raised slightly, and the radius approaching the valve is larger and more rounded. The top of the port will still flow more air, but the ratio of flow between the top and the bottom is much smaller.



In the case of the PBS 8-P head, because of the 45 degree angle of the intake runner the short side radius is very well formed and the flow around the valve is almost equal.

The exhaust port on all of the heads, Fiat/A112/PBS is much the same . The exhaust port also has a very abrupt short side radius. However, because we are not simply relying on atmospheric pressure for getting the burnt charge out of the cylinder, the effect is much less pronounced. You still want to make sure that the flow radius is a smooth as possible, without any abrupt changes. For the most part the port at the exhaust manifold face is almost too big. Generally you want an exhaust valve that is between 83-86% of the intake valve. This makes the 27mm exhaust valve just right, as the actual valve seat diameter will be around 26.5mm. The remainder of the port is somewhat rectangular, ending at the exhaust manifold in a round shape that should be slightly smaller than the internal diameter of the primary tubes. This slight step will provide a small amount of anti-reversion and aid in exhaust tuning.

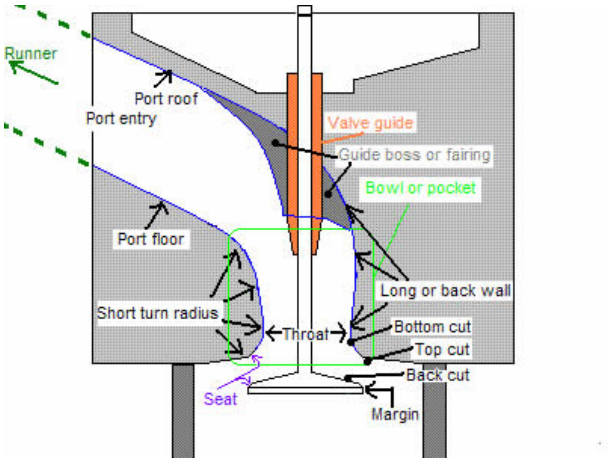
The Bowl – The next most important area is the shape of the bowl. This area can have a large impact on the swirl of the fuel charge as it enters the cylinder. Increased swirl generally means higher efficiency and increase flow. The disturbed characteristic of a swirling air mass also means that the fuel will stay in suspension better for improved combustion efficiency.

The bowl area also includes the portion of the valve guide that protrudes into the intake port. Some head experts like to remove this portion of the guide, while others like to leave a small amount of material there and use it as a small "air director" to get the air to assume a swirl attitude before it passes the valve seat.

The bowl on the exhaust port is no less important and should provide the best possible exit path. Due to cooling considerations, this may not always be optimum.

There is not set formula for how a bowl should look. It is one of those "black art" areas, and only a flow bench will partially answer the question of where to remove material. There are some things that we know work, but the last detail is often just a matter of trial and error.

Head seat configuration – If we can achieve a 83-86% intake/exhaust size ratio, then the next thing to consider is the seat angles in the head. The principal seating surface can be either 45 or 55 degrees.



Above you will find a typical intake port.

I then use a 45-55 degrees seat angle with a bottom cut to blend into the bowl area and a top cut to aid flow into the cylinder. If you go for a 55 degree seat, then you will have to run Beryllium or AMPCO copper seats, or you may have transfer of material from the seat to the valve surface, affecting the seal of the valve. For configurations where the intake valve may be shrouded this higher angle may be of some help, as it assists flow into the cylinder.

For the exhaust port I generally use a 45 degree seat, principally for reliability.

Valve seat and valve head configuration – This is a very complex interplay of components, theory, and practical experience. Each "expert" will have his or her ideas of what works.

The following link is a very good technical explanation of what you should try and achieve in modifying any head. While it is written around typical American production head work, much of it is directly applicable to Fiat heads.

http://www.tmossporting.com

Once you get on to this site, click on the Tech Articles link and go to the last item "Head Porting Principals".

5.3 Valve Lift and Flow

One of the best engine technicians was Smokey Yunick of stock car racing fame. He concluded from air flow observations that he made that to open a valve more than 33% of the valve diameter may not be productive due to the additional parasitic loss generated. Therefore for a 31mm valve, the maximum valve lift should be 10.9mm.

I know from my own tests on the flow bench that the "incremental" gains in air flow over 11mm of lift are indeed small. However the real advantage to higher lifts lies in the fact that in order to achieve these lifts a more aggressive cam grind must be employed. In doing so we increase the "area under the lift curve" in general, and because the lobes on the can are more aggressive, we increase the "rate of lift". This improves low lift flow.

Our biggest camshafts have 12.4mm of lift at the valve, and as long as low friction components are used in the valve train, and sliding friction losses are reduced to a minimum, additional benefit will result from the increased gross valve lift and rate of lift.

5.4 Head Gaskets

There are all types of head gaskets available and each has properties that are preferred for specific applications, but all depend on having <u>an absolutely flat, clean and smooth head and block surface</u>. This means a flat surface, at 90 degrees to the bore, with an RA (Average Surface Roughness) of between 14 and 20.

If you have a road car, with 10:1 compression, flat top pistons and a road camshaft, then any standard head gasket will be sufficient for your needs, PROVIDING you use the appropriate grade of fuel to prevent detonation. *Note: You will read a great deal more about the subject of detonation in the technical sections and the FAQ section, as it is a little understood, and often neglected, area of engine tuning.*

One step up would be to use the Spesso Competition head gasket. This is 1.6mm thick (uncompressed) and compresses to approx 1.2mm (0.048 inch). These head gaskets have special silicone beads around certain oil and water areas for improved sealing. In addition the "fire rings" are general made of stainless steel.

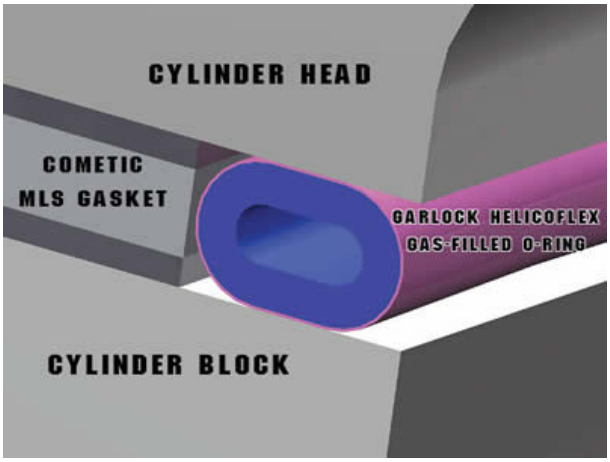
If you have a competition engine up to 12.5:1 compression, with a more aggressive camshaft with higher "dynamic compression" then you should consider the Spesso Competition or a Scuderia Topolino solid copper head gasket. The copper head gasket is 0.043 inch (1.09mm) thick. Care should be taken that the copper head gasket is properly softened (annealed) and coated with sealant on both sides before used.

The next step up is to use an Multi-Layer-Steel gasket, such as the one made for Scuderia Topolino by Cometic.

This gasket consists of a minimum of upper and lower stainless steel layers with a number of intermediate layers. In this way the gasket can be varied according the squish area requirements of the engine. The standard thickness for this gasket is 0.035 inch (0.9mm). The upper and lower surfaces are coated with a rubber material and not additional coatings or sealers are required.

Finally, if you have a very high compression motor (13.5:1 computed compression) with a very aggressive camshaft and high dynamic compression (10:1 or more), then all of the above options may eventually fail. In this instance you have an additional option.

The ultimate head gasket is a combination of two gaskets. It consists of a special MLS gasket with a separate 0.031 outside diameter stainless steel, pressurized o-ring, added as a pressure sealant, to contain the combustion pressures.



The "Helicoflex" o-ring is hermetically sealed during manufacture and has an internal gas pressure to maintain a positive contact against the block and head. As the ring heats up the pressure inside the ring increased, resulting in a higher pressure seal. The diagram above shows the combination of the two gaskets.

This combination of head gasket materials gives the best possible combination. The MLS head gasket acts as both a media sealer for oil and water, and a peripheral stop for the gas filled o-ring. The pressure filled o-ring acts as a super efficient pressure barrier for the combustion process. Another company that implemented this pressure ring technology is Coopers, who provided this technology for the Cosworth turbo-charged, FI motors.

If the piston is installed with "0" deck height, then the squish distance would only be 0.035 inch (.98mm). Given that connecting rods so stretch up to about 0.015 inch (.38mm), this would leave only 0.023 (.6mm) piston to head distance at high RPM. This would be the minimum piston to head distance for a 1050 motor, that I would recommend.

Postscript - Below are some photos of a head gasket that was destroyed by detonation. The detonation was caused by trying to run an engine with high dynamic compression(9.8:1) on 100 octane fuel. It probably would not have made any difference as to what type of head gasket was used, as even with a pressure ring the end result would have either been the same, or the next weakest link (probably the pistons) would have failed.

Three of the gasket fire rings have been deformed into a "teardrop" shape. This is a classic end result of detonation. In detonation the cylinder pressures would have spiked to over 3000 PSI. This amount of pressure would have lifted the head, releasing the clamping force on the head gasket and simply pushing it aside. The only reason the fourth cylinder did not suffer the same fate was that the car had already stopped running.

SCUDERIA TOPOLINO when only the best will do

Cooling System

6.1 General Considerations

In any high performance, internal combustion engine one of the principle issues is managing the heat produced by the internal combustion process. This heat can be your friend, however it can also be a major contributor to three very destructive processes: pre-ignition, detonation and alloy temper degradation.

Even very efficient engines are rarely more than 35 % efficient in converting heat to motive energy. The other 65 % is lost either out the exhaust pipe, or has to be dissipated through the cylinder head, engine block and cooling system.

The circulation of coolant through both the cylinder head and engine block cooling passages provides the mechanism for transferring this heat to the radiator system. The cylinder head accounts for the majority of the heat generation and transfer to the coolant, namely 65%, whereas the engine block makes up the remaining 35 per cent. While these are numbers are estimates on my part, they are supported by the various temperature loads that are generated in the head and block respectively and will suffice for illustration purposes.

Within the cylinder head the "hot spots" are the areas directly around the exhaust valve seats and the area around the spark plug seats. These are also the areas that are likely to become super-heated. This super-heating is likely to result in the creation of super heated steam pockets. These steam pockets have three potential negative effects.

- If the steam remains trapped, then it will act as an insulator and this area of the cylinder head or engine block where the steam is located will prevent the transfer of heat to the coolant. This condition is "regenerative", and the steam pocket will eventually enlarge, and an ever-increasing overheating situation will occur.
- Once this steam reaches the water pump then it is possible for the entire cooling system to become ineffective, as modern centrifugal type pumps are not designed to pump steam, only liquid. In systems where the coolant temperature is very high, it is also possible for the water pump itself to boil the coolant on the suction side of the pump. As the pump rotates the "system" pressure is lowered, and thus if the coolant is sufficiently heated, the liquid may boil of its own accord and cause pump cavitation. This would lead to a catastrophic failure. Note: This "pressure drop" situation occurs when you open the radiator cap on a hot motor. *Instant pressure drop equals instant boiling.*
- Aluminum head castings are usually heat treated to T6 hardness specifications. This means that the head casting will have been heated to 1000F initially, then water quenched for a short period of time, and finally left in a heat soak oven at 320F, for a period of approx. 5 hours, and finally allowed to cool naturally to ambient temperature. They key is to make sure that the aluminum head does not exceed this 320F temperature in normal operation, as it will lose it temper. Once this happens then it will be very difficult to maintain proper head gasket clamping forces. In addition pressed-in valve guides and seats, particularly for the exhaust valve, will be compromised.

So the idea is to keep a sense of equilibrium within the engine cooling system, noting that it would be best to keep the average cylinder head temperature well below 300F.

Before going any further, let put to bed a myth *(I will admit that I fell victim to this early in my engine work).*

If the coolant flows through the radiator too quickly, it cannot transfer the heat to the radiator.

> *At a fluid velocity of "X" the coolant will release a given amount of heat to the radiator. If we double the velocity of the flow, it will release less heat to the radiator obviously because the coolant is in contact with the radiator for half the original period of time. Makes sense so far, right??*
>
> *However, since the water now makes* ***two revolutions*** *in the same period of time (remember we doubles the velocity of the flow), the net result is the same. So thermodynamic law says that in the same period of time the same amount of heat is released. <u>This part of the myth is definitely busted</u>*

One reason this rumor persists is that the 2 most often cures actually work sometimes. The first "cure" is to slow the coolant pump rotation with a larger pulley. The problem that is actually solved, in that many stock pumps will cavitate at even moderate RPM. A cavitating pump is very inefficient at best and may stop working altogether, at worst. Therefore slowing the pump the pump actually becomes more efficient. The second part of the "cure", restricting the outlet of the engine (supposedly slowing the flow), actually causes a pressure buildup behind the restrictor thus pressurizes the block by a few pounds more that the overall system pressure. Therefore, the boiling point of the coolant is raised by a small amount. This works to preventing local boiling in stagnant flow areas.

Remember that steam bubbles can slow, or stop, coolant flow through small passages. The idea of a restrictor is actually a worthwhile implementation in a competition motor. The increased pressure also creates some back-pressure and this, in turn, reduces the onset of cavitations. A more optimized solution would be to run the coolant pump at 50% of engine RPM and to place a restrictor (in a production car this would be a thermostat) wherever the coolant flow exits the block. By not overly slowing down the coolant flow we can introduce a level of "turbulence". This will have, as a side benefit, the ability to sweep some of the stagnant areas of any steam accumulation. So I guess that this part of the myth is at least "partially" true. Stewart Racing pumps actually built a coolant pump dynamometer and has demonstrated these effects. There is also a significant body of literature, within the Society of Automotive Engineers, addressing this very problem. I know that some of you reading this are probably saying, "OK, I understand all of this, but how does all of this get me more performance?" Stay with me a little longer, as it will all become clearer.

Next, we need to consider both heat generation and heat release mechanisms (engine and radiator). Let me walk you through a "mental" exercise. Consider the instance where the coolant flow-rate is just such, that the outlet temperature of the radiator is near ambient temperature. In essence, hot coolant (200F) going into the radiator, and ambient temperature (72F) coolant coming out. Not realistic but good for this exercise. This must be MAXIMUM EFFICIENCY for sure. **Wrong**.

Assuming that the coolant loses its heat linearly, the top part of the radiator, where the hot coolant enters the radiator, absorbs almost all of the heat, while the lower part, where the coolant is at, or close to, ambient temperature, the radiator absorbs almost none. The area in between works proportionally.

Similarly, in the engine, where the ambient coolant enters the block, maximum heat is absorbed by the cooling liquid. Where the coolant now exits the engine (at the cylinder head), only limited additional heat can be absorbed. Thus, the cooling ability is non-uniform. Worse, areas in the head that have the highest heat load (such as around the exhaust ports and spark plug areas) may suffer localized boiling. Once a film of steam forms, almost all cooling is lost.

Almost all current production systems utilize what I will refer to as a "bottom-up" approach to the cooling system. Coolant flow is moved by a pump, located on the cylinder block. This takes coolant fluid from the radiator and pumps it into the block first, then through the head gasket into the cylinder head and finally from the head back to the radiator system.

As discussed earlier both the absorption and release of heat is linear. This means that the "bottom" up approach severely compromises the coolant's ability to deal with the more significant areas of heat generation in an engine, namely the cylinder head. As such, most cooling systems are much larger than actually would be required, were the system biased to where the majority of the heat is generated. We only have a given temperature delta (Δ) to work with within a closed system. This is the difference between the input and output temperatures of the coolant as it enters and leaves the radiator.

In terms of proportional heat generation, the block cooling passages account for about 35% of the heat generation, whereas the cylinder head cooling passages account for the other 65%. In a "bottom up" arrangement, by introducing the coolant into the block first, and thereby drawing a significant portion of the block heat into the cooling fluid first, we have already narrowed the temperature Δ available to deal with the much more substantial heat load generated by the cylinder head cooling passages. This will increase the likelihood of steam pocket formation at higher load levels encountered under racing conditions.

It may be that there is significant justification to consider adopting a "top-down", reverse flow, approach to the cooling of the Fiat /A112 competition engine. This idea is far from new, although contemporary developments have lent some new insight. Pontiac, as early as 1956, used a coolant pump to pump water in through the heads first, and then to the block.

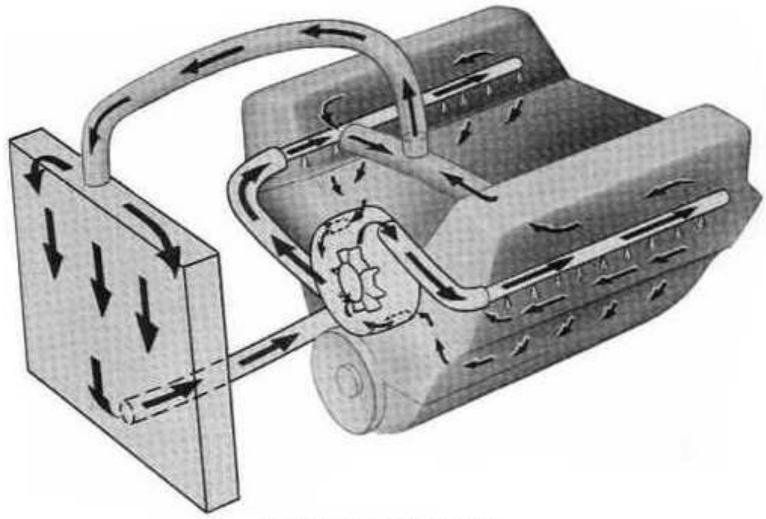


Fig. 6A-1 Engine Cooling System

More recently, John W. Evans filed a patent (5,255,636)) that makes very interesting reading. I may be more than coincidental that Mr. Evans has a pending, large lawsuit against General Motors for using his cooling ideas on the LT1 Chevrolet V8 installed in the latest Corvette. This motor is capable of running at 10.5:1 on unleaded gasoline, with far greater spark advance than normal.

You might ask, "Just how much advantage can be gained by reversing the coolant liquid flow?" In terms of overall cooling performance the percentage change may only be in the high single digits, but there are other "follow-on" benefits that are much more substantial. Since the coolant is now first going to the head (the place where the most heat generation takes place) there is the full coolant temperature Δ available to deal with the hottest part of the motor. Since detonation can take place at temperatures as low as 190F, controlling the areas around the exhaust seat and the spark plug seat provides large dividends in terms of minimizing steam pocket generation, but also allows more distributor advance and compression to be used. Head gasket surfaces are also likely to run at more even temperatures. If the overall temperatures at the cylinder head/gasket interface could be reduced by 10%, this would be a substantial additional safety factor.

A further benefit is derived with regard to the engine block. As the engineers at Chevrolet discovered, because of the relatively even temperature distribution, of the coolant entering the block from the cylinder head, the block was heated much more evenly. Even thought the temperature was slightly elevated, this turned out to be an advantage. In a "bottom-up' cooling system, as low temperature coolant enters the front of the block and has to work its way to the rear cylinders, there is an inherent uneven heat distribution, with the rear cylinders always running hotter than the front ones. While with "top-down" cooling it is uniformly higher and therefore promotes uniform expansion. This assists in better ring sealing and lower piston/cylinder wall friction, all attributes that are important in any competition engine.

In reading Mr. Evans' Patent, it is obvious that some careful thought went into the design of the system, particularly for a road car. Here you have to account for quick warm-up and providing coolant to the cabin heater core. In principle the system is pretty simple. Here are the items that are required.

1. A pump capable of reasonable coolant velocity so as to keep the interior surfaces of the head cooling area and block cooling area well scrubbed of steam bubbles, yet not rotating fast enough to cause serious cavitations.
2. A method of routing any steam bubbles in the top of the head, away from the cylinder head and condensing them back to liquid, to then join the normal fluid flow again. The Patent information is informative in this regard.
3. A restriction on the output of the block, either in the form a thermostat, bi-directional thermostat, or a restrictor plate.
4. A condensing unit.

Items #2 and #4 are THE SECRET to making this work. Anyone familiar with refrigeration, or air conditioning, design will quickly realize what is required. A small line, or a larger line with an internal restriction orifice, must be attached to the top of the A112 head and then routed to the remote header tank in the engine compartment. Is that all there is to it?

Well, yes. You see, as the coolant pump is now pumping into the top of the head, and there is a restrictor on the side of the block (where the current water pump sits) there is a pressure differential between the coolant in the block/head and the coolant in the remainder of the system, including the header tank. By forcing the steam/coolant through a small orifice, it will be subject to a pressure drop and the surface area of the line, as well as the overall surface area of the header tank to which the other end of the line is attached, is sufficient to cool the steam and cause it to condense to a liquid state. If there was any doubt as to this you could always make a header tank that incorporated a finned heat sink. This is the same principle that takes place in a modern air conditioning or refrigeration system where gas is converted back to liquid, after carrying away heat.

Another option-

Another way to deal with the same problem would be to use a "dry deck" configuration. In this implementation a head gasket would be used that had NO water passages. Water would be channeled into and out of the head first and then routed to the block. This may also be a viable solution as no steam venting/condensing port for the head is required, although one could be implemented as a safety measure. Overall the amount of heat load to the radiator system would be the same as a conventional cooling system. This type of system would likely need an electric pump, remotely located from the engine.

I will not go into how I would implement these systems in detail, but I believe they hold much promise in being able to run higher dynamic compression ratios, and greater ignition advance, without the onset of detonation.

Acknowledgements:

John W. Evans – US Patents 5.255,636 and 4.550,694

John De Armond – Hot-Rod Magazine

6.2 Radiators

In modern vehicles radiators are most oftem made of either all aluminium or a combination of an aluminium cores with plastic end tanks. For competition purposes I do not recommend the second type, as the stress placed on them by racing may cause a premature failure.

For competition purposes radiators can be made from either aluminium or brass. The heat dissipation qualities of the two metals is similar, with of course the aluminium radiator weighing less.

Careful attention should be paid to the number of fins on the core tubes, and the overall thickness of the radiator. If the fin spacing is too close, or the core is too thick (or perhaps both), then insufficient air will travel through the radiator core. Radiator efficiency depends on air traveling through the core.

If you are going to run a front and rear radiator, then the flow should be as follows:

Engine thermostat housing to top tank of rear radiator
Bottom tank of rear radiator to top left of front radiator
Bottom right of front radiator to input to water pump
Output of water pump to input of engine.

If you are only using a front radiator on a rear engined car, then the following would apply:

Engine thermostat to top of rear mounted expansion tank
Bottom of expansion tank to top left of front radiator
Bottom right of front radiator to input to water pump
Output of water pump to input to engine

In both cases there should be a small, manual valve installed on the top left tank of the front radiator to allow for venting of any trapped air during system filling. As an alternative, a small tube could be run from the top left tank of the front radiator to either the top tank of the rear radiator or the top half of the rear mounted expansion tank. In this way and air or steam bubbles trapped in the front radiator will automatically be dealt with.

6.2.1 Air Flow – Radiators require air flow. The more flow the greater the efficiency of the radiator. This also means that all the air "caught" by the radiator should be made to go through the radiator, rather than around it. Air will always take the course of least resistance, so if there is a gap around the radiator, it will always flow around it rather than through it.

The radiator should be shrouded, to direct the air through the core. This hold true for systems where the movement of the car forces air through the radiator, as well as systems where the air flow is wholly dependent on an air conveyor as in the Fiat 600 and 850 type cars. Air conveyor systems do depend on engine RPM, and therefore may not be able to handle extreme competition temperatures as well.

6.2.2 Radiator Sealing – Most modern radiators have a radiator cap, and this includes Abarths and Fiats. What is generally not known is that the radiator neck on Fiat cars is different to most other cars. It is slightly deeper. Therefore the use of an aftermarket cap, even if it says 22 lbs on it, will generally only result in a 3-4 lb cap. So, if you are using a standard Fiat radiator, use a standard Fiat cap. It will at least give you 12-14 lbs of system pressure.

The other alternative is to have the radiator neck replaces with a standard aftermarket one. Now you can run any high performance radiator cap you want.

Here is an illustration of several stant competition caps that range in pressure from 19-22 lbs. While on the subject of radiators, all hose connections should have a raised lip on the end of the spout, so that once the hose is secured with a clamp, if cannot slide off..

The importance of being able to run higher system pressures will become obvious later in this section.

6.2.3 Radiator Hoses and Clamps – At a minimum I would recommend replacing the radiator hoses at least once each racing season. These items are often neglected and forgotten (at least until one fails and a motor is ruined). These hoses should be installed with NEW screw type hose clamps, and then the hose clamp wrapped in 3-M self-bonding electrical tape.

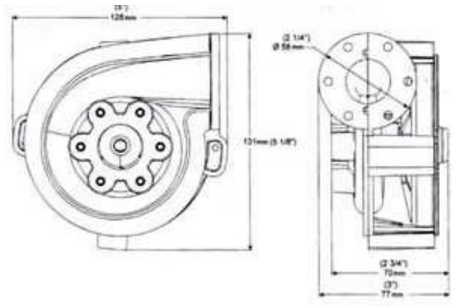
Other hose connections have been developed specifically for high performance purposes. These in include both fabric and stainless steel jacketed material with AN type screw on fittings, and Wiggins type connectors for joining and terminating solid cooling tubes.

6.3 Water Pumps

Mechanical – All of the mechanical water pumps currently available for the various type of Fiat blocks (817.843,903, 965, 982, 1050) are quite sufficient in terms of transport of fluid, even if using a remote front mounted radiator, provided they are properly driven.

By this I am not referring to whether you use a v-belt or some type of toothed belt, but rather to the size of the respective crankshaft and waterpump pulleys, and the ratio between them. The crankshaft pulley should always be 50% the diameter of the waterpump pulley. This will insure that the waterpump runs at 50% of the engine RPM. This will not only mean that it uses less horsepower, but also that it be less likely to cavitate.

Electric – There are now several type of electrical water pumps available that can be adapted to the Fiat/Abarth type motors. Some are meant to run at constant speed, whereas others incorporate a "controller", to regulate the speed of the pump. With a controller the water is only enough water is moved to maintain a given, preset water system temperature. I know of one competitor who has quite successfully used two "water guppy" type water pumps in his race car for over 10 years, so the idea is not at all new.



Technical Specifications:

Operating Voltage	4V DC to 14.5V DC
Maximum Current	7.5A
Flowrate	20L to 80L/min, (300 gal to 1300 gal/hr) 13.5V DC
Operating Temperature	-20C to 130C (-5 F to 270 F)
Pump Design	Clockwise centrifugal with volute chamber
Motor Life	2000 hrs continuous at 80 C (180 F) and 12V DC
Pump Weight	900 grams (2lb)
Pump Material	Nylon 66, 30% glass filled
Maximum Pressure	tested to 50 psi
Fits Hoses Sizes	32mm to 50mm (1-1/4" to 2")

Oil Coolers

Lubricating oil has two distinct purposes. Obviously, one purpose is lubrication, however a second purpose is to carry away heat from vital components within the engine. Components such as cam bearings, main and rod bearings can only stand so much heat before they fail. Likewise, boundary layer interfaces such as rocker arms and rocker shafts, also rely on sufficient oil flow though the interface to carry away heat.

So oil is an important "vital fluid" and its temperature must be maintained at an operating level that keeps it from breaking down. The "terminal temperature" for most organic oils used in automobiles is around 300 degrees Fahrenheit (148 deg. Celcius). For synthetic oils this number is slightly higher, however anything over 350 degrees will get dangerously close to the "flash point" temperature of most oils.

Fluid –to-Air Coolers

The most common oil coolers are similar to a conventional water radiator, only smaller. The idea, for any oil coller, is to provide enough temperature delta to maintain an oil temperature between 220-270 deg. Fahrenheit (104-132 deg. Celcius). In the fluid-to-air oil cooler oil is fed through a number of tubes, over which air flows to carry away the heat. To assist in heat dissipation, the tubes generally have small, thin fins attached.

Generally, oil from the oil pump should be fed to a filter first, and then to the oil cooler, before returning to the main oil gallery of the motor. In this way the cooler is protected from debris, and cooled oil is presented directly to the camshaft, crankshaft and connecting rod bearings.

Fluid-to-fluid Cooler

This type of cooler is better referred to as a “heat exchanger”, and has been used on the maritime industry for many years. Here water, (fresh or salt water) is routed through the large fittings, and oil is routed through the AN fittings. This is then 2-way system.

In a racing application several different versions have been utilized. In one implementation the oil cooler element is placed in the low delta side of a standard aluminium radiator. Thus, using a dual pass radiator, the first pass of the cooling fluid through the radiator sheds much of the engine generated heat, then the water passes around the exchanger element in the left header tank and dissipates the oil system heat in the second pass through the radiator before returning to the motor.

The unit above is a self-contained exchanger and a portion of the water flow, at the low delta temperature point returning to the motor, is fed through the exchanger.

The beauty of this type of oil cooling method is that the oil cooler can be placed almost anywhere with in the vehicle, obviously keeping it as low as possible, and it does not need to be in the air stream in order to be effective.

In the original Abarth TC and TCR implementations there was a dual radiator, placed in the front-mounted radiator shroud. This required both water and oil fluid lines to be routed from the back of the car to the front, and back again. Using a self-contained heat exchanger could save some weight in eliminating one set of the long lines to/from the front of the car.

Both types raise the overall water temperature slightly, so it is important to make sure that the water cooling system is up to the additional task with sufficient temperature delta to handle the extra load.

Induction and Exhaust Systems

7.1 Induction and Exhaust Systems

Making more horsepower is all about getting more air and fuel into the cylinder. Now there are many ways to accomplish this, some more technically challenging than others. Put differently, horsepower is all about optimizing "volumetric efficiency" (VE).

Since almost all historic racing clubs do not allow the use of turbocharging, supercharging or fuel additives, I will not spend a great deal of time on these issues except just to review how they affect volumetric efficiency.

Turbocharging and Supercharging - Here the idea is to simply cram more fuel mixture (air and gasoline) into the cylinder under mechanically induced pressure greater than one atmosphere. Using these methodologies the only limiting factor is how much additional fuel and air can be reasonably burned, and this is usually dictated by the ability of the engine components and the cooling system to handle the additional heat load.

Fuel Additives - There is a distinction that has to be made here between those chemicals that can be added to fuel to increase resistance to knock, and those that can be added to increase VE. The best known of these is MTBE which has an oxygen content of 18.2% and is in use in California in reformulated lead free gasolines sold to the public. Others are Methanol (49.9% oxygen) and Ethanol (34.7% oxygen). These are all readily miscible with gasoline. There are also other chemical agents such as Propylene Oxide and Aniline, both known to be very dangerous and require special handling, that can be used as octane enhancing additives. Some of these additives have extremely low boiling points, such that on a hot day over 95 degrees Fahrenheit they would simply boil on there own. This adds to the danger of their use, as in a vapor state they become highly volatile and explosive. If you choose to use these take special precautions. Quite obviously, additives like Nitrous Oxide and Nitromethane will also provide a performance boost, but these are easily detected.

7.2 Intake Manifold Variations

Now lets get back to what can be done to increase volumetric efficiency through non-chemical means. The Otto cycle combustion process is all about "timing". This includes cam timing, valve opening timing, and as you will see intake charge timing. Just as a properly designed exhaust system utilizes the "scavenging effect" to extract spent gases from the cylinder, the same effect can be put to good use in maximizing the amount of fuel/air mix goes in to the cylinder. This will deal with a number of issues in the fuel induction system.

The whole idea is construct an intake arrangement whereby the intake pulses arrive at the intake valve just before the valve opens, at the desired RPM range. This has a supercharging effect which increases the amount of air/fuel entering the cylinder by as much as 20%.

To examine this we will look at three areas:

- Intake port diameter and length
- Plenum volume
- Ram Pipe length and Diameter (Helmholtz Theory)
- First intake port diameter and length - In general the diameter should be approximately 85% of the valve seat diameter. While this is am approximation, it is pretty close as this is based on assumption that the valve itself will provide some restriction even at full valve lift. If therefore the valve seat diameter is 31mm, then the port diameter as it approaches the valve seat should be around 26mm.

There are two types of heads in popular use on Fiat Abarth rear engined cars, the standard/modified head and a 8-port aftermarket head like the PBS 8P.

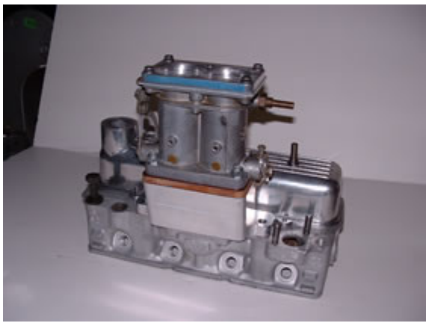
Now on a PBS head, with individual intake port runners, you have the ultimate flexibility in setting up your intake system. There might be a temptation to say that "big is better", but be careful. Large diameter ports may seem like they would flow a large volume of air, but they may lose a great deal of their velocity. I believe that in order to maintain low/mid range performance that the intake runner should maintain a certain amount of taper so that just before the intake valve seat the port is around 27mm, then blending out to the 31mm seat diameter. This means that the air will travel a path from the velocity stack bell (40mm) through the secondary venturi (30mm), through the throttle plate (40mm), down the intake runner to an area just before the valve (27mm) and through the valve seat. I have calculated that the theoretical length of the runners to achieve best performance around 7000 RPM should be 8-9 inches. The 8P design comes very close to this, and it would be difficult to change it. The design of the PBS 8P head promotes good midrange performance, while not restricting top RPM operation. I know of many engines that reliably run up to 9000 RPM.

The standard Fiat head is a different engineering exercise entirely. In reality we have an arrangement of two tri-Y intakes. If we divide the head into the right and left side, then inlet port of the head forms basically the bottom leg of the "Y" and the two short runners to the intake valves form the upward extensions of the "Y". This is identical for the left and right hand sides of the head. Just like in an exhaust manifold, this tri-Y arrangement has the effect of broadening the optimum RPM range over which the intake system is most effective. The intake manifold should be considered as a "plenum volume" common to all four cylinders. Unfortunately, as this head was designed originally for a 27 horsepower engine with limited RPM capability, the standard dimensions of the head do not lend themselves to a high performance application. It is possible to apply the port diameter theory in terms of making sure that the port diameter immediately prior to the intake valve seat is around 27mm. This will induce a secondary venturi effect just before the intake charge enters the cylinder. As far as port length goes, this is a combination of the two parts of the "Y" on each half of the head, being the very short individual port sections and the longer bottom of the "Y" section. Even so, this is not as long as ideal. As each cylinder that forms part of the Y fires alternatively to the other, the entire runner (both parts of the "Y") may be considered as the runner length for either cylinder. One could argue that if you had a two barrel carburetor with both venturis opening at the same time, then you could separate the two tri-Y areas. As we will see later on this may not be the case. The opening in the head should be of such dimension as to insure that adequate air flow is present and should match that of the bottom of the intake manifold/plenum.

Plenum Volume - Plenum chambers are designed to diminish the pulsing effect of the intake system, and this is particularly effective when more than one cylinder is fed from the same plenum. Plenums are not generally effective for more than 4 cylinders, although multiple plenums could be used for 6 and 8 cylinder engines.

For an 8-port head like the PBS unit there really is no plemum at all. Each port/intake runner combination acts as a standalone system. It is my view that this could be enhanced by adding a connection between the four intake runners just prior to entering the head itself. This would not be unlike the multiple tuned port arrangement in the BMW M44 engines, where an opening between the ports is opened at a certain RPM. This would then modify the effective RPM band to provide most efficient operation over a larger RPM band. Therefore the tunable range of the intake system of the 8P head will be fairly tight.

For Fiat Abarth heads, based on the standard Fiat head, the intake manifold provides for plenum chamber of sorts. To be ultimately effective around 7500 RPM the volume of the plenum chamber should be around 400cc on a 1000cc motor. This is the ideal number but is not critical. The plenum has the dual effect of dampening down intake pulse effects and also adding effective length to the intake runner. It is this very combination of intake port and intake manifold that then make up the effective port length that allow engines with standard heads to produce good power right up to 8000 RPM. Something that the original designer I am sure never intended.



Ram Pipe Length and Diameter (Helmholtz Effect) - The Helmholtz Theory was originally derived based on the harmonic effects of audio. He had postulated that a tone, or noise, was a combination of a primary frequency and a number of other secondary additive audio frequencies. Others applied this to fluid and plenum theory. There are three portions to the Helmholtz theory, namely plenum volume, ram pipe diameter and ram pipe length.

In the case of the PBS 8 port head the Helmholtz principle can be applied to each runner of the intake manifold. As such, about the only part that can be conveniently changed is the length and possibly the diameter of the velocity stack. Most velocity stacks concern themselves with providing a smooth air entry into the carburetor by ensuring a proper minimum radius on the bell of the stack. It may be that there is some advantage in actually making a velocity stack in which the bell opening, while having the suggested entry radius, might actually be smaller than the diameter of the secondary venturi. This would in effect increase the velocity of the air entering the carburetor and the adjustment in length would insure proper timing arrival of the pulse. Even with the PBS intake manifold there is a small difference in overall runner length for the outside cylinders, so one might consider using slightly shorter velocity stacks on these cylinders.

For the standard Fiat Abarth head, using a 2 barrel downdraft carburetor, the Helmholtz principle may have some further implications. In essence the volume of the area below the throttle plates, up to the back of the valve head is a plenum chamber. Therefore the velocity stack on the caburetor, along with the area of the secondary venturi in each carburetor throat makes up the entry to the "plenum". We can vary the tuned entry length both in diameter and length to get greater effectiveness at certain RPMs.

There may be an opportunity for further gains by employing yet another plenum chamber which would enclose the top of the carburetor and the radiused carburetor throat entry. Again, this plenum chamber should at a minimum be 500cc in volume on a 1000cc engine.

There are some basic generalizations that can be applied.

- A longer the intake tract will work better at lower RPMs
- A smaller diameter intake tract will promote better torque at lower RPMs
- A shorter intake tract will work better at higher RPMs
- A larger diameter intake tract will promote better HP at higher RPMs

Therefore, we need a good combinations that will promote good torque at around 5000 RPM, yet still provide good HP in the 8000-9000 RPM range.

7.3 Exhaust System Design

Exhaust systems, sometimes referred to as "headers", have been a subject of fascination for me for some time. Like other things, the design of header systems is sometimes is regarded as a "black art". However, when examined closely there are some basic rules that can be applied. This is not to say that the formulas and ideas that I am to present are the absolute truth on the matter, as every engine will have its own particular characteristics. The concepts put forward will however get you close to an optimum exhaust system.

Before I go too much further I have to give credit for much of the technical material in this article to the authors of a 1966 Hot Rod publication called "Supertuning". Bill Allen, the 1969 SCCA D-Sedan National Champion, used the same information to design an exhaust for his winning NSU. As the displacement of that car is similar to that of the Abarths that we still race today, I thought it a good place to start.

On all engines, including Abarths, a tuned exhaust system provides significant advantage, but this advantage is limited to a particular RPM band, depending on the design. Making a good tuned exhaust is not particularly difficult. The object is to get the reflected wave of one cylinder to help scavenge The next cylinder in the firing order. To better understand this it must be understood how waves are generated. The primary exhaust has a very high positive value. When this wave reaches to end of the exhaust pipe it is reflected back up the exhaust pipe as a negative pressure wave, which in turn is reflected again when the exhaust valve closes. Given the right timing, this secondary wave will help in extracting the residual exhaust gases. This wave movement provides further benefit in engines with camshafts with long duration and large overlap. In this case the intake and exhaust systems can drop below atmospheric pressure. When this occurs, the fresh mixture will begin to flow into the cylinder even before the piston initiates its intake stroke. This contributes to Volumetric Efficiencies in excess of 100%.

The real problem is in determining when the primary exhaust wave reaches full force, There is no sure way of determining this, but there is a fairly good empirical formula. I apologize in advance for not providing metric equivalents, but it became far too complex I should also mention that this model only deals with 4 into 1 type systems, although the computations are fairly close for 4:2:1 systems as well. It is:

$L = V120/N$

L is the length of the exhaust pipe primary from the head of the exhaust valve to the end of the merge collector (not including exhaust pipe)

N is the desired peak RPM

V is the speed of the wave in the exhaust gases (1700 ft/sec)

A "tuned exhaust" is generally only resonant over a relatively small RPM range, about 1500 RPM. If we consider that 8500 RPM is our absolute maximum, then subtract 1500 RPM, this leaves us with a midrange RPM of 7750. This would make the effective range from 7000-8500 RPM. At all other times the exhaust manifold would be less than 100% efficient. You may prefer to tune the exhaust somewhat lower in the RPM band, as with most Abarth engines peak torque is achieved at around 5500 RPM.

Let's model an Abarth 1050 motor as an example. Using the above formula with a peak 8000 RPM, we find that the ideal primary pipe length is 25.5 inches. Remember that this overall length includes the exhaust track in the head and the length of the merge collector, but not the secondary exhaust pipe. This is an approximation, but it will get you close. If you are using a cam with a lot of duration (say 300 degrees or more) then the pipe should be 1-2 inches longer.

Of great importance is the diameter of the primary pipes of the tuned exhaust. The diameter of the pipe will directly influence the <u>velocity of the gas flow</u> (note: not the wave) . The diameter should be such that, at the engine's power peak, the mean velocity of the exhaust gases is about 300 ft/sec. At this speed there is a balance, between the internal friction on the pipe on the one hand and the benefits of increased gas speed on scavenging on the other. When gas velocity is high enough, the sudden rush of exhaust products from the cylinder will tend to pull much of the residual gases along, and may even leave a slight vacuum in the cylinder, which will further aid cylinder filling during the intake stroke. If the primary pipes are made too small the the gas speed becomes so great that that the scrubbing along the inside of the pipe impedes gas flow and creates back pressure. The 300 ft/second point appears to be a good compromise.

The next step is to calculate the gas speed. Here is the formula:

V= (piston speed/60)X(D2/d2)

V is gas velocity in in feet per second

D2 is the piston diameter squared

d2 is the inside pipe diameter squared

Piston Speed is in feet per minute

Now then, for our Abarth 1050 engine lets start out with the following characteristics: 8000 RPM, 2.913 inch (74mm) stroke and 1.25 inch (31mm) pipe diameter. To determine piston speed in feet per minute we take the stroke times two (remember there are two strokes, one up and one down, for each turn of the crankshaft), namely 5.826 inches. This is them divided by 12, being the number of inches in a foot. This gives a result of .485 foot. This must then be multiplied by the desired crankshaft speed of 7500 RPM to give the piston speed. This computes to 3884 feet per minute or 64.73 feet per second. We then finish off the last steps by squaring the piston diameter of 2.657 (7.059) and the pipe 1.25 inch (31mm) pipe diameter squared is 1.562 inch (39.7mm. We proceed then by dividing 7.059 by 1.562 for an answer of 4.519, then multiplied by the piston speed of 64.75 ft/sec. The exhaust gas velocity computes to 292 ft/second.

Well, this is within 8 ft/sec. of our target. Almost perfect!!

Therefore we can now indicate that a 4:1 exhaust system for an Abarth 1050 motor having a 67.5mm bore and a 74mm stroke, with a mean peak RPM of 8000 RPM, will require exhaust primaries of between 27 and 29 inches, 1.25 inches in diameter. This would give a peak power RPM range between 7000 and 8500 RPM.

As the primaries come together in the collector, care must be taken to make sure that the tubes are <u>fitted so that the next firing cylinder is adjacent in the collector</u>. This will greatly increase the scavenging effect. Once in the collector, the exhaust pipe can be a minimum 15-20 inches in length. You will only find the "right" length by trial and error. There are different schools of thought as to whether the exhaust “pipe” should be a megaphone, or straight. Generally a tapered exhaust pipe will promote a slightly wider power band. You will remember that Abarth used such an exhaust megaphone on many of his race cars. On some engine designs, for sake of ease of installation, you may find having to use a “resonant multiple” for the exhaust pipe. So if a pipe of 22 inches (560mm) works well, but does not exit the car, then the next choice would be 44 inches (1120mm). The TCR exhaust is a good example of this.

A word of caution!! There is a temptation to make the exhaust primary tubing too large in diameter. Bigger is not always better, unless some thing else has been compensated for.

There we have it. Easy, right !! Well actually the math is easy. Now comes the hard part of finding the right tubing sizes, and making sure that the final design has equal length tubes, and will actually fit inside the engine compartment.

A Note of Interest - I went back and computed a complimentary intake length for this exhaust system and it indicates that the total length of the intake path should be 13.2 inches (33.5cm) from the bell of the velocity stack to the back of the head of the intake valve. If you look at a 1000TC intake manifold with a 36DCD7 carburetor and velocity stack, I believe you will find that this is very close to what Abarth used.

7.4. Carburetion and Fuel injection

7.4.1 Carburetion

The standard head, be it a Fiat 600/600D 850 or A112, have all been equipped with various 2 barrel, downdraught carburetors. The two manufacturers most often used as Weber and Solex.

The most often seen Weber models are 36DCD7 and DCN/DCNF carburetors.

The 36DCD7 can be found in two versions. One has both venturies opening simultaneously, whereas the other version has a progressive secondary. I have had experience with both types. At full throttle there is virtually no difference in performance, however the progressive carburetor may have a slight advantage in terms of fuel economy at partial throttle.

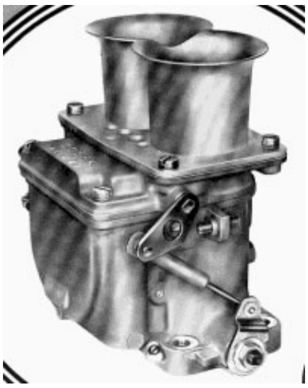
The DCN series of carburetors come in various varieties and have and have been used for many years on marques such as Ferrari and Maserati. For these engines the carburetors were cast in lightweightt aluminum, whereas most versions are cast from the conventional alum/zinc mix. All DCN type carburetors are non-progressive

Both of these types of carburetors should be mounted with phenolic spacer, to isolate it from heat generated by the cylinder head.

A good starting point for jetting would be as follows

28-29mm venture
130 Main
175 Air
45-50 Idle

For a number of years Solex supplied a carburetor specifically suited for competition purposes. This unit, the 36-40CCI carburetor was designed for competition. Again this is a non-progressive design, made from lightweight aluminum alloy.



These carburetors are always popular and when they do become available they are quickly purchased. They have been out of production for over 25 years, but they are still one of the best designs.

Abarth equipped the TCR head with dual Weber 40DCOE carburetor. This is probably the most widely implemented carburetor design for competitive applications.

With an individual throttle for each cylinder, this provides the ultimate in fuel control. The PBS 8P head, which also incorporates individual runners, is perfectly suited to use these carburetors.

A good starting point for jetting would be as follows

29-32mm venture
125 Main
180 Air
F11 Emultion Tube
45-50 Idle

There are other carburetor types that have been used successfully on Fiat/Abarth competition vehicles. These include Del'Orto units from Italy and various flat slide carburetors.

7.4.2 Tuning Weber Carburetors

All Weber carburetors have four circuits.

Idle
Cross-over
High Speed
Accelerator

The Idle and Cross-over circuits are inter-connected, in that the idle jet is responsible for the fuel supply for both circuits. It provides fuel to the idle mixture screw (responsible for idle supply up to about 1500 RPMO), and the crossover ports, which are hidden by the butterfly and come into play as soon as the throttle open, because the incoming air literally "sucks" the fuel into the airstream. The cross-over ports provide additional fuel in this critical phase and If the idle jet is too small, then there is insufficient fuel to satisfy the needs of both the cross-over ports, and of course the idle mixture screw as well, as both circuits are active simultaneously. This mechanism take care of the "sudden" rush of air when you open the throttle butterfly. If the fuel is not sufficient at this point, the engine will run lean and stumble momentarily. Of course once the butterfly is fully open, the vacuum on the intake tract drops, and the motor is running on the main jet and air corrector jets. The crossover ports and the mixture screw have done their job and are no longer active, at least until you close the throttle and it all starts again.

If the idle mixture jet is correctly sized, then the idle mixture screw should provide a stable idle when it is 2-4 turns from being fully closed (seated on the seat) at an idel speed of 1000-1200 RPM. Be careful when you screw it fully in, on the seat, so as not to damage anything. I suspect with the large venturi that you need to change the idle jet to the next larger size. It probably has a 45 in it now. I would change it to 50, or perhaps even a 55. Then you can back off the idle speed screw and you will then have to reset the idle mixture screw and it should come back to 2-4 turns from fully closed. This should get rid of the stumble at 1500-2000 RPM. I would also increase the main jet on both the primary and the secondary and raise it 5 points from what it is now. So if they are 125, I would change them to 135. The LAST thing that you want is to have the engine run LEAN at high RPM. It will run great for a short period of time, before the pistons melt. The carburetor also has an accelerator pump that squirts a pre-determined amount of fuel with the opening of the secondary venturi. Much like the cross-over ports help the primary venturi with a small additional amount of fuel, the accelerator pump does the same for the secondary venturi. If this were not so, then there would be an instant when the secondary venturi would run lean (on first opening) and this would cause a hesitation in the power delivery. It is important to get this part of the carburetor correct, as this is what gives the engine that "crisp" response so important when accelerating out of a corner.

Just in case you friend does not have the right jets, get on the Web and go to www.mcmaster.com or some equivalent Canadian company and order an assortment of miniature drill bits and a "Pin Vise". I would order drill bits from 1 to 2.5 millimeters in .05mm steps, or some 30 drill bits, and a .45, .50, and a .55mm drill bit as well. You are now equipped to deal with any jet in any carburetor (except those that use needles, but that is another story). They should be between $1 and $2 dollars each. Make a little holder from a block of wood and label each of the drill bit by its size. A 125 jet has a hole 1.25mm, and a 130 jet has a hole 1.30mm. As you can see, as a last resort you can make your own jets. If you need to go smaller, a soldering iron and some regular soft solder will close the hole, and then you can re-drill it for whatever size you need. The drill bits are also your "gauge" as you can with them measure any jet to make sure what size it is.

As far as high speed running is concerned a good rule of thumb is to start with 125 Mains and 175 Air Correction jets (a numerical spread of 50) if you have no idea where to start. The fuel and air as joined together in the "emulsion tube". In the cavity in the emulsion tube the air/fuel is mixed (emulsified) and aerated before it is fed to the opening in the secondary venturi. The air stream passing through the secondary venturi literally carries the fuel with it into the combustion chamber. I like F11 emulsion tubes for DCOE carburetors, but regardless of the type of carburetor, the emulsion tube is the LAST thing that you fine tune, if necessary. The engine should at least run on this combination. Next I recommend that you use a Air/Fuel meter to monitor high speed fuel situation. The key number is 13:1. Yes, a stochiometric mixture would be 14.7:, and this is fine if you are tuning to pass a smog test, or for ultimate fuel economy, but it is much too lean for a competition motor. If course if it reads 10:1, then you are much too rich. Be careful, you need to test the entire RPM range, as it could be that slow running is OK but wide open throttle is too lean, or vise-versa. If you do not have an A/F meter (also known as a Lambda meter), then you have to use a bit of intuition.

If the engine misses at high RPM, then it is entirely likely that the main jet is too small, particularly if the exhaust pipe is very light in color (white to very light grey). The engine is leaning out at high RPM and if this situation persists, then you will do major damage. You can have the same miss if the engine is grossly too rich, except that the tailpipe will now be black and it will be spewing large amounts of black smoke (unburnt fuel) and it may foul the plugs. If you are standing behind such a car, it will not be long before your eyes begin to water.

This brings us to how to adjust main jets. Adjustment steps in the main fuel jet are generally done in .05 sizes (from a 125 to a 130). Of course the larger the number the more fuel it will pass. Main jet changes affect the carburetor fuel delivery throughout the entire RPM range, once the carburetor is past the cross-over circuit (from about 2200 to ????? depending on how brave you are).

Air Correction jet use a different formula. In short, 4 steps of air correction change is roughly equivalent to a single incremental step in fuel jet. Change the air correction from 175 to 195, and that would be the equivalent to changing the fuel from 130 to 125. Confused? In essence you are letting more air into the emulsion tube in relation to fuel, so the mixture is "leaner". Here comes the interesting part. Whereas a fuel jet change affects the entire RPM range equally, changes in air correction jet have greater effect at RPMs above 5000 RPM. So if the engine is running fine everywhere, but a little lean at top RPM, then you could fine tune it by going 5-10 points smaller on the air correction jet.

Tuning Webers is as much an art as it is a science, as every now and then you will have an engine that needs something totally different, but this is VERY rare.

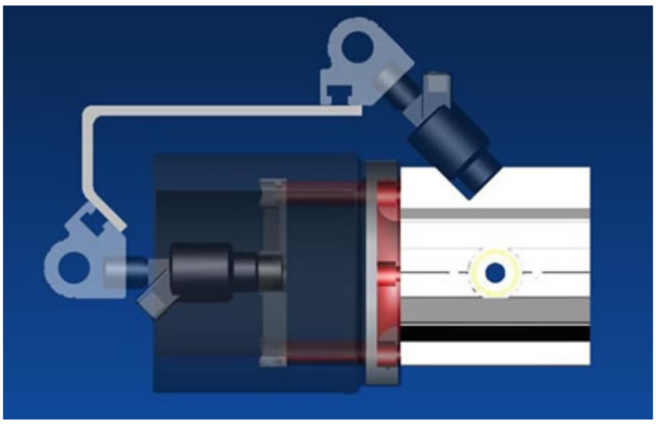
7.4.3.Fuel Injection

As we are talking about competition purposes, I will only discuss those variations of fuel injections that are directly applicable to competition vehicles.

There are basically two type, mechanical and electronic. From a design perspective they both employ a “throttle body” per cylinder. On the mechanical side the only one that was actually implemented on an Abarth vehicle was the Kugelfischer type. This had a “slide” type throttle, instead of butterfly type, and a complex belt driven mechanical, high pressure pump.

The latest ECU driven, electronic fuel injection systems therefore have a real advantage over the earlier mechanical system. If you look at the early Kugelfisher systems, these were really only totally efficient at near, or full throttle. It is for this reason that the injectors were placed well upstream. This aided high RPM performance, but did little for lower RPMs and made for a very narrow usable power band.

It would be possible to use an ECU that could be programmed to fire two banks of injectors, either singly or together. This would allow me to place one set of injectors just past the Injector Throttle Bodies (ITB) or as close as practical to the valve, and the other set above the inlet trumpets. (See diagram). I found a company called Extrabody that was working on something similar and we are now collaborating on a solution that could be produced as a kit for the PBS 8P cylinder head.



The Extrudabody ITB units can be mounted on a manifold designed for a Weber DCOE carburetor. This makes the system simple to mount to the 9P head. Because the system is modular, additional extrusions can be added, either before or after the ITB portion to custom tune the intake runner length to meet almost any design criteria. As the second drawing illustrates, the system can also cater for the two injector idea that I described earlier.

Using one of several aftermarket ECUs, a map can be derived that automatically sequences between the two injector banks, either using both injectors at the same time, or switching from one bank to another.

Now, I understand that this type of system is not allowed in many racing clubs, but where it is, it hold some reasonable promise, particularly when you add to this the ability of the ECU to control spark advance as well.

8.1 Ignition Systems

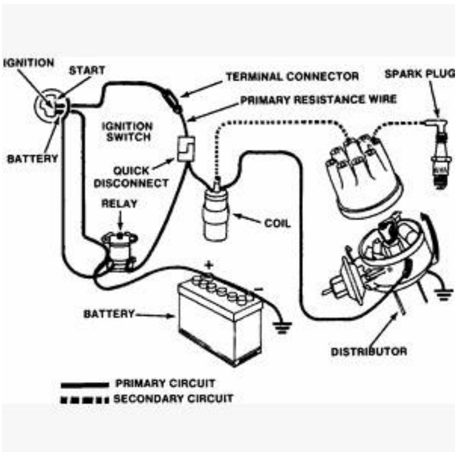
The ignition system sends an extremely high voltage to the spark plug in each cylinder when the piston is at the top of its compression stroke. The tip of each spark plug contains a gap that the voltage must jump across in order to reach ground. That is where the spark occurs.

The voltage that is available to the spark plug is somewhere between 20,000 volts and 50,000 volts or better. The job of the ignition system is to produce that high voltage from a 12 volt source and get it to each cylinder in a specific order, at exactly the right time.

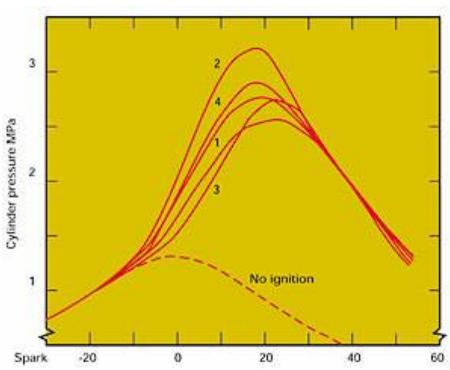
Let's see how this is done.

The ignition system has two tasks to perform. First, it must create a voltage high enough (20,000+) to arc across the gap of a spark plug, thus creating a spark strong enough to ignite the air/fuel mixture for combustion. Second, it must control the timing of that the spark so it occurs at the exact right time and send it to the correct cylinder.

The ignition system is divided into two sections, the primary circuit and the secondary circuit. The low voltage **primary circuit** operates at battery voltage (12 to 14.5 volts) and is responsible for generating the signal to fire the spark plug at the exact right time and sending that signal to the **ignition coil**. The ignition coil is the component that converts the 12 volt signal into the high 20,000+ volt charge. Once the voltage is stepped up, it goes to the **secondary circuit** which then directs the charge to the correct spark plug at the right time. Here is a diagram of such a system.



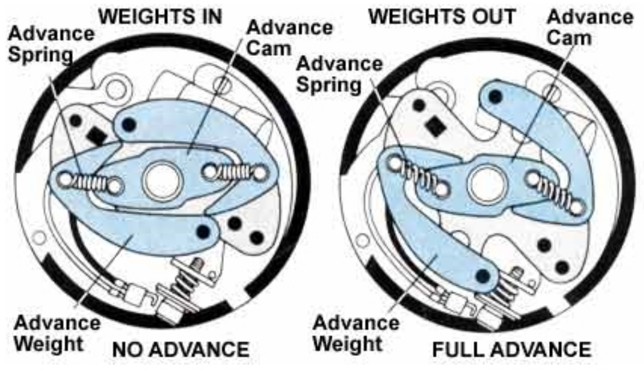
Without exception, almost all ignition systems were “mechanical” systems, up to the early 70s. Engines, to operate efficiently, require the spark to fire at some point BEFORE the piston reaches TDC.



This is to allow the explosion to build enough pressure (push) on the top of the piston, at just the right time, to provide optimum power. If it is started too soon (advanced) then this explosion reaches piston while it's still traveling upward and you lose power, (trying to push the piston the wrong way) waste energy, and create heat in the combustion chamber area (and usually knocking or detonation from an explosion instead of a nice smooth flame traveling from the upper cylinder to the piston top). If started too late (retarded) then you loose power because the piston is already traveling downward, before the flame explosion can "push" it. This also creates heat in the surrounding combustion chamber because remember, heat is energy. This energy, if not used to push the piston, is released either into the surrounding water jacket or the exhaust manifold instead of powering your vehicle. Both are inefficient as far as maximum power is concerned, but it makes an effective heater! As the engine RPM's increase, given that the flame propagation speed remains the SAME, then the combustion cycle needs to be started earlier to achieve the desired "push" on the top of the piston. Also, as the pressure (more fuel/air) inside the cylinder increases, then the less advance the engine can handle at a lower RPM (bigger explosion). So as you can see it depends upon the speed (RPM) of the engine, AND the amount of air/fuel mixture (throttle position) that the engine is operating at. OK, elementary internal combustion education is out of the way.

These systems all used an “advance” mechanism to alter the timing of the spark pulse to the cylinder. A good starting point for setting the initial timing (static timing) is 10 BTDC. You might call this “idle advance” at 1200 RPM or below. As the engine RPM increase, the amount of time available for ignition decreases, so the initiation of spark must occur earlier. There is a limit to the amount of advance, but for standard engines this would be around 28 degrees.

So how do we get from the 10 degrees at idle to the 28 degrees at higher RPM?

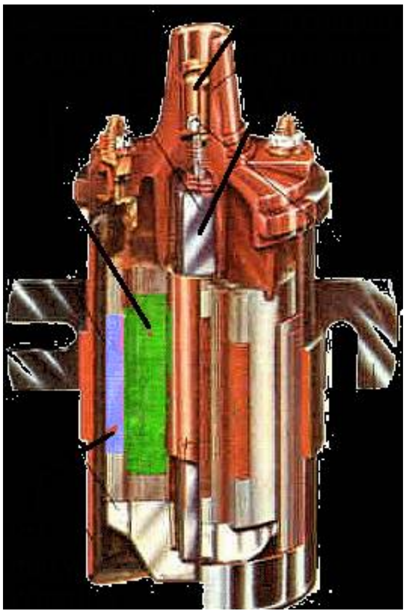


Above is a good illustration of the standard mechanical advance mechanism. The point plate in the distributor has two advance weights on separate pivot pins. The advance weights are connected to the advance cam via small coiled springs. These springs provided tension to keep the cams against the cam. As the distributor spins, the centrifugal forces generated cause the weights to want to move away from the centerline of the distributor, however the springs provide a resistance to this centrifugal effect. As the weights swing outward the points plate is rotated so that the “points opening event” occurs earlier. By varying the cam design, and the tension of the advance springs, it is possible to change the advance characteristics of the distributor. In most competition engines, using a mechanical advance mechanism, the distributor is generally fully advanced by 3000 RPM.

Now say your cruising at 2000 RPM little load, again low cylinder pressure, optimum advance (30 deg) engines happy. Suddenly you snap open the throttle. Now you have maximum cylinder pressure, low engine speed and advance needs to be at say 12 deg to prevent detonation. If the advance were purely mechanical again, and set for optimum advance (30) at the no/low load condition, then we would have too much advance for this high load condition, and one unhappy engine because of detonation. However, during high load conditions, the intake manifold pressure drops to zero (equals outside manifold pressure or no vacuum). IF the mechanical timing were now optimized for high load, low speed conditions (12 deg@2000 RPM), then the vacuum unit can optimize the timing at light or no load conditions (30 deg) because it is in effect not operating at high load conditions, and the mechanical advance can be optimized for high cylinder pressure or maximum load conditions.

So in this case, when you stomp on the pedal, the timing (at 30 deg light load, relatively high vacuum) would drop back to 12 deg, because the vacuum is now not operating, as stated before, the manifold pressure increased (vacuum dropped to zero) and the diaphragm returned to it's no vacuum position. In this way, timing can be optimized for all engine conditions. For racing, and max power applications, you don't really need a system for controlling advance at low or no load conditions because these engine are operating at maximum power most if not all the time. (and is one reason why some tend to overheat at idle) Also, another reason that early emission systems with idle retard, or advance cutouts have a provision that during extended idle periods, when the engine begins to overheat, it restores PROPER advance to prevent that overheating! Note: High performance engines generally do not use vacuum advance mechanisms, as the vacuum generated is relatively low due to the longer camshaft duration and overlap.

This would be a good time to speak about ignition coils. To put it simply a coil is nothing more than a “step-up” transformer. It has a primary side (12V DC from the car’s power supply) and secondary side. Each side of the transformer has a number of windings, and the number will be sufficient to step up the 12V to between 20,000-50,000 Volts. It is this secondary voltage that causes the spark in the cylinder when it jumps from the center electrode of the spark plug to ground.



8.2 Points Type

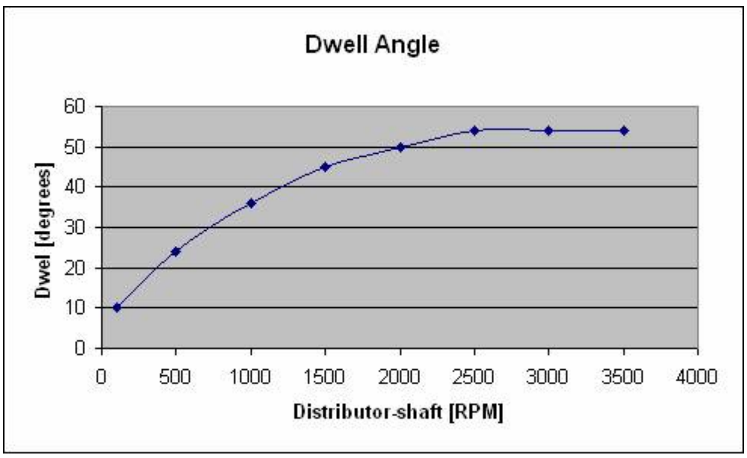
Ignition points are a set of electrical contacts that switch the coil on and off at the proper time. The points are opened and closed by the mechanical action of the distributor shaft lobes pushing on them. The points have a tough job, switching up to eight amps of current many times per second at highway speed. Indeed, as engine speed increases the efficiency of your ignition system decreases, thanks to heating problems and fundamental electrical laws. This declining efficiency has a serious effect on your spark voltage and results in poor high-speed performance, incomplete combustion and other drivability problems.

Condenser: Those same principles of inductance create a kind of paradox, because when the points open and the magnetic field collapses it also induces a current in the primary as well. It's not very much because there are only a few windings in the primary, but it's enough to jump a small air-gap, such as the one between the just-opening points in the distributor. That tiny spark is enough to erode metal away from the points and you'll 'burn' the points. It prevents the points from arcing and prevents coil insulation breakdown by limiting the rate of voltage rise at the points.

Ballast Resistor: This is an electrical resistor that is switched in and out of the supply voltage to the ignition coil. The ballast resistor lowers voltage after the engine is started to reduce wear on ignition components. It also makes the engine much easier to start by effectively doubling the voltage provided to the ignition coil when the engine is being cranked. Not all car manufacturers used a ballast

A points type of distributor uses a set of spring loaded points to "initiate" the spark event. When the points close, current through the coil primary increases from zero to maximum in an exponential manner, rapidly at first, then slowing as the current reaches it's maximum value. At low engine speeds, the points are closed long enough to allow the current to reach a higher current level. At higher speeds, the points open before the current has time to reach this maximum level. In fact, at very high speeds, the current may not reach a level high enough to provide sufficient spark, and the engine will begin to miss. This current through the coil builds a magnetic field around the coil. When the points open, the current through the coil is disrupted, and the field collapses. The collapsing field tries to maintain the current through the coil. Without the Condenser, the voltage will rise to a very high value at the points, and arcing will occur. Another possible problem with points type distributors, particularly at higher RPMs, is that you may get "point bounce". This would be and indication that the points spring is not strong enough to run at higher RPMs.

Setting the Dwell Angle - The Dwell-Angle is the number in degrees of rotation of the distributor-shaft, whereby the breaker-points are closed. *(The same for an electronic ignition module, discussed immediately below), only here the breaker-points are replaced with a control-module and the lobes on the distributor-cam are replaced by a reluctor. The reluctor induces pulses which are past on by the ignition-signal sensor to the ignition-module. The ignition-module "tells" the power-transistor to turn the current through the primary-coil on or off. The time in which the power-transistor turns the current "on", is also expressed in degrees of rotation of the distributor-shaft.)*



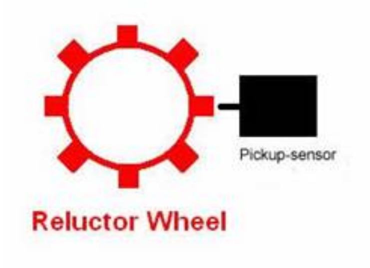
For a mechanical system, the Dwell-angle is during operation a fixed number, about 50 degree for a 4-cylinder engine. This means that the current flows through the primary-coil for 50 degrees of distributor-shaft rotation, regardless the RPM. For example when running idle: 500 rpm, the crankshaft makes 1 revolution in 120ms. The distributor-shaft, at half speed, 240ms. It takes 4ms to charge the ignition-coil till saturation. The required Dwell-angle is (360° / 240ms) x 4ms = 6°. In reality the ignition-coil is charged for 50° duration, 44° more than required. The 50° Dwell-angle is required if the crankshaft makes 1 revolution in (360°x 4ms) / (50x2) = 14.4 ms. This is 4,166 rpm, nearly full throttle!!! So above the 4,166 rpm the ignition coil is charged below saturation, and the spark intensity will therefore be less. Certainly at 9000 RPM we may be pushing the limit of the ability of the coil to reach sufficient saturation to support combustion for extended periods of time.

8.3 Electronic

Electronic Ignition systems are not as complicated as they may first appear. In fact, they differ only slightly from conventional point ignition systems. Like conventional ignition systems, electronic systems have two circuits: a primary circuit and a secondary circuit. The entire secondary circuit is the same as in a conventional ignition system. In addition, the section of the primary circuit from the battery to the battery terminal at the coil is the same as in a conventional ignition system.

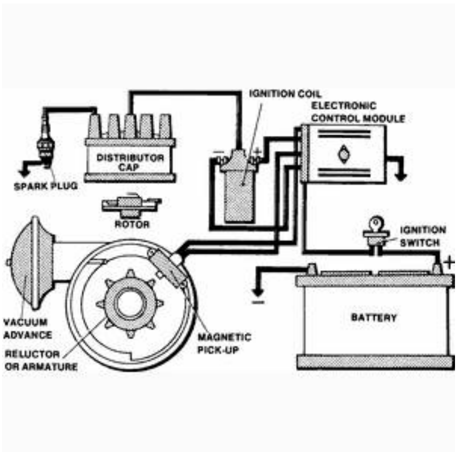
The primary circuit of the electronic ignition systems operate on full battery voltage which helps to develop a stronger spark. Whereas, Breaker point systems needed a resistor to reduce the operating voltage of the primary circuit in order to prolong the life of the points.

Electronic ignition systems differ from conventional ignition systems in the distributor component area. Instead of a distributor cam, breaker plate, points, and condenser, an electronic ignition system has an armature (called by various names such as a trigger wheel, reluctor, etc.), a pickup coil (stator, sensor, etc.), and an electronic control module.



Essentially, all electronic ignition systems operate in the following manner: With the ignition switch turned on, primary (battery) current flows from the battery through the ignition switch to the coil primary windings. Primary current is turned on and off by the action of the armature as it revolves past the pickup coil or sensor. As each tooth of the armature nears the pickup coil, it creates a voltage that signals the electronic module to turn off the coil primary current. A timing circuit in the module will turn the current on again after the coil field has collapsed.

So in the case of the Magneti Marelli "Marelliplex" system. The distributor still has a mechanical advance system, on which is mounted the magnetic pickup. Because the signal from this pickup is extremely low voltage, it has to be amplified, so that it is sufficiently strong enough to trigger the coil. In addition the amplifier module, located on the large aluminum heat sink (to which the coil is mounted) has the timing circuitry to turn off the coil primary current. The module used by Magneti Marelli is actually a standard unit made by Delco, and the coil is an ordinary coil without a voltage drop resistor.



It would therefore be perfectly feasible to build your own electronic ignition system from readily available parts. Here is a short shopping list.

A) Standard Fiat 850/903/A112 distributor

B) Pertronics (or equivalent) sensor module and reluctor

3) Delco or Chrysler Ignition amplifier

4) MSD Coil (without ballasts resistor)

8.4 Capacitive Discharge & Transistorized Ignition

An advantage of the capacitive discharge ignition system is that the energy storage and the voltage ‘step up' functions are accomplished by separate circuit elements allowing each one to be optimized for its job.

Capacitive discharge ignition systems work by storing energy in an external capacitor, which is then discharged into the ignition coil primary winding when required. This rate of discharge is much higher than that found in inductive systems, and causes a corresponding increase in the rate of voltage rise in the secondary coil winding. This faster voltage rise in the secondary winding creates a spark that can allow combustion in an engine that has excess oil or an over rich fuel air mixture in the combustion chamber. The high initial spark voltage avoids leakage across the spark plug insulator and electrodes caused by fouling, but leaves much less energy available for a sufficiently long spark duration; this may not be sufficient for complete combustion in a “lean burn” turbocharged engine resulting in misfiring and high exhaust emissions.

The high voltage power supply required for a capacitive discharge system can be a disadvantage, as this supply provides the power for all ignition firings and is liable to failure.

Ignition in lean fuel mixtures by capacitive discharge systems can sometimes only be accomplished by the use of multi-spark ignition, where the ignition system duplicates the prolonged spark of inductive spark systems by sparking a number of times during the cycle. The MSD unit is a good example of this. At engine RPMs below 3000 the MSD provided multiple spark events. Above this RPM it reverts to a single spark event.

Below is a link to a very good discussion of the aspects of electronic CDI ignition.

http://www.mclarenelectronics.com/Products/All/App_Act_Ign.asp

8.5 Crankshaft Triggered or Distributorless Ignition Systems (DIS)

The third type of ignition system is the distributorless ignition. The spark plugs are fired directly from the coils. The spark timing is controlled by an Ignition Control Unit (ICU) and the Engine Control Unit (ECU). The distributorless ignition system may have one coil per cylinder, or one coil for each pair of cylinders.

Some popular systems use one ignition coil per two cylinders. This type of system is often known as the waste spark distribution method. In this system, each cylinder is paired with the cylinder opposite it in the firing order (usually 1-4, 2-3 on 4-cylinder engines). The ends of each coil secondary leads are attached to spark plugs for the paired opposites. These two plugs are on companion cylinders, cylinders that are at Top Dead Center (TDC) at the same time. But, they are paired opposites, because they are always at opposing ends of the 4 stroke engine cycle. When one is at TDC of the compression stroke, the other is at TDC of the exhaust stroke. The one that is on compression is said to be the event cylinder and one on the exhaust stroke, the waste cylinder. When the coil discharges, both plugs fire at the same time to complete the series circuit.

Since the polarity of the primary and the secondary windings are fixed, one plug always fires in a forward direction and the other in reverse. This is different than a conventional system firing all plugs the same direction each time. Because of the demand for additional energy; the coil design, saturation time and primary current flow are also different. This redesign of the system allows higher energy to be available from the distributorless coils, greater than 40 kilovolts at all rpm ranges.

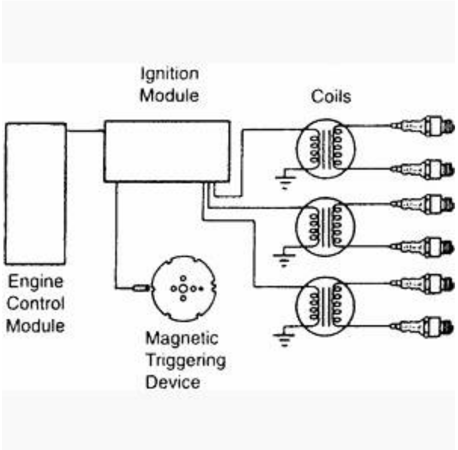
The Direct Ignition System (DIS) uses either a magnetic crankshaft sensor, camshaft position sensor, or both, to determine crankshaft position and engine speed. This signal is sent to the ignition control module or engine control module which then energizes the appropriate coil.

The advantages of no distributor, in theory, is:

- No timing adjustments
- No distributor cap and rotor
- No moving parts to wear out
- No distributor to accumulate moisture and cause starting problems
- No distributor to drive thus providing less engine drag

The major components of a distributorless ignition are:

- ECU or Engine Control
- Unit ICU or Ignition Control
- Unit Magnetic Triggering Device such as the Crankshaft Position Sensor and the Camshaft Position Sensor
- Coil Packs



8.6 Ignition Timing and Combustion

Under ideal conditions the common internal combustion engine burns the fuel/air mixture in the cylinder in an orderly and controlled fashion. The combustion is started by the spark plug some 5 to 40 crankshaft degrees prior to top dead center (TDC), depending on engine speed and load. This ignition advance allows time for the combustion process to develop peak pressure at the ideal time for maximum recovery of work from the expanding gases.

The spark across the spark plug's electrodes forms a small kernel of flame approximately the size of the spark plug gap. As it grows in size its heat output increases allowing it to grow at an accelerating rate, expanding rapidly through the combustion chamber. This growth is due to the travel of the flame front through the combustible fuel air mix itself and due to turbulence rapidly stretching the burning zone into a complex of fingers of burning gas that have a much greater surface area than a simple spherical ball of flame would have. In normal combustion, this flame front moves throughout the fuel/air mixture at a rate characteristic for the fuel/air mixture. Pressure rises smoothly to a peak, as nearly all the available fuel is consumed, then pressure falls as the piston descends. Maximum cylinder pressure is achieved a few crankshaft degrees after the piston passes TDC, so that the increasing pressure can give the piston a hard push when its speed and mechanical advantage on the crank shaft gives the best recovery of force from the expanding gases.

Detonation: A violent explosion; also called combustion knock. This usually occurs near the end of the combustion process when highly compressed, high-temperature end gases spontaneously ignite, radically increasing the cylinder pressure. This pressure spike moves at the speed of sound in the combustion chamber, and the pressure can cause damage to pistons, cylinder walls, and the head gasket.

Pre-ignition: The onset of combustion before the spark plug fires. This is generally caused by some type of glowing ignition source such as a hot exhaust valve, too-hot spark plug, or carbon residue. Pre-ignition is especially damaging to engine components like pistons and head gaskets, since excessive cylinder pressures can occur even before the piston reaches top dead center (TDC).

These are the classic definitions of detonation and pre-ignition. Perhaps a more fun definition of detonation would be to imagine the piston screaming up to TDC while you whack that piston as hard as you can with a 10-pound sledgehammer. The clang that you would hear is the same noise that occurs when your engine goes into detonation. Even if detonation doesn’t break any parts, as soon as an engine experiences detonation, the power drops way off. If you ever have a situation where at a certain point, when you give it more gas, the car physically slows, then STOP.

If you get the idea that detonation and pre-ignition are bad, that’s good. Of all the things that can kill an engine, detonation should be right at the top of your Public Enemy Number One list. The quickest and easiest way to cure detonation is to use a high-quality, higher-octane gasoline

Perhaps the easiest and least- expensive way to reduce an engine's sensitivity to detonation is to cool the engine-inlet air. Not only is cooler air more dense, which makes more power, but cooler air is also less prone to detonate. The classic performance rule-of-thumb is that for every 10 degrees you reduce the inlet air temperature, the engine makes 1 percent more power. This is why drag racers use ice to cool the intake manifold and why all those cold-air inlet systems work on late-model cars. Forcing your engine to breathe hot underhood air will also make it more prone to detonate, so make sure your carburetors have ready access to air that is at least at ambient temperature. In addition, keep your fuel as cool as possible as well.

Ignition timing is another cheap and easy area to work on. If your engine detonates at low engine speeds at part throttle, consider retarding the initial timing by 2 or 3 degrees and then adding that amount back into the total by increasing the mechanical-advance curve. For example, let's say you have 10 degrees initial timing with a total of 30 degrees and your engine rattles a little at part throttle, especially right off idle. You could cut the initial back to 8 degrees and add 2 degrees to the mechanical advance. The total remains at 30, but now the engine doesn't death rattle every time you let the clutch out..

Camshaft timing also plays a huge role in dynamic cylinder pressure, especially with street-driven performance engines. As you increase intake duration, this means the intake valve now closes later than it does with a shorter-duration cam. This later-closing intake valve bleeds some cylinder pressure back into the intake manifold at lower engine speeds. The longer the duration of the cam, the later the intake closes. This reduces cylinder pressure at lower engine speeds, which reduces the tendency for the engine to detonate.

Late closing of the intake can also be accomplished by retarding the camshaft's installed point. For example, many small-block Chevy cams are installed with the intake centerline at 106 degrees after top dead center (ATDC). This tends to close the intake valve sooner, which improves low-speed torque by increasing cylinder pressure at low speeds. But if the engine rattles at low speeds, retarding the closing point of the intake valve can by 3 or 4 degrees (from 106 to 110 degrees ATDC) softens the engine's need for higher-octane fuel.

Obviously, this is a little more difficult to do than playing with ignition timing but may pay off by allowing you to run a lower-octane fuel. If you do retard the cam, it's important to go back and perhaps add a degree or two of initial ignition timing.

You can also experiment with camshaft overlap. Unfortunately, this requires a new camshaft. Tightening the lobe separation angle, from 114 degrees to 110 degrees, for example, increases the amount of overlap since the exhaust valve closes slightly later and the intake valve opens a little sooner. This tends to bleed off cylinder pressure at lower engine speeds, which could be beneficial since this is a little like built-in exhaust gas recirculation (EGR) in the intake manifold.

There are several other ideas that you can try to reduce your engine's sensitivity to detonation and allow it to live on lower-octane fuel. Any kind of oil contamination in the combustion chamber is bad news. Oil is a great breeding ground for creating detonation. The best way to avoid this is to ensure your combustion space enjoys the benefits of tight valve-to-guide clearances and good leak-free valve guide seals. Of course, you want to seal up that intake so it doesn't suck oil into the cylinders, and your short-block should be in good shape.

8.7 Spark Plugs

Spark plugs provide one of the elements, without which an engine simply will not run. Well, at least a non -diesel engine any way. Most of us that dabble in cars would recognize that the primary purpose of spark plugs is to ignite the air/fuel mixture that enters the cylinder.

Spark plugs provide you with a window into the combustion chambers and also provide you with an evidence trail of what is going on in there. Spark plugs and their condition are one of the most important diagnostic tools. They tell you what happened in the cylinder. They will help you in tracking down what the root cause is for many problems and to maximize the air/fuel ratio

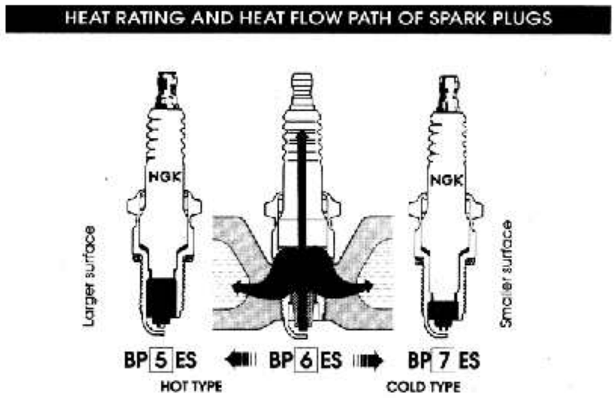
The two things that spark plugs do are:

1) Ignite the air/fuel mixture
2) REMOVE heat out of the combustion chamber.

If a sufficient amount of voltage is applied to the spark plug, so that the resultant spark spans the gap between the electrode and ground, then it is said to have sufficient *electrical performance*. In addition, the temperature of the spark plug's working end must be kept low enough to prevent pre-ignition, yet not too low so as to permit fouling. This is often referred to as the thermal performance of the hear range selected.

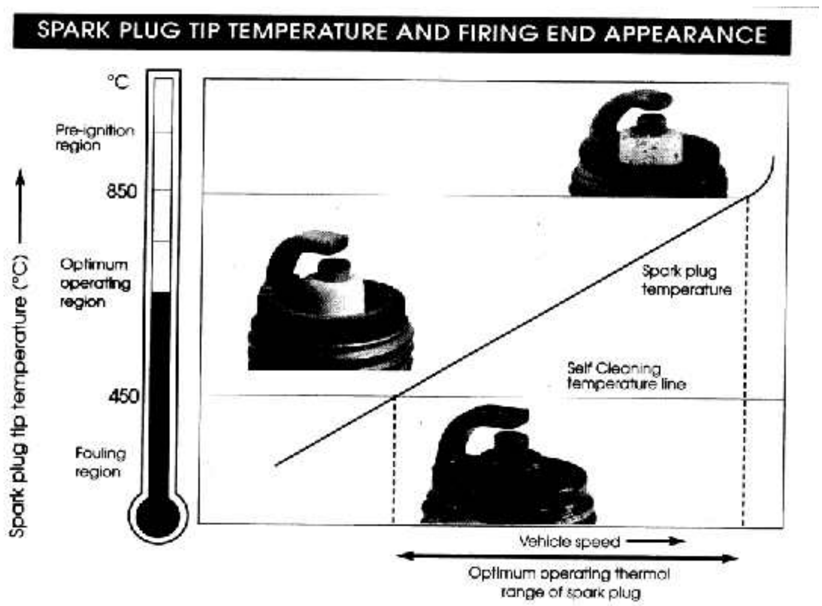
One popular misconception is that spark plugs CREATE HEAT. This is absolutely incorrect, and in point of fact spark plugs can only REMOVE HEAT. The spark plug is like a radiator, taking heat out of the combustion chamber and transferring it to the engine's cooling system. Therefore the heat range of a particular plug indicates the plug's ability to to shed heat. The rate at which a plug shed's heat to the cooling system is determined by:

1) The insulator nose length
2) The volume of gas round the insulator nose
3) Materials/construction of the center electrode and porcelain insulator.



The hear range of a particular spark plug has no relationship to the actual voltage transferred through the spark plug. As stated earlier, the heat range is simply an indicator of the spark plug's ability to remove heat from the combustion chamber. This heat transfer effectiveness is a function of ceramic insulator nose length and material composition of the insulator and center electrode.

The insulator nose length is the distance from the firing tip to the point where the insulator meets the metal shell of the spark plug. The insulator tip is the hottest part of the spark plug and therefore plays a crucial and primary role in both pre-ignition and fouling. No matter what the application, from lawn mower to race car, the tip temperature must remain between 450-850 degrees Centigrade. If the temperature is below 450C, the plug will not burn off carbon and combustion chamber deposits, including lead deposits if high performance fuels are used. This will lead to a misfire. On the other hand if the combustion temperatures are over 850C then the ceramic tip will overheat and fracture and the cause the electrode to melt. Once pre-ignition sets in, major damage can be done to an engine in a very short period of time. I know this to be the case, as I have personally experienced both ends of this scale.

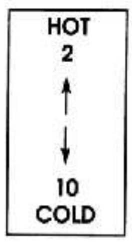


Presuming you use the spark plugs with the same electrical/mechanical characteristics, moving one heat range colder will allow the plug to remove between 70-100 degrees from the combustion chamber.

Projected tip plugs run about 10-20 C degrees hotter than standard plugs. By the same token retracted tip plugs MAY runs somewhat cooler, but much will depend on their construction.

The optimum is to find a plug heart range that will work in the crossover range between fouling and optimum operating range. This is sometimes referred to as the "self-cleaning" range and is somewhere around 600 degrees Centigrade. Here there is little chance of pre-ignition or detonation, yet there is little buildup of carbon and combustion chamber deposits.

Let's get back to the length of the insulator, as this sometimes causes much confusion. The longer the distance between the tip and the spot where the insulator meets the spark plug body, the longer the heat of the combustion chamber has to travel before it can be dissipated to the cooling system. Hence this would be a "HOT" plug. The shorter this path, therefore the colder the plug and the colder plug will remove heat more quickly and reduce the chance of pre-ignition/detonation. A good rule of thumb is to use the coldest plug that is available that does not foul. THen you can work with carburetor jetting to get the air/fuel mixture to where it produces maximum results.



WARNING: Each plug manufacturer has their own way of denoting heat ranges. By example NGK uses a low number (2) to denote a HOT plug, whereas a number 10 would be a very COLD plug. Quite the opposite, Champion uses low numbers to denote a cold plug and high numbers for hot plugs. Make up your own cross reference chart so as not to get confused.

Credits: Some of the information on this page was derived from NGK literature.

Recommended Torque Specifications

SPARK PLUG THREAD SIZE	CAST IRON HEADS		ALUMINUM HEADS	
	WITH TORQUE WRENCH	WITHOUT TORQUE WRENCH	WITH TORQUE WRENCH	WITHOUT TORQUE WRENCH
FLAT SEAT w / GASKET				
18mm	3.5 kg-m ~ 4.5 kg-m 25.3 lb-ft ~ 32.5 lb-ft	1/2–2/3 turn 180°–240°	3.5 kg-m ~ 4.0 kg-m 25.3 lb-ft ~ 28.9 lb-ft	1/2–2/3 turn 180°–240°
14mm	2.5 kg-m ~ 3.5 kg-m 18.0 lb-ft ~ 25.3 lb-ft	1/2–2/3 turn 180°–240°	2.5 kg-m ~ 3.0 kg-m 18.0 lb-ft ~ 21.6 lb-ft	1/2–2/3 turn 180°–240°
12mm	1.5 kg-m ~ 2.5 kg-m 10.8 lb-ft ~ 18.0 lb-ft	1/2–2/3 turn 180°–240°	1.5 kg-m ~ 2.0 kg-m 10.8 lb-ft 14.5 lb-ft	1/2–2/3 turn 180°–240°
10mm	1.0 kg-m ~ 1.5 kg-m 7.2 lb-ft ~ 10.8 lb-ft	1/2–2/3 turn 180°–240°	1.0 kg-m ~ 1.2 kg-m 7.2 lb-ft ~ 8.7 lb-ft	1/2–2/3 turn 180°–240°
TAPERED SEAT				
18mm	2.0 kg-m ~ 3.0 kg-m 14.5 lb-ft ~ 21.6 lb-ft	1/12–1/8 turn 30°–45°	2.0 kg-m ~ 3.0 kg-m 14.5 lb-ft ~ 21.6 lb-ft	1/12–1/8 turn 30°–45°
14mm	1.5 kg-m ~ 2.5 kg-m 10.8 lb-ft ~ 18.0 lb-ft	1/12–1/8 turn 30°–45°	1.0 kg-m ~ 2.0 kg-m 7.2 lb-ft ~ 14.4 lb-ft	1/12–1/8 turn 30°–45°

All engine manufacturers have recommended torque specifications for spark plug installation, and most are represented in the chart above. Install the spark plug finger tight until the gasket or taper seat contacts the cylinder head. Then give it the recommended turn or angle.

Some content attributed to the following authors.

Vincent T. Ciulla