itself, to meet the previously-stated minimum surface/volume requirement, would again be a spherical segment – with a radius that provides the total volume, added with that from the clerance space between piston and squish band, to give the desired compression ratio.

The clearance space between piston and cylinderhead must be enough to avoid contact at high engine speeds, yet close enough to keep the mixture held there cooled during the combustion process. This vertical clearance between squish band and piston should not be greater than .060-inch, and it is my opinion that the minimum should be only barely enough to prevent contact – usually about .015-inch in small engines (with tight bearings and cylinder/rod combinations that do not grow, with heat, disproportionately) and up to about .045-inch in big engines.

Some disagreement exists as to the validity of claims that the squish band aids combustion by causing turbulence in the combustion chamber as a result of the piston "squishing" part of the charge between itself and the head. I don't know about that, but I do know that holding squish band clearance to a minimum means that there will be the smallest volume of end-gases escaping the combustion process, and that can be more important than you might think. For example, a 250cc cylinder with a full-stroke compression ratio of 10:1 will pack its entire air/fuel charge into a volume of only 28cc by the time its piston reaches top center. Assuming that it has a 3-inch bore, and a 50-percent squish band with a piston/head clearance of .045-inch, then the volume of the charge hiding in the squish area will be in the order of 2.6cc, or almost 10-percent of the total. That can be reduced to 5-percent merely by closing the squish band's clearance to .020-inch – and you'll never find an easier 5-percent horsepower difference. True, the difference measured at the crankshaft might prove to be more like 2½-percent, but the addition of those small percentages can make a very large final difference.

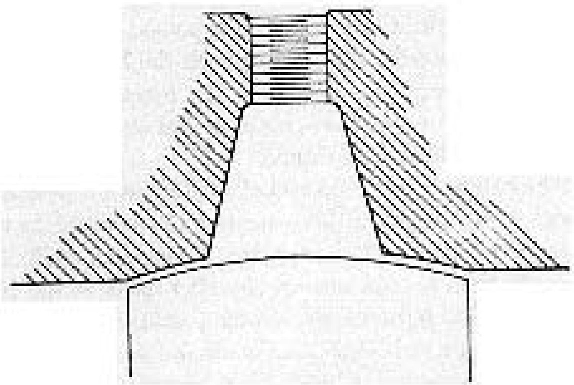
PLUG LOCATION

Tests have shown that the best location for the spark plug is, by and large, squarely in the center of the combustion chamber, and with its gap as close to the center of the volume of trapped mixture as possible – which is logical, as that position provides the shortest flame travel in all directions. However, a number of other considerations do intrude. First, the plug gap will necessarily be at the periphery of any part-sphercial chamber, and not at its center, and trying to form a nob in the chamber roof – to move the plug deeper into the mixture volume – will upset the chamber's surface/volume ratio. Secondly, moving the plug too close to the piston seems to cause a local overheating of the piston crown, which can impose an unnecessarily low ceil-

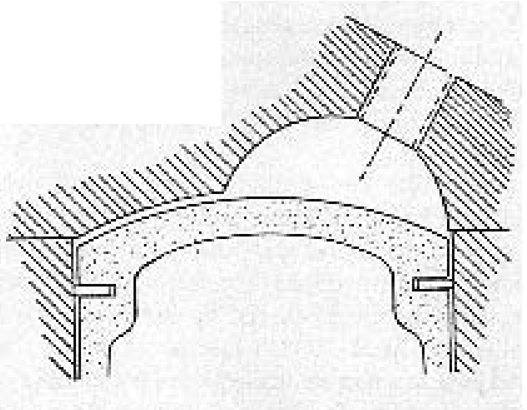
ing on compression ratio.

This last consideration has, in many instances, led development engineers to use combustion chambers with forms that allow the plug to be positioned well away from the piston: modified spheroids; conical sections, etc. Also, chambers with higher roofs (like those shaped as cones) with their spark plugs up at the top and the broader base down at the piston, provide a slightly slower pressure rise as combustion progresses, and are in consequence a bit more kind to bearings. Other switches in plug location may be made in the interest of easing the job of plug replacement: it is difficult to change a plug centered in the cylinderhead when the bottom of a fuel tank, or frame tube, is directly overhead.

Fortunately, most engines usually are relatively insensitive to plug location as long as the gap isn't moved too close to the piston. Which raises an interesting point: The common practice of shaving material from the cylinderhead's lower surface not only raises the compression ratio, and thus the thermal load on the piston, but it brings the plug gap close to the piston crown – compounding the problem. A better approach to obtaining increases in compression ratio is to purchase a cylinderhead developed to do the job properly. Yamaha's GYT-kit heads, for example, provide the right compression boost, have their spark plugs properly located, etc. Other made-for-the-job cylinderheads offer the same fundamental advantage, which is that you get



Conical combustion chambers reduce thermal loads on the piston and mechanical loads on rod and bearings, but have poor surface/volume characteristics and give a too-broad squish to be efficient.



Heat input into the piston tends to be higher on its exhaust-port side. Moving the combustion chamber over to the intake side will help balance the thermal loads and permit higher compression ratios.

to buy a lot of other people's engineering at a very low cost.

Not all cylinderheads have their spark plugs and combustion chamber pockets centered over the cylinder bore, and there are good reasons for most of the variations in form one sees in the products of the major manufacturers: For instance, piston crown temperatures seldom are even, and while the overall temperature distribution pattern is understandably inclined toward maximums in the center of the crown, circumstance can also lend a bias toward the exhaust port. That bias comes not from any heat-input pattern, but rather from the manner in which the piston crown is cooled – by heat transference into the air/fuel mixture below, and into the piston skirt, from whence it is transferred out into the cylinderwalls. Cooling provided by the turbulent crankcase charge is more or less even; the same cannot be said of heat losses into the cylinder, for the temperature gradients around the cylinder's walls are most uneven. The area around the exhaust port is hotter than that back at the intake port, even though the exhaust-side of the cylinder is in most instances the recipient of the direct cooling-air blast. Moreover, the exhaust-port side of the piston skirt is bathed in fire every time the port opens at the end of a power stroke. The overall result is to move the maximum temperature point on the piston crown toward the exhaust port.

Now, when that maximum temperature bias begins to seriously overheat the side of the piston, you are likely to see some severe piston ring problems

develop: Too-high temperatures will eventually be a disaster for the ring itself, but more often it will not have a chance to show its displeasure because another disastrous situation will already have developed, with the lubricating oil. Sometimes, if a relatively high ash-content or inadequately de-gummed oil is used, the ring will be glued solidly in its groove by varnish and carbonized oils. More often, the temperatures prevailing in that section of the piston skirt adjacent to the exhaust port will cause a breakdown of the oil film in that area and the piston will seize. And this can happen even though a generous margin of safety still exists all around the rest of the piston skirt. A common, and highly sensible solution to this problem is to move the combustion chamber pocket away from the bore axis, toward the back (inlet) side of the cylinder. This measure shrouds more of the piston crown's exhaust side under the squish band – which becomes crescent-shaped, instead of being a symmetrical ring – and reduces heat input there from combustion (the skirt will still be getting plenty of heat when the exhaust port opens) enough to provide a more even distribution of heat around the piston skirt. Then, with piston-skirt temperatures evened-out, a slightly higher compression ratio may be used without incurring seizure, or localized overheating of the piston ring.

There is another solution to the problem that has nothing whatever to do with the cylinderhead: you simply add metal to the piston crown, and that, too, will tend to equalize skirt temperatures – but it also makes the piston heavier. Even so, it is a solution much-loved by manufacturers, as adding thickness in the piston costs virtually nothing, while any departure from symmetry in combustion chamber configuration entails multiple machining operations (it being extremely difficult to cast, with sufficient accuracy, the combustion chamber's small volume) and machining-time is expensive.

There may be another reason for employing an asymmetrical combustion chamber, and/or relocating the spark plug from its normal position over the bore axis. In loop-scavenged two-stroke engines, the fresh charge is directed upward, and at, the rear cylinderwall, as it emerges from the transfer ports. Ideally, the mixture streams converge and sweep up and over at the top of the cylinder to clear away exhaust products and push them out the exhaust port, following the rear cylinderwall upward, and then curling back smoothly under the cylinderhead. In practice, the scavenging stream tends to be much less ordered in its habits, and the general turbulence can make it leap and dodge all over the place, impinging strongly at one point and only eddying at others. This leads, in some engines, to a reshaping and repositioning of the combustion pocket – the purpose of such changes being to aid scavenging by using the combustion chamber's form to give the scavenging stream direction.

In such cases, the spark plug may also be moved to a position where it will be washed by the mixture stream, which tends to cool the plug between firings, and thus make the engine somewhat less sensitive to plug heat range.

Also, as noted before, the plug may be moved away from the combustion chamber center to create a slightly longer path for flame travel, which lowers the rate at which pressure in the cylinder rises during the combustion process and, in some instances, makes for smoother running. To a lesser extent, the same treatment may be used to combat a tendency toward detonation, as the lower pressure-rise rate gives all the pockets of end-gases time to lose their heat into the surrounding metal. This last effect is, of course, better obtained with a conical combustion chamber, rather than by offsetting the plug. Incidentally, moving the spark plug over too close to any edge of the bore is usually poor practice: At times, particularly when starting from cold, the piston ring will scrape oil off the cylinderwalls and pitch it up at the cylinderhead, and if you place the spark plug in the line of fire, it definitely will show a weakness for oil-fouling.

HEAD/CYLINDER SEALING

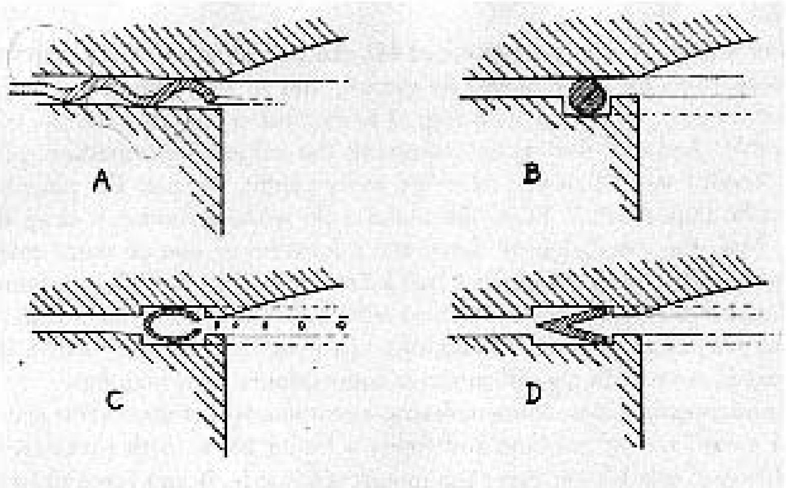
A major problem with cylinderheads on high-output engines that began life as low output engines is persistent leakage around the head/cylinder joint. The combined increases in temperature and pressure seem always to be too much for the joint, and you will find evidence of fire puffing past on the surfaces after disassembly even though you may not have observed anything out of the ordinary when the engine was running. This leaking will occur even if you have retained the engine's stock compression ratio, and it may become very serious if the head has been thinned to get a compression ratio increase. Many manufacturers, perhaps most, feel some awful compulsion to skimp on section thicknesses when they make a cylinderhead, a habit that often stands revealed as a questionable economy when you test their handiwork on a dynamometer: first, the thin sections often do not have the cross-sectional area required to transfer heat away from the head's lower surface quickly enough to keep the spark plug temperatures stabilized; second, most of these cylinderheads are secured to their cylinders by only four widely-spaced bolts, which presumes heavily on their beam-strength to maintain a tight seal at the joint.

This last situation becomes especially marginal when metal has been machined away to raise the engine's compression ratio, and the stock head gasket (usually cut from light-gauge, soft aluminum) will in many cases not be strong enough to hold even the pressure increases involved in a simple switching of exhaust systems. Shave the head (which both weakens the head's beam strength and increases the forces acting upon it) and you'll very likely find that it becomes impossible to hold the head/cylinder seal — the gasket will fail after only minutes of running. Also, attempting to use the stock

cylinderhead, in either standard or modified form, often will increase the heat input around the spark plug to such extent that the engine becomes impossibly fussy about plug heat range. Use a plug cold enough to avoid trouble at maximum output, and it will foul at anything less than full-throttle operation. There is nothing like masses of metal to equalize the temperature gradients through the cylinderhead, and — sad to say — those masses are not provided in many stock cylinderheads.

Cylinderhead design also can strongly affect overall cylinder cooling. When the cylinderhead's lower surface is cooler than the cylinder itself, heat will be drawn away from the latter; conversely, a cylinderhead can also put heat into the cylinder if the situation is reversed. All things considered, the engine's best interests probably are served by isolating, to such extent as is possible, the cylinder and head — which means restricting the contact area at the cylinder/head joint to a narrow sealing band which bulges to encompass the hold-down bolts, or studs. In that way any cooling problems will be isolated, and can be dealt with separately. That, of course, assumes that it will be possible to improve cylinder cooling should such improvement become necessary. Actually, making a new cylinderhead is fairly easy (it can be either cast or simply machined from a block of aluminum) while the cylinder itself presents a far more difficult problem in fabrication. So you may very well want to use an oversized, deeply-finned cylinderhead to help cool a particular engine's stock, cast-iron cylinder. And if that should be the case, remember that you'll need a maximum contact area between head and barrel, and surfaces that will seal without any kind of gasket. There is a very sharp temperature gradient across any joint, and even a solid copper gasket presents one more pair of surfaces across which heat must flow.

You may find that providing a seal between the head and barrel is one of the more difficult facets of the overall job. As I have said, stock aluminum gaskets are almost certain to fail, being a bit weak at ambient temperatures anyway — and impossibly frail at the temperatures to which they will be subjected. Copper is a better material, for while it is nearly as soft as aluminum at ambient, its hot-strength properties are better. Copper is soft enough to make a good gasket in the annealed state, but hardens in use, and must be re-annealed frequently to keep it soft and thus retain its properties as a gasket. Brass should never be used as a gasket material, but steel may be used if it is very thin and has one or more corrugations rolled, in rings, around the bore — in the manner of the head gaskets used in some automobile engines. You can also get a good seal by machining a narrow groove in the cylinder's upper face and inserting in it a soft copper ring (made from wire) to bear against the head's lower surface. Other, even better seals may be had with gas-filled metal O-rings, piston rings (they'll work here, too) and one of the best sealing rings I've seen has a V-shaped section, laid on its side, with the V's point aimed



Shown here is a rolled-steel head gasket (A); a soft copper wire located in a groove (B); a hollow, gas-pressurized O-ring (C); and a V-section stainless sealing ring also sealed by gas pressure (D).

away from the bore. Gas pressure tries to force the V open, bringing one arm to seal down against the cylinder while the other is pressed against the cylinderhead. Another sealing ring that works in roughly the same fashion is a hollow metal O-ring with vent-holes drilled through from its inner diameter to admit gas pressure from the cylinder – which expands it outward and thus creates a seal even between somewhat uneven surfaces.

Nominal compression ratios, as I have said before, have little meaning in high-output two-stroke engines. However, you can work with *trapped* compression ratios almost as effectively as by measuring cranking pressures. An engine's trapped compression ratio is the ratio between the cylinder volume at the moment of the exhaust port's closing and the volume with the piston at the top of its stroke. To find this, you must first measure the combustion chamber volume, with the piston in position at top center. The job can be done with the engine assembled, using a graduated cylinder and pouring in oil until the level comes up to the spark plug hole. Or you can calculate the volume. When the combustion chamber has a simple shape (part-spherical, conical or cylindrical) I prefer to do the job by calculation, but more complex shapes send me scurrying for a can of oil and a graduated cylinder. In fact, the process of actual measurement may appeal to you as a regular thing, because you will need a graduated cylinder for more than this single task, and a slide-rule may not be a part of your basic equipment. In any case,

remember when figuring the compression ratio, that it is *not* the ratio between piston displacement and combustion chamber volume, but between cylinder volumes from the point of exhaust port closing to top center, as in the following formula:

$$CR = \frac{V_1 + V_2}{V_2}$$

Where CR is compression ratio

V_1 is cylinder volume at exhaust closing

V_2 is combustion chamber volume

Traditionally, compression ratios have been measured "full stroke". That is to say, V_1 would represent the combustion chamber volume plus piston displacement from bottom center to top center. Thus, a combustion chamber volume of 28cc and a piston displacement of 250cc, calculated full-stroke, would be

$$CR = \frac{250 + 28}{28}$$

$$CR = 9.93{:}1$$

But a far more realistic figure is obtained when V_1 represents the cylinder volume above the upper edge of the exhaust port, and if we assume that our hypothetical engine has an exhaust port height equal to 45-percent of stroke, then V_1 becomes 55-percent of piston displacement plus V_2, and calculation goes like this:

$$CR = \frac{(.55 \times 250) + 28}{28}$$

$$CR = 5.91{:}1$$

Coincidentally, that compression ratio (5.91:1) is very nearly all a non-squish combustion chamber will permit in an otherwise fully-developed two-stroke engine. With small-bore engines you may push the compression ratio up to perhaps 6.5:1 without serious consequences, using a non-squish cylinderhead, but that is very near the limit. Good squish-band cylinderheads, on the other hand, permit compression ratios up to as much as 9.5:1 in motocross engines with exhaust systems that provide a wide boost without any substantial peaks, but for road racing engines I cannot recommend anything above 8.5:1 even when unit cylinder size is only 125cc. You will find that higher compression ratios than those suggested can produce marvelously impressive flash readings on a dynamometer; as soon as the engine has a chance to get up to full temperature, the output will drop well below that sustained by an otherwise identical engine with a lower compression ratio. Sustained, and not flash horsepower, is what wins races.