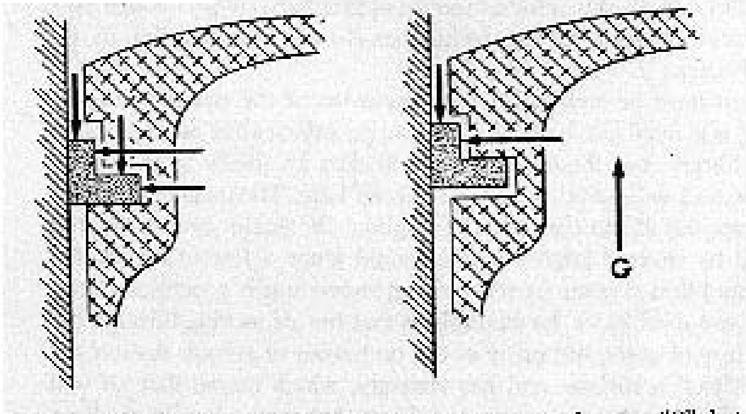
at that speed, maximum piston acceleration will be (with the answer rounded off by my slide rule; I'm too lazy to do it all with paper and pencil) no less than 135,000 ft/sec². Now if you will recall for a moment that the acceleration of gravity is only 32 ft/sec², it will be clear that the load on the Yamaha's pistons—and thus on its rings—is very high indeed. But is the loading high enough to make the Yamaha's rings flutter? Obviously, it is not, as the engine remains not only reliable but crisp in comparatively long races. The limit, for the TD-2 engine, is slightly higher than 135,000 ft/sec²—but not much higher, as you will see in the following table listing ring thicknesses and the accelerations at which they begin to flutter.

For rings having a .125-inch thickness,			40,000 ft/sec²
.094	"	"	53,000 ft/sec²
.063	"	"	80,000 ft/sec²
.047	"	"	106,000 ft/sec²
.039	"	"	138,000 ft/sec²

The Yamaha, with rings having a thickness of 1mm, or .039-inch, and a maximum piston acceleration of 135,000 ft/sec² at 11,000 rpm, would seem to be operating very near the limit—as indeed it is. But it probably is not quite as near the limit as the numbers suggest, for a racing ring (with its exaggerated thickness/width cross-sectional aspect) is somewhat less subject to flutter than a ring made for application in a touring engine. Still, the numbers given are fairly close for rings with normal-range proportions, and if you have an engine with rings for which flutter is predicted at 80,000 ft/sec² and intend using crankshaft speeds that would raise maximum piston acceleration to something more like 100,000 ft/sec², then I strongly urge you to fit new pistons with thinner rings. You may interpolate between the figures given to find the safe acceleration levels for ring thicknesses not listed.

There are piston rings that resist very strongly piston acceleration's efforts toward making them flutter. The best known of these is the Dykes-pattern ring, which has an L-shaped cross-section and fits into a similarly-shaped groove in the piston. The Dykes ring is made flutter-resistant by the fact that its horizontal leg fits quite closely in its groove, as compared to clearances around the vertical leg, and therefore even if acceleration lifts the ring it cannot lift high enough to close off the pressure behind the ring's vertical leg. In consequence, the ring's sealing abilities are maintained at accelerations that would be the undoing of rings in the conventional rectangular-section pattern. However, the Dykes ring's ability to maintain a seal does not free it of all the unpleasantness attending too-high piston acceleration: while it may seal under those conditions, it is still being rattled about vigirously and if the rattling continues long enough, the Dykes ring, and the groove trying to restrain it, both become badly battered. At that point, its ability to seal



Dykes-pattern rings are subject to the effects of piston acceleration, "G", but gas pressure is maintained behind the ring because its horizontal leg limits its vertical travel.

vanishes and mechanical failure of the ring, piston, or both, follows very closely. Bultaco has long used Dykes-pattern rings, as have certain others, but most manufacturers prefer rings that do not require such careful and intricate machining. There are other flutter-resistant rings, and many excellent reasons for using rings of conventional configuration, but these details are discussed elsewhere in this book and in greater depth than would be appropriate here.

After establishing all these mechanical limits, with regard to piston speed and acceleration, and after deciding how much power you are likely to get from a particular engine, you should subject the engine to a complete survey. This would include the measuring of port heights and widths, combustion chamber and crankcase volumes, and charting piston travel against crank rotation. This last effort may at first seem rather pointless, but as your work progresses you will find that the chart, which will show almost but not quite a sine curve, provides an instant readout between degrees at the crankshaft and the position of the piston from top center that is most useful. It will tell you, for example, how much to raise the top edge of an exhaust port to make a given change in timing, and how much to trim from the piston skirt (in a piston-port engine) to get the intake period you want—or think you want. The chart also will provide you with all the *mean* port-open points, and it will provide an exceedingly useful relationship between ignition timing expressed in degrees and in piston travel from top center. You may devise your own methods for deriving all this information according to your preference and resources; I have explained my own techniques elsewhere in this text, in the appropriate chapters.

An item that must be included in any discussion of the two-stroke cycle engine's basics is general gas dynamics. You can get information on the subject at your local library, but the applicable particulars are likely to be widely scattered there, so I will cover the subject in brief here. The manner in which what follows applies at specific points throughout the engine and its related plumbing will be covered later, but you should know a few of the fundamentals now, and thus save me from becoming unnecessarily repetitious later.

One thing you must know, for example, is that the air moving through the engine, a mixture of gases, has many of the properties of a fluid. It even has the ability to "wet" a surface, and has viscosity, which means that air will cling to all surfaces within an engine in a layer that moves hardly at all no matter what the midstream velocity may be. This boundary layer's depth is influenced by gas temperature, and by the temperature of the surface on which it forms, as well as by the shape of the surface. Please understand that the layer is not solid; it is "shearing" with general flow throughout its depth—which may be as much as .100-inch—with movement increasing as to distance from the surface on which it is formed. And as close as .020-inch from the surface, flow may still be in the order of 80-percent of that in midstream, which means that the restriction formed by the boundary layer is not very great. Nonetheless, it is there, and it accounts for such things as round ports having less resistance to flow than square ports, area for area, and for the ability of a single port to match the flow of a pair of ports of somewhat larger area. It also accounts for the fact that flow resistance increases in direct proportion with the length of a port, and much of the resistance resulting from the shape of a particular port is due to that shape's creating a thick boundary layer, which becomes literally a plug inside the port.

Generally speaking, boundary layers will be held to minimum depth on surfaces that "rise" (relative to the direction of flow) and gain in thickness on any surface that falls away. Thus, an intake trumpet, for example, should be tapered in slightly from the inlet end to the carburetor—by perhaps 2-3 degrees—in the interest of holding boundary layer thickness to a minimum. In that configuration, it will have appreciably less resistance to flow than a straight, parallel-wall tube. Similarly, transfer ports should diminish in cross-sectional area from their entrance in the crankcase toward their outlet in the cylinder.

These gases also have inertia: once set in motion they tend to remain in motion; when at rest they resist all efforts to get them moving. In practice, this means that there always is a lag between the intake port's opening and

the movement of air in the intake tract. Fortunately, this lag can be amply compensated toward the end of the intake period, when the pressure inside the crankcase has risen to a level that should push part of the charge back out the port—but cannot because of the effect of inertia on the incoming gases. Inertia also has its effect on the flow of gases through the transfer ports and out the exhaust system, but I will deal with that while treating those subjects separately.

These inertia effects are useful, but difficult to manage as something apart from other processes occurring as the engine runs. For example, intake tract length usually is established more with an eye toward resonances than inertia, and its diameter set by the flow rate required by the carburetor to meter properly—balanced against the resistance that attends high gas velocities. Therefore, virtually the only thing we can do about inertia effects is to attempt to find the intake timing that will make maximum use of those provided by an intake system proportioned mostly to suit other requirements.

Resonances are another matter. Sound waves will travel through any elastic medium, such as air, and in their passage they pull together or force apart molecules, just as the similar energy waves travelling through the ocean pull the water into peaks and troughs on its surface. And, as in the ocean, the waves move steadily onward away from their source but the transmitting medium does not. Take, for example, the activity surrounding a single condensation, or positive-pressure wave, as it moves through the air. In its center, molecules have been pulled together, condensed, but as it travels it releases those molecules and compresses others as it reaches them. In the same manner, a rarefaction, or negative-pressure wave, pushes molecules apart. Both waves behave in a curious, but useful way when confined in a tube and the effects of inertia are mixed with them. For one thing, they will be reflected back when reaching the end of the tube—whether that end is open or closed. But at the tube's open end, the wave changes in sign: a condensation is inverted and becomes a rarefaction, and *vice versa;* at the closed end, the wave will be reflected, but retains its sign.

How is all that useful? For example, in the intake system the opening of the intake port exposes the crankcase end of the tract to a partial vacuum, and that in turn sends a rarefaction shooting off toward the opposite, atmospheric, end of the tract. It travels out to the intake bell, inverts in sign to become a condensation, and instantly moves back toward the crankcase—to arrive there as a clump of compressed molecules, which surge into the crankcase to be trapped, if the piston then closes the intake port, as part of the scavenging charge. That effect, overlayed with inertia in the inrushing gases, makes all the difference in getting the job of charging done in two-stroke engines—which provide only an absurdly short time for such chores.

How short a time? That is at the same time one of the least complicated and most depressing calculations you can perform. Let us consider the Yamaha DT-1, which in fully developed configuration had an intake duration of 160-degrees, a transfer duration of 123-degrees, and an exhaust duration of 172-degrees. Yamaha claims a power peak at 7000 rpm. Let's have a look at the actual time, in fractions of a second, available for the completion of these functions. To arrive at these times, use the following formula:

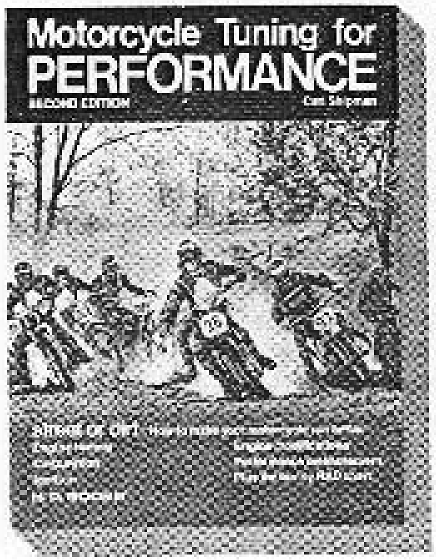
$$T = \frac{60}{N} \times \frac{\theta}{360}$$

Where T is time, in seconds
N is crankshaft speed, in revolutions per minute
θ is port open duration, in degrees
(this formula can be abbreviated to $T = \frac{\theta}{N \times 6}$)
Thus, to find T for the 160-degree intake duration,

$$T = \frac{60}{7000} \times \frac{160}{360}$$
$$T = .0038\text{-sec.}$$

With application of the same formula to the transfer and exhaust periods, we find that the former is open .0029-second, and the latter open .0041-second. Even the longest of these, the exhaust-open duration, is only 41/10,000-second, and that is not very much time in which to empty exhaust gases out of the cylinder. Actually, that particular process is substantially finished in the 29-degrees, or .0007-second, between exhaust- and transfer-opening. In that short period, pressure in the cylinder must fall to something very near atmospheric, or the exhaust gases would force their way down into the crankcase through the transfer ports. Of course, the exhaust gases are provided quite a large aperture by means of which they may make their escape, and that they do so, successfully, is less remarkable than the fact that the fresh charge compressed in a two-stroke engine's crankcase is able to make its way through the far more restricted transfer ports, propelled by a far lower pressure, to refill the cylinder in the extremely brief moment available. It seems nothing short of astonishing that this recharging operation is accomplished in the .0027-sec provided by the Yamaha DT-1's 114-degree transfer period; that the same process takes place in a Yamaha TD-2 engine in only .0017-sec appears a minor miracle. Obviously, divine intervention is not really a factor in the functioning of two-stroke engines, and cylinder recharging is possible simply because the process gets a lot of help from the activities of the exhaust system, gas velocities through the transfer ports have a mean value in the order of 300 ft/sec, and the cross-sectional areas of the ports involved are relatively large as compared with the volume of gases to be transferred.

As it happens, it is possible to calculate correct combinations of port-open times and port areas for any motorcycle engine, at any engine speed. The maximum safe speed for any engine is also calculable, as explained earlier in this chapter, along with expansion chamber dimensions, carburetor size and many other factors influencing both maximum power output and overall power characteristics. It should be noted here that none of the values derived purely from calculations are necessarily optima, and fine adjustments must always be made experimentally, but it is far better to employ the simple formulae presented in the chapters to follow than to attempt a purely-experimental approach. The mathematics involved are not terribly complicated, though sometimes the arithmetic is laborious, and you can use paper and pencil to arrive at a basic engine/pipe combination that will be very near the optimum. Much nearer, in fact, than would be obtained by even the most experienced tuner's unsupported guesswork, and near enough to a fully-developed configuration to minimize the outlay of time and money entailed in the building of a racing engine. You start by determining, mathematically, an upper limit for engine speed, then use more math in establishing a maximum for piston-ring thickness, in establishing all the port dimensions to suit the projected engine speed, in selecting a carburetor, and in designing an expansion chamber. Suitable values for compression ratios, both primary and secondary, are provided in the chapters dealing with crankcase pumping and cylinderheads, respectively, and with the rest of the material included in this book it all adds up to being a fairly complete engine redesign manual for the two-stroke engine-fixated "tuner". My own experience indicates that engines built along the lines suggested here never fail to deliver high specific horsepower (which is more than may be said for any cut-and-try system) even without the benefit of experiment-indicated adjustments. I dislike guesswork, have made a serious effort to eliminate it from my own projects, and am hopeful that the lessons learned – and outlined in this text – will reduce the generally high level of guesswork among most experimenters. If I have forgotten to cover anything, the omission is inadvertent, because my distaste for Speed Secrets is even greater than for guesswork. There is only one "Secret" in the game: to know what you are doing, and to do it thoroughly.



There is a small amount of overlap between Gordon Jennings' *Two-Stroke Tuner's Handbook* and the highly-recommended *Tuning for Performance*, by Carl Shipman, contributing editor to *Dirt Bike* magazine. In the overlap area, it amounts to two points-of-view, well-taken and interesting.

The major utility of these two books is in different areas, both important. As you have seen, Jennings focuses mainly on achieving a balanced array of engine modifications working together to produce more power at more RPM. Then he reminds you that the payoff finally must be extracted through careful adjustment of carburetion, ignition, sparkplug type and heat range.

Shipman starts there. After some elaboration of the basics, *Tuning for Performance* explains the theory of carburetion, ignition and the practical aspects of jetting and timing with lots of pictures and drawings to show you what to do. This applies to any engine—stock or modified.

Tuning for Performance has a comprehensive discussion of tuning for altitude and temperature variations, with a handy chart to help you. The book advocates keeping records—the tuner's secret weapon—and gives you a methodical test procedure leading to the best engine settings. It also gives you a handy chart to use for record-keeping, and shows an example of its use. *Tuning for Performance* is widely used in vocational schools. 176 pages with over 300 illustrations.

As was noted in the chapter of this book dealing with basics, power output from an engine of any given displacement is a function of gas pressure in the cylinder during the power stroke, and the number of power strokes per unit time. Implicit therein is the suggestion that the horsepower ultimately to be had from an engine has little to do with port shapes and port timings, exhaust systems, carburetion or indeed any of the things on which our attention usually is fixed. Why? For one thing, increases in gas pressure bring corresponding increases in heat flow into the piston — and no high-output two-stroke engine can operate beyond its thermal limit. Similarly, you cannot increase the rate at which power strokes occur without increasing crankshaft speeds, with increases in this direction sooner or later taking you beyond the engine's mechanical limit. The horsepower you ultimately will extract from any given engine depends therefore very directly upon your ability to expand those thermal and mechanical limits to the greatest extent possible, and only then to make the most of the territory thus gained.

THE PISTON

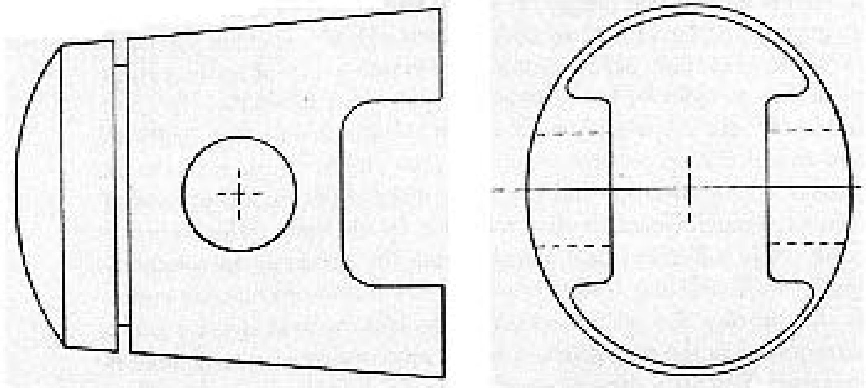
For a very long time subsequent to Dugald Clerk's creation of the two-stroke engine, the thermal limit was the only limit, but it was enough to hold power output from such engines to extremely modest levels. Then, as now, it was primarily a limit imposed by available piston materials. Cast-iron has its advantages in terms of wear resistance, hot-strength and low thermal expansion rates, and it was used quite frequently in the low speed engines of years past. Unfortunately, iron is heavy, and heavy is the last thing you want in a piston — which in modern engines is subjected to accelerations well in excess of 100,000 ft/sec^2. Aluminum, used as the primary constituent in virtually all piston alloys today, is conveniently light, but disagreeably insists on melting at much lower temperatures than that of the fire to which it is directly exposed. Moreover, it loses strength very rapidly with increases in temperature above ambient, so that piston failures do occur at crown temperatures well below the material's melting point. Finally, aluminum is a high expansion-rate metal, which makes a piston made of it a variable-clearance fit in any cylinder. But aluminum is a very light metal, and that alone was enough to recommend it for use in pistons, even though the drawbacks listed were enough to severely limit the specific power outputs attainable with two-stroke engines for a long time.

Aluminum-based piston alloys improved slowly over the years, with the addition of small percentages of, say, copper, to improve their hot-strength, but it was not until means were found to add considerable amounts of silicon that large improvements were made. Today, the best piston alloys contain between 15- and 25-percent silicon, and this addition has all but transformed the "aluminum" piston. Admixtures of silicon in excess of 15-percent not only drastically reduce aluminum's expansion rate, they also effect a proportionate increase in hot-strength and improve the piston's wear-resistant properties. In all of these respects the improvement is large enough to almost exactly equal the percentage gains in horsepower during the years in which aluminum-silicon alloys have been in use. I am inclined to think that most of what we consider to be "modern" improvements in two-stroke engine design – with particular reference to expansion-chamber type exhaust systems – might have been applied as much as fifty years ago had good pistons been available. There was little point in such development work without the aluminum-silicon piston; aluminum or aluminum-copper pistons would melt at specific power outputs well below what we now consider only average.

With all that, high silicon-content piston alloys still are not universally employed. As it happens, such alloys do have their disadvantage, which is that they are difficult to manufacture. Just casting pistons of aluminum-silicon alloy is a task for specialists using specialized equipment; machining the raw castings into finished pistons is an even more formidable task. You may encounter this last difficulty if you have occasion to modify a cylinder cast from the material in question – and you will find that it blunts cutting tools of any kind with remarkable rapidity. For you, that will be an inconvenience; for the mass-producer of pistons it is a disaster, as the need for frequent resharpening of tool bits entails losing output from his machinery while such repairs are made, and it means the expense of the man-hours required for the repairs. Thus, the manufacturer has every reason to restrict the silicon content of the piston alloys he uses to the minimum required by the use to which his engines will be put, which is the reason why Yamaha, for example, uses different alloys for touring and racing pistons.

In point of fact, the Japanese seem to manage high silicon-content pistons better than anyone else, which may well account for their notable superiority in coaxing power from two-stroke motorcycle engines. All of the major Japanese manufacturers employ piston alloys in their touring engines having percentages of silicon high enough to be considered "racing only" in much of the rest of the world. And, sad to say, many of the "racing" pistons being offered by speed equipment manufacturers are inferior in this regard to the ordinary off-the-shelf parts you'll find at your local dealer in Japanese motorcycles. For that reason, I am inclined to use either stock or "GYT-kit" pistons when I am working with engines carrying a "made in Japan" label, rather than waste my money on a specialty replacement. There are, of course, exceptions to this rule, which evolve principally around ring widths, and I will deal with that in due course.

Unless you happen to be a piston manufacturer, there isn't much you can

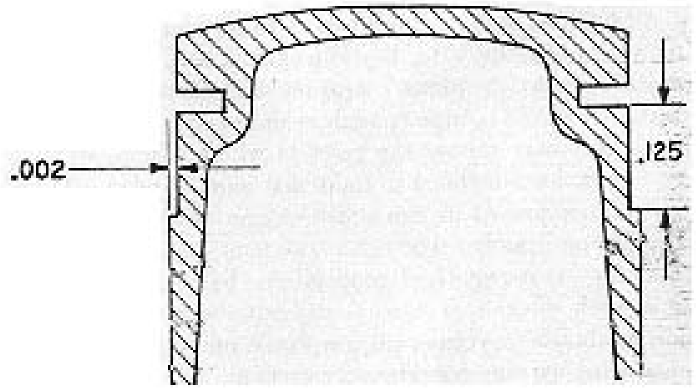


The cam and taper of the piston shown here are exaggerated to illustrate a piston's shape at ambient temperature. When hot, expansion changes the piston to fit the engine's cylindrical bore.

do about piston alloys, beyond seeking out pistons having a high silicon content. Neither is there anything you can do about piston shape – which is most unfortunate, because a piston is not, as it first appears, simply cylindrical. Even with the use of aluminum-silicon alloys, pistons do expand as they are heated, and they do not expand at all evenly. The greatest increase in diameter will occur up at the crown, because that is both the area of maximum mass and highest temperature. So there must be more clearance, measured cold, up at the piston's crown than is required down around the lower skirt. In fact, clearances vary continuously from the piston's crown to the bottom of its skirt – and from side to side, as the piston is elliptical rather than round. Someday, someone may be able, with the help of a computer, to actually calculate all the clearances and ellipse ratios involved; for the present they are decided in a process of trial-and-error by even the most experienced of manufacturers.

Presumably, you will not have the facilities to alter whatever shape your engine's piston(s) may have, but you can vary running clearances by changing cylinder bore diameter. The problem here is one of "How much?" and I regret to say that it is a problem for which there is no convenient solution. Clearances, measured at the piston's maximum diameter, across its thrust faces, may vary from about .002-inch to as much as .007 inch, depending on: the shape and composition of the piston itself; the absolute cylinder bore diameter; the material from which the cylinder is made, as well as its configuration; and the thermal loadings to which the piston will be subjected – which will themselves vary according to gas pressure, fuel mixture, cylinder configuration and the vehicle's rate of motion. Many people have expressed great faith in rules relating clearance to cylinder bore diameter; I have not found the choice to be that simple. If there is a rule, it would be that you can add perhaps .0005- to .001-inch to the clearance recommended by your engine's maker, but even this is a gross over-simplification and I mention it only because it is somewhat better to have too much clearance than too little. In the former, the excessive clearance adversely influences heat transfer from the piston to the relatively cooler cylinder walls and may lead to any of the several unpleasantries associated with overheating the piston, which range from a tendency for oil to become carbonized in the ring grooves, to the appearance of a large hole in the piston crown. Too little clearance will reveal itself in the form of scuffing, or outright seizure – unless the piston is only marginally too tight, in which case the only symptom of distress will be a power loss in the order of 2- to 3-percent.

Often, in modified engines, you will find that the straightforward increase in overall piston clearance by slightly enlarging the cylinder bore is not a complete answer. If the manufacturer has done his work properly, his pistons will, as they expand with temperature, assume a round shape when the engine is hot. Your problem will be that with the modifications you have made, more heat will be forced into the piston's crown, raising its temperature above the level anticipated by the manufacturer, which results in a completely different set of temperature gradients down the length of the piston. Specifically, while the whole piston will assume a diameter slightly larger than that planned for by its maker, the area around the crown will "grow" more than the rest. It will thus be impossible to correct for the altered conditions simply by honing the cylinder bore larger, for if you enlarge the bore enough to provide running clearance for the top of the piston, its skirt will be given too much clearance (leading to rocking, and trouble with the rings). In such cases, which are not the exception, but the rule, the solution is to machine what is called a "clearance band" around the top of the piston. Usually, this band will extend down from the crown to a point about .125-inch below the ring groove, or grooves, and the piston's diameter reduced by perhaps .002-inch over the entire band's width. Although the clearance band is not a particularly clean solution to the piston-expansion problem, it is one that can be applied by anyone with access to a lathe, and it has one advantage over the generally more desirable "pure" contouring of the piston: if a piston with a clearance band seizes partially, aluminum will not be smeared above and below the ring groove – an event which will lock the ring in its groove and upset its ability to seal against gas pressure. In practical terms, this means that the clearance-banded piston will absorb a lot of punishment before it is damaged sufficiently



Stock pistons are not contoured to cope with the expansion produced by temperatures in modified engines and it frequently is necessary to machine a clearance band around the piston as illustrated here.

to cause retirement from a race.

Excessive deep clearance bands must be avoided, for they expose the sealing ring to too much heat, and heat has a devastating effect on the service life of a piston ring. But for these effects, there would be every reason to locate the ring as close to the piston crown as is mechanically possible, because we would then obtain the cleanest opening and closing of the ports; with the ring in its usual position, about .200-inch below the piston crown, there is a tendency for gases to leak down the side of the piston, and the port-opening process thus becomes more gradual than is desirable. The effect is slight, but it is there, and for that reason ring location always is a matter of juggling the conflicting requirements of keeping the ring cool, and obtaining sharp, clean port-opening characteristics. And in most instances, the balance of this compromise will be in favor of the former, for an overheated ring quickly fails. The cause of this failure is twofold: first, excessively high temperatures effectively anneal the ring, and it loses its radial tension; second, an overheated ring warps like a potato chip, and no longer maintains close contact with the bottom of its groove. In both of these cases, the ring's ability to seal is reduced, which allows fire to start leaking down past the ring, and that further raises its temperature – starting a cycle that soon results in outright ring failure.

The single exception to the unpleasantness just described is the L-shaped "Dykes" ring, which also is excepted from the immediate effects of ring-flutter