(described elsewhere). A number of engines have been fitted very successfully with Dykes rings located right at the tops of their pistons, and the dire effects of excessive heating are avoided because the Dykes ring's vertical leg has enough area in contact with the cooler cylinder wall to draw away heat faster than it can be added by the ring's contact with high-temperature gases. At least, that's the way the situation can be, if everything is right. On the other hand, it is worth remembering that many users of Dykes-pattern rings have been forced to fabricate them from stainless alloys to overcome temperature-related troubles, and even then have experienced problems with oil carbonizing in the ring grooves. Probably the best thing to be said for Dykes-pattern rings from the experimenter's viewpoint is that they can be used to overcome the problem of using stock pistons at very much higher than stock crankshaft speeds. If, for example, you would like to use the stock piston, but cannot because it has been grooved for rings 2.0mm thick and you must use 1.5mm rings to avoid ring flutter, you can simply cut a new groove at the top of the piston for a Dykes ring and the problem is solved – unless you encounter some of the other difficulties just discussed.

PISTON RINGS

Of all the problems that can be experienced with a modified engine, those connected with the pistons' rings are the most insidious. Borderline sealing failures can send fire shooting down along the pistons' sides to cause seizures and/or holing of the piston crown that appear to be the result of lean mixture, excessive ignition advance or too-high compression, but are not. These failures are, I suspect, much more frequent than is commonly supposed, for the 2.0mm rings that have become almost standard will begin to flutter when piston acceleration rises above about 60,000 ft/sec² and it is entirely too easy to exceed that limit with a modified touring engine. Therefore, I would again urge you to do your homework before starting a development program with any engine. A formula for predicting the onset of ring flutter is provided in the chapter headed, "Fundamentals", and you may save yourself a lot of grief by determining your engine's red-line with paper and pencil instead of through experimentation. At the same time, I must caution you against simply assuming that very narrow rings are an advantage in all engines. In fact, there is no detectable power difference between the standard 2.0mm ring and the "racing" 1.0mm ring below 7000 rpm, and the wider ring has the advantage of better durability right up to the point where piston acceleration starts it fluttering. Neither is there any advantage, below 7000 rpm, in the use of single-ring pistons. Above that level the lower friction of the single-ring piston begins to make a difference, but in the lower speed ranges you may as well

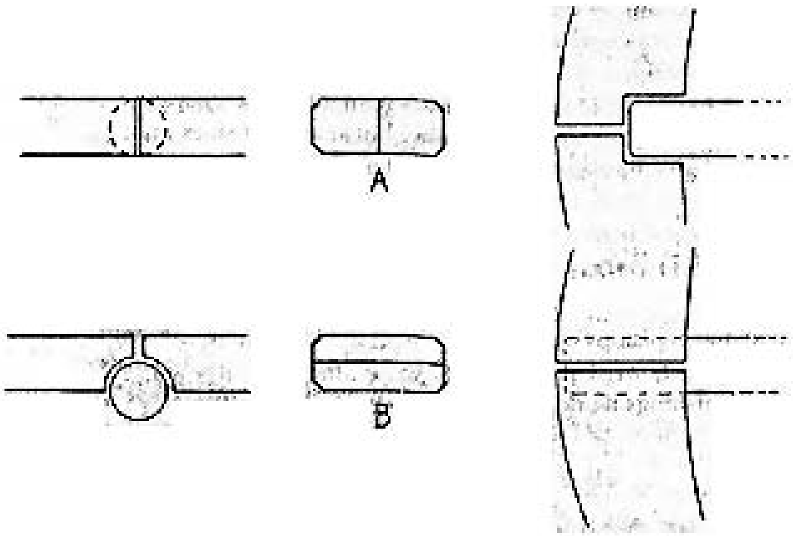
use two-ring pistons and take advantage of their "second line of defense" capability.

Selection of ring-type will usually have been made for you by the piston manufacturer, and my advice is that you do not try to improve upon his judgement, which will be almost impossible in any case. You cannot, obviously, remachine a piston made for 2.0mm rings to take 1.0mm rings – unless you cut a new ring groove above the existing grooves, and that would position your ring perilously close to the piston crown and almost certainly lead to immediate ring failure. The only way around this is to fit a Dykes-pattern ring, right up at the piston crown – as was noted previously. Such modifications can be very successful, if you have the right ring for the application and cut the groove correctly for the ring, but I cannot recommend the procedure simply because there is so much room for error. In general, I think it is far better to replace the stock piston with one fitted with thinner rings – even if the replacement piston is cast of somewhat inferior material, as is often the case. After all, the best of pistons will fail if its rings are not suited to the job it is being asked to perform. On the other hand, rings of less-than-desirable material will perform very well in racing applications if replaced frequently, and if they have not been crudely finished. Much of the ring's ability to function is related to this latter aspect. The ordinary cast-iron ring is fragile, and will shatter very quickly if allowed to flutter, but it will perform entirely satisfactorily if its lower surface is smooth and true, and seals against the bottom of the ring groove. Rings made of nodular cast-iron have the same wear-resistant properties, and are vastly stronger, for which reasons this material is almost universally used. Surface coatings, ranging from chromium to teflon, are often applied to the piston's ring's face, to improve service life and/or prevent scuffing during break-in.

Ring sticking is a problem to be faced with all high-output two-stroke engines. Carburized oil may lock the ring in its groove after a remarkably short period of running if the ring lacks sufficient vertical clearance (usually, from .0015- to .0040-inch) or if the ring is located too near the piston crown. More frequenly, the problem stems from the oil being used for lubrication, and it is most unfortunate that the very oils providing the best lubrication are the ones most likely to cause ring sticking. Castor-based oils, particularly, will build up thick layers of varnish inside the ring groove, unless the oil contains a considerable percentage of detergent chemicals.

Apart from the L-section Dykes ring, most piston rings have a basically rectangular cross-section, but you will find many minor variations on this arrangement. Currently very popular is the "keystone" ring, which has a tapered section, with either the upper or lower surface, or both, sloping away from the ring's outer face. The reason for this primarily is to keep the ring and its groove scrubbed free of carbon and varnish. In four-stroke engines the rings are free to rotate, and do, and their rotation performs this scrubbing. Two-stroke engines nearly always have their rings pinned, to prevent them from rotating and the ring's ends from springing out and becoming trapped in a port. Hence, the need for some other form of scrubbing action. Seldom is the taper in a keystone-type ring more than 7-degrees, and it is all too easy to attempt installing one of them upside-down, so you should give particular attention to the ring's markings. Such markings vary in kind, but without exception they will be on the ring's upper surface.

Another point of trouble can be the ring's locating pin, and if you encounter difficulties with locating pins working loose, the source of the trouble nearly always will be in the exhaust port. The racing engine's very wide exhaust port (width representing, in extreme instances, up to 70-percent of cylinder bore diameter) leaves a lot of the ring's diameter unsupported when the piston is down in the lower half of the cylinder, which allows the ring to bulge out into the port. Making the port opening oval and chamfering its edges will prevent the ring from snagging, as these things ease the ring back into its groove as the piston sweeps back upward. However, while the ring may not snag on the port, it does get stuffed back into its groove fairly rudely, and that may have a very bad effect on the locating pin. On most two-ring pistons, the locating pins are positioned adjacent to the areas of blind cylinder wall between the intake and transfer ports — placed about 90-degrees apart — to provide a long



A locating pin recessed into the ring groove and covered by the ring ends (A) gives slightly better sealing than the arrangement in which half the pin is below the groove and is overlapped by the ring.

path for gas leakage. Thus, when the ring bulges out into the exhaust port and then is stuffed back, the end of the ring is pushed into hard contact with the pin, and after a sufficient number of hard blows (and these accumulate rapidly at, say, 10,000 rpm) the pin begins to loosen and it will gradually enlarge the hole in which it is inserted enough to work completely loose. Then the ring is free to rotate, and it quickly works its way around to catch the end in a port. At risk of seeming immodest, I will admit to having isolated this problem for Yamaha several years ago, and today that firm's racing engines have pistons with locating pins positioned 180-degrees from the exhaust port. Touring engines, which have much narrower exhaust port windows and thus treat their rings more gently, usually benefit from having their two rings' end gaps placed more nearly on opposite sides of the piston, as described before.

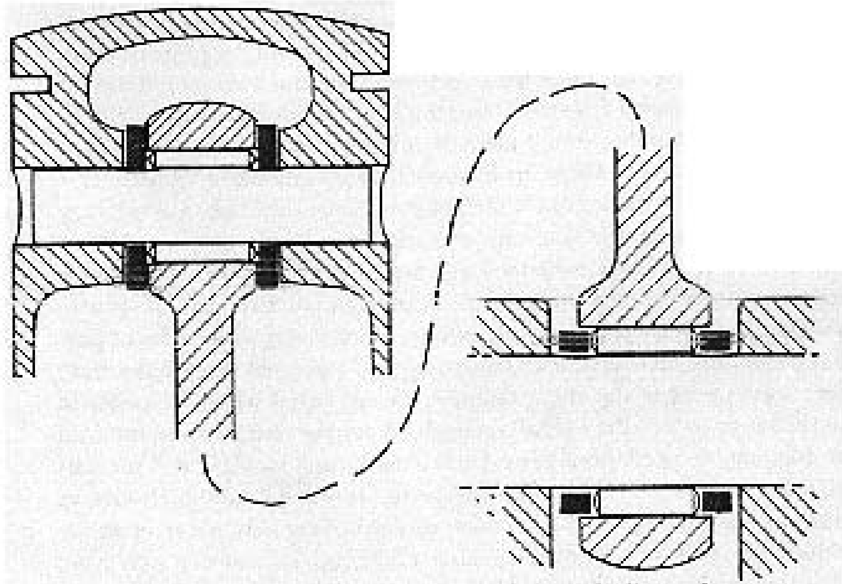
In some racing applications, the standard rings are adequate to the engine speeds anticipated, but overall performance may dictate a much wider-than-stock exhaust port. Then, the "offset" ring-locating pin may prove prone to precisely the sort of loosening and subsequent failure described in the preceding paragraph, which will lead you into a piston modification that can be very tricky: installing a new locating pin in the back of the ring groove. This gets tricky because in many cases the pin will be half-in, half above, the ring groove and it is impossible to drill the hole for a new pin location after the groove is machined. Impossible, unless you cut a small piece of aluminum to exactly fit the ring groove, filling it flush, in which case you drill your hole half in the piston and half in the filler piece. Then you remove the filler and your hole is ready for the pin — which introduces yet another problem: what to use for a pin? Steel wire is a good choice on grounds of strength, but is likely to work loose simply because the aluminum in which it is pressed grows and contracts so much with changes in temperature. A small-diameter "split pin" (which is like a rolled tube) is a better choice, but if suitable sizes are not available, then a pin made of hard brass is at least as good.

WRISTPIN/CRANKPIN BEARINGS

Back in the days when pistons were uniformly poor and two-stroke engines wouldn't be run very fast, wrist pin bearings were almost always a simple brass bushing. Such bushings work very well in four-stroke engines, but lubrication is much less lavish in the crankcase-scavenged two-stroke and added difficulties are created by the essentially uni-directional loads placed upon it, which prevent the piston pin from lifting away from the lower part of the bearing and admitting oil to the load-carrying surfaces. For those reasons, the plain bushing has now almost universally been replaced by "needle" roller bearings, which are more easily penetrated by such oil as is available and in any case need much less oil. This last is of very particular importance in high-output engines, as the heat flowing down from the piston is certain to thin any oil present to a viscosity approaching that of water. But all these difficulties notwithstanding, the needle-roller bearing is wonderfully trouble-free, and if you encounter problems at the hinge between the connecting rod and piston pin, those problems will almost invariably be with breakage of the bearing cage. Given the extremely low rotational speed of the bearing in question, no cage is really needed except to make engine-assembly easier: the cage holds all the needle-rollers in place while the piston is being fitted to the connecting rod. The arrangement certainly makes working on the engine less complicated, but as it happens, the cage becomes the bearing's weakest link. Piston acceleration at high speeds is also applied to the bearing cage, and it may shatter under the strain — which sends a shower of particles from the broken cage and loose needles down into the crankcase. The debris thus liberated invariably gets pumped up through the transfer ports, into the cylinder, and more often than not a roller will get trapped hanging half out of a port by the piston with dire consequences to both.

Yamaha's TD1 was particularly prone to small end bearing cage failures, and I learned the hard way to replace these bearings if I saw over 11,000 rpm on the tachometer even for a moment, for their cages required only a moment's battering before cracks would start to spread and outright disintegration soon followed even if I indulged in no more excursions past the red-line. This difficulty has been overcome with cages made of tougher material; it is possible to accomplish the same thing by using crowded needles and no cage at all, which does require that a washer be fitted on each side of the connecting rod, to take up clearance so that the rollers cannot escape. Getting the thing assembled (with the roller glued in place with grease) is enough to make strong men weep with frustration, but it absolutely insures reliability at this point in the engine and is a measure worth remembering if problems with broken wrist-pin bearing cages do occur.

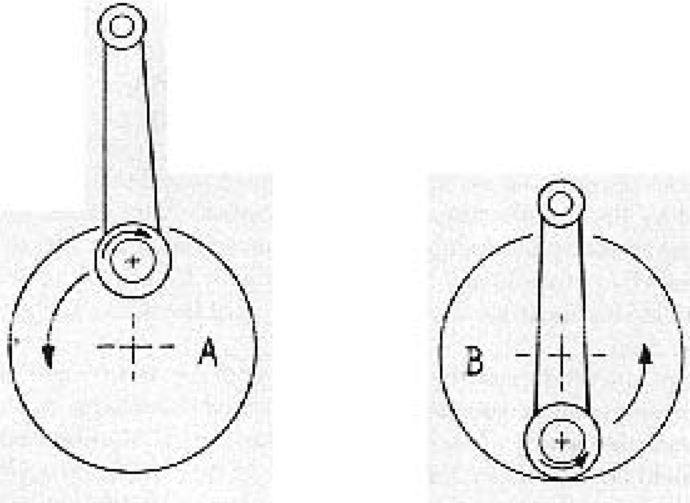
McCulloch, the chain-saw people, have used an arrangement similar to the one just described for years, but they have reasons other than simply working around bearing cage failures at the wrist-pin end of the rod. It was discovered at McCulloch that failures at the crankpin bearing were traceable to the thrust washers most manufacturers of two-stroke engines use to center the rod on the crankpin. These washers usually are made of brass, or steel with a copper coating, and they do not find high rubbing speeds and scanty lubrication at all agreeable. At very high crankshaft speeds, they register their protest by overheating, and this causes a rise in temperature all around the connecting rod's big end, which thins the oil present enough to create yet more friction, more overheating, until at last the thrust washers, roller bearing



Difficulties in assembly have prevented wide use of uncaged rollers and retaining/thrust washers at the wrist pin, but the arrangement reduces friction and improves oiling down at the critical crankpin.

and cage are hot enough to "flash" the oil. At that point, lubrication is nil and friction quickly melts the bearing cage and wears flats on the rollers. McCulloch's engineers reasoned that the point of failure could be pushed upward materially simply by removing the thrust washers, which is what they did. Of course, the connecting rod still had to be centered over the crank, but this task was given to a pair of thrust washers up inside the piston. The improvement in terms of elevating the McCulloch kart engine's maximum crank speed was in the order of 1500 rpm, and it is worth noting that Yamaha borrowed this idea for use in the 17,000 rpm GP engines the company raced in 1968. It is interesting that in those engines, the piston rings were only 0.8mm in thickness.

Crankpin bearing failures also stem from the use of excessively heavy bearing cages. Sheer rotational speed is not enough to burst a cage of such small diameter and mass, but the fact that the cage must accelerate and decelerate, relative to the crankpin as the connecting rod swings, will cause difficulties unless the bearing cage is very light. In effect, the rollers must push the cage up to speed and then slow it, and if the cage has enough inertia it will resist this pushing and pulling enough to skid the rollers — at which point they momentarily become a plain bearing — a job for which they are poorly constituted. The skidding rollers generate a lot of heat, through friction, and the heat leads the bearing into the same deteriorating cycle to outright



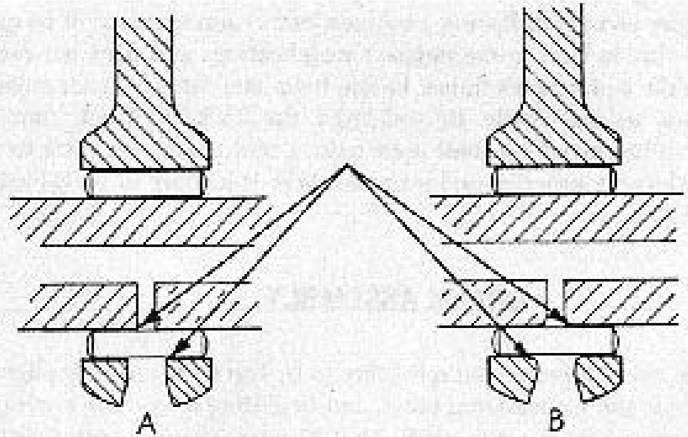
Connecting rod swing effectively increases crankpin bearing rotational speed at TDC (A) and slows it at BDC (B): this speeding and slowing tends to make the bearing rollers skid at high crankshaft speeds.

failure as was outlined for the thrust washers. Most modern engines have steel crankpin bearing cages, copper- or tin-plated to provide a low-friction surface to bear against the rollers, crankpin and connecting rod eye. These replace the phosphor-bronze cages of the recent past – which replaced the inelegant aluminum and brass cages of a yet-earlier era. But the best current "big-end" bearing cages are made of titanium and silver-plated. Experimenters with near-unlimited funds may like to try titanium bearing cages, but when having them made they should know that the bearing retaining slots must be machined with edges parallel to within 1/200 with each other and with the crankpin (assuming a parallel condition between cage and crankpin axis). It is not a job for someone with a bench-vise and a file. On the other hand, if employing silver-plated titanium cages and moving the thrust washers from the crankpin to the piston will elevate your engine's red-line by 2000 rpm, then they clearly will pay dividends in horsepower – if port-timing, etc., is adjusted correspondingly.

Connecting rods should not be lightened, or even polished, unless you intend going all the way in this direction and will finish the job by having the part shot-peened. Forgings acquire a tough skin in the process of being pounded into shape, and I know of instances where connecting rods that were entirely satisfactory in standard condition promptly broke after having been polished. I do think, on the other hand, that there is a margin of safety to be

gained by smoothing off the rough edges where the flash has been sheared away from the forgings. Notches are, in the engineer's language, "stress raisers" and you can do the connecting rod no harm in removing them. Lightening the connecting rod is, however, a poor choice of ways to use one's time, because a rod intended for the loads at, say, 8000 rpm is going to be overstressed at 10,000 rpm and if anything, material should be added to the rod, not removed. On the other hand, one sometimes can improve bearing reliability by opening slightly the oil channels at the ends of the connecting rod. I do not recommend that you actually cut into the bearing surface, but oil delivery to the bearing will be improved by tapering the entry. *Do not* extend the taper all the way to the bearing surface, as the sharp edges thus formed will flake away as the engine runs and cause a bearing failure.

Crankshaft mainbearings seldom are troublesome, except in engines that have been in storage for a long time and have had corrosion at work in these bearings – or unless the bearings have been mishandled. Bearing steels are very tough, but you definitely can pound small pits in the races by injudicious use of a hammer, and pits also can be formed by rusting. Bearings damaged in either fashion should be replaced, as the pits will soon spread and become minor trenches, as a result of an activity called "Brinelling", which actually is a form of work-hardening. The bearing's rollers and races have case-hardened surfaces, but the metal under this thin case is relatively soft, and



The sharp corners at the rod's oiling slot and around the crankpin oil feed hole (A) are subject to flaking that can cause a bearing failure. Small radii ground at these points prevent such problems.

it is compressed and released (at any given point) as the bearing turns under a load. If the load is high enough, or the bearing in service long enough, the repeated compressions will literally fatigue the metal, and tiny particles of the surface will start flaking away – which becomes visible as the "tracking" seen in the races of a worn-out bearing. Any bearing will start flaking at some point in its life; bearings with races damages by rust, etc., will begin such flaking almost immediately. Incidentally, in very highly loaded bearings the flaking may be started by the sharp edges around *any* interruption in the bearing's surface, if the rollers pass over those edges. Oiling slots in the rod's big-end are prone to develop this kind of failure, and the same sort of flaking is sometimes observed around the oil feed holes in the crankpins of engines equipped with "direct-injection" oiling systems, like the Suzukis and Kawasakis. Remove the sharp edges, and you remove the problem – if any. There is sufficient margin of strength in stock production engines so that the problem does not occur; you may find it in the course of reaching for crankspeeds substantially above the stock specification.

Somebody is always telling me about having an engine "balanced", and I always smile nastily when the engine in question has fewer than four cylinders. In point of fact, the single-cylinder motorcycle engine cannot be brought into dynamic balance, for if you counterweight the crankshaft to compensate for the full weight of the piston and rod, you will simply have moved the shaking force from being in-plane with the cylinder axis 90-degrees. "Balancing" one of these engines consists of finding a balance factor, in percentage of reciprocating mass, that is kind to the engine's mainbearings and does not excite resonance in the motorcycle's frame. In-line twin- and three-cylinder engines always have a rocking couple. By and large, the stock crankshaft counterweighting will be correct for most applications, and unless you want to get into a really lengthy experimental program there is nothing to be gained in making changes.

CRANK ASSEMBLY

There are gains in power and reliability to be had from carefully aligning your crankshaft and mainbearing bores, and in getting the cylinder axis precisely perpendicular to the crankshaft. As it happens, there is more variation in production tolerances when the various parts of a crankshaft are made than can comfortably be tolerated in a racing engine. Crankpin holes in flywheels are not all precisely the same distance from the mainshaft axis; factories "select-fit" these parts, and you can be fairly certain that a new crankshaft is true, but if you manage to ruin any of its flywheels, do not assume that a replacement flywheel, selected at random from the nearest parts bin, will be

a *satisfactory* replacement. Crankpin holes, in facing flywheels, should be matched to within .0002-inch with regard to their offset from the mainshaft. If your local source cannot supply a single replacement wheel within that tolerance limit, I strongly urge that you purchase a complete, new crankshaft – with flywheels matched at the factory. And when rebuilding a crankshaft, with new crankpins and bearings, be certain that it is aligned to at least the tolerances suggested by the manufacturer's workshop manual. Also, check your crankcases for mainbearing-bore alignment – and, more important yet, that the cylinder is exactly perpendicular with the crank axis, for any tilting will be reflected in added friction in the bearings (especially at the thrust washers) and in the piston itself.

Do not attempt to second-guess the manufacturer with regard to crankshaft and crankpin bearings unless you have very specialized knowledge in this field or can obtain the advice of someone who is an expert. Mainbearings, particularly, should not be replaced with just anything that will fit, as a very special kind of bearing is employed in these applications, with clearances to accommodate the expansion and contraction of aluminum bearing housings. And the same cautionary note must be added with regard to crankshaft seals, which in the high-speed, two-stroke engine must survive extremes in temperatures and rotational speeds with very scanty lubrication. Not so very long ago, seal failures were common, but now that means have been found to Teflon-coat seal's lips, trouble is usually encountered only when the seals have been damaged in the course of installation. So handle the seals carefully, and pre-coat them with a good high-temperature grease before assembling your engine. You can also improve their reliability somewhat by polishing the area on the mainshafts against which they bear to a glassy finish. The seals themselves will polish the shaft eventually, but at considerable expense to their working life.

By and large, problems with piston, connecting rod bearings, crankshaft and seals can be avoided simply by following the recommendations made in the manufacturer's shop manual. The single exception to this is in the fit between piston and wristpin, for the very high temperatures in a modified engine tend to cause a breakdown in the lubrication between pin and piston. Trouble can be avoided in the racing engine if the wristpin is a light, sliding fit through the piston; it should slip through of its own weight, without forcing, for if it is tight enough so that you have to tap it through with a mallet, you eventually may have to remove it with a hydraulic press. Too-light fits may be corrected by using an old wristpin as a lap, and a dash of some fine, non-imbedding lapping compound to polish out the piston's pin-bore to size.

For the Otto-cycle engine, of which the two-stroke is an example, there is a theoretical level of efficiency, in terms of converting heat into work, referred to in basic engineering texts as "air standard efficiency". In this, it is assumed that the cylinder is filled only with dry air, and heat then added, which ignores the fact that in practice the air contains some moisture and a considerable percentage of hydrocarbon fuel. Even so, this theoretical level of efficiency, calculated against compression ratio, provides a useful yardstick against which actual efficiency can be measured – and it tells us a lot about the effects, on power output, of compression ratio. For example, at a compression ratio of 5:1, air standard efficiency is 47.5-percent, while at 10:1, it is 60.2-percent. That is, of course, a very great gain, and the consequences – measured at an engine's output shaft – are the reason for many experimenters' fixation on "raising the compression". Certainly, increases in compression ratio, which may be accomplished simply by trimming a few thousandths of an inch from the cylinderhead's lower surface, can work minor miracles with an engine's performance.

But higher compression ratios can also bring about a mechanical disaster: improvements in power gained in this manner are purchased at a disproportionate cost in peak cylinder pressure, leading to reduced bearing life and sometimes to an outright failure of a connecting rod or crankpin. Moreover, because the higher pressures are reflected in a proportionately greater side-thrust at the piston, frictional losses are such that net power gains are always less than the improvement one would expect from the calculated air standard efficiency. Finally, heat flow from the combusion gases into the surrounding vessel (piston crown, cylinderhead and the cylinderwalls) rises increasingly sharply with compression ratio, so that a number of thermal-related problems intrude into the already complicated relationship between compression ratio and power.

The worst of these problems is the overheating of the piston crown. A too-high compression ratio will raise piston crown temperatures to the point where heating of the mixture below the piston, in the crankcase, reduces the weight of the charge ultimately trapped in the cylinder during the compression stroke to such extent that net power suffers – no matter what Mr. Otto's air standard efficiency formula may say. And if the compression ratio is high enough, heat input into the piston may raise the crown temperature to the point where detonation, and then pre-ignition occur. These phenomena will, in turn, very quickly further raise piston crown temperature to such extent that the piston material loses enough of its strength to yield to the gas pressure above – the piston crown then becoming either concave (which drops the compression ratio to a tolerable level) or develops a large hole (and that reduces the compression ratio to zero:zero).

Many people have encountered this last effect, and the tuner's one-time