Chapter 7
The Bottom End

THE BOTTOM END is certainly the least glamorous part of the two-stroke engine, and if you are like me it is also the part you would prefer to forget about until something actually goes wrong. Because the two-stroke is so easy to dismantle, the bottom end seems difficult to get at by comparison.

Fortunately the modern day engine has a crankshaft-rod-bearing assembly that in normal service is very reliable and requires little attention. But this is not to say there is nothing that can be done to improve the crankshaft assembly. Your careful attention in this area will not only pick up power and reduce fatigue induced by vibration, it will also decrease the number of crank rebuilds required and lower the cost of being competitive.

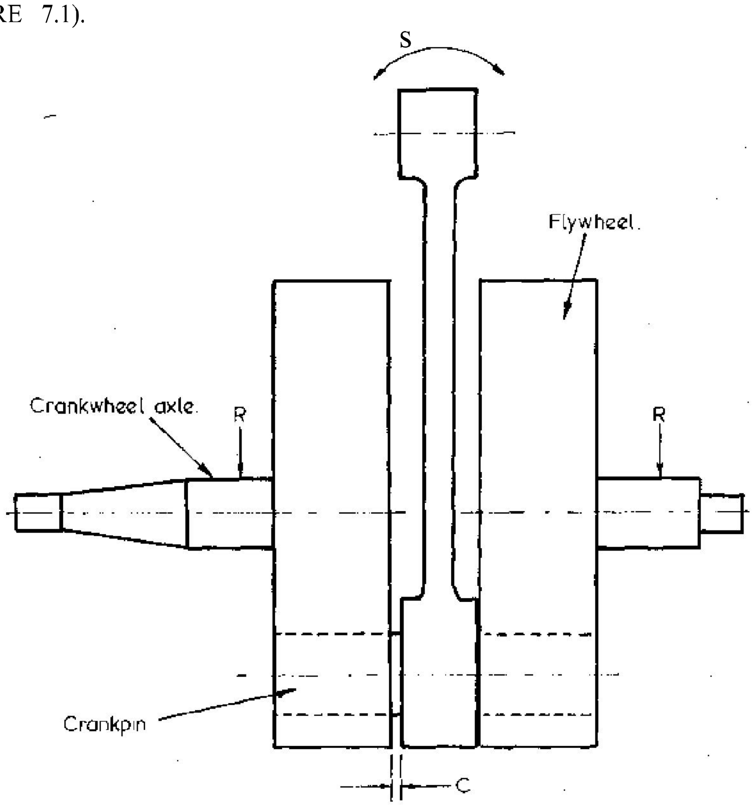
Most two-stroke crankshafts are a pressed together affair. During assembly at the factory, or from useage, the crank can get out of alignment. This sets up vibration in the engine which soaks up power, wrecks the bearings and fatigues you. The only way to overcome the problem is to blueprint the crank assembly. If you have a single cylinder engine this can wait until the bearings or crankpin are due for replacement. However, if your machine is a twin cylinder road racer, I would encourage you to set up properly even brand new crankshafts. When you do this you can be assured of 700 miles trouble free from TZ250 and TZ350 crankshafts.

If you don't have a press, dial gauge and centres, you should take your crankshaft to a reputable firm to have the work done. However, don't assume every motorcycle or engineering shop will do the crankshaft work to the accuracy required. Before you hand your crank over, have a talk with the shop foreman, tell him what you require and why you insist on accuracy. If he wants your job, he will probably show you other crankshafts he has done and prove their accuracy.

The first step is to press the crankpin out and separate the crankwheels. Then have the wheels magnaflux crack tested. Next, check each crankwheel for concentricity. A crankwheel is concentric when the axle is exactly in the centre of the flywheel. Generally, the shafts are not exactly in the centre of the flywheels, which produces an imbalance and vibration. For example, the radius from the shaft's centre to the top of the flywheel might be 2.498ins. The radius to the bottom of the flywheel might be 2.502ins, indicating that the shaft is 0.002in off centre. The dial gauge would show the runout to be 0.004in. What we want is not more than O.OOlin runout, so the crankwheel will require very light machining in the lathe to bring the radius of the flywheel to within 0.0005in of centre.

After the wheels are true, the crankpin holes and crankpin diameter must be checked to ensure an interference fit of 0.002-0.003in per inch of diameter. If the fit is too loose, the crankshaft will not stay in alignment.

When a suitable crankpin is found, fit a new big end bearing on the pin, slip the con-rod on and measure the small end side shake. The amount the small end moves from side to side indicates the big end radial clearance. Unfortunately, even all new parts at times indicate up to 0.055in small end movement, which is very close to what I consider to be the serviceable limit, 0.065in. A bearing, rod, pin combination with 0.030-0.040in play should be sought. This will ensure relatively good big end life (FIGURE 7.1).



C - big-end side clearance (0.5mm to 0.6mm)
R - crankshaft runout (0.04mm maximum)
S - little-end side shake (0.8mm to 1.3mm)

Fig. 7.1 Critical crankshaft measurements.

For the ultimate in big end bearing reliability, a lot of painstaking work is required. Basically, the big end assembly will give the best service if the bearing rollers do not skid on the pin or con rod eye. To achieve this ideal situation you will have to work the big end assembly to several very close tolerances.

Looking at TABLES 7.1 and 7.2 you will note the dimensions in which we are interested. Naturally, we have to select components which not only fall into either of the two selective fit categories dimensionally but which also exhibit true parallelism. If the pin or big end eye, or any of the individual rollers, are not parallel, the rollers will still skid no matter how carefully we match these parts for fit.

TABLE 7.1 Typical big-end bearing radial clearance

Crankpin diameter (mm)	Radial clearance (mm)	
	Minimum	*Maximum*
18	0.023	0.035
20	0.025	0.037
22	0.028	0.040
25	0.031	0.043
27	0.034	0.046
30	0.038	0.050

Note: the above clearances are for high speed racing engines. Low speed road and play bike engines could use clearances 25% less.

TABLE 7.2 Typical big-end assembly tolerances

	Crankpin	Big-end eye	Bearing rollers
Nominal dimension (mm)	20	26	3
Tolerance	−0.006 −0.010	+0.010 +0.020	−0.002 −0.006
Selective fit A	$20^{-0.008}_{-0.010}$	$26^{+0.010}_{+0.015}$	$3^{-0.004}_{-0.006}$
Selective fit B	$20^{-0.006}_{-0.008}$	$26^{+0.015}_{+0.020}$	$3^{-0.002}_{-0.004}$

Also, if we are to avoid skidding rollers, we must ensure that the connecting rod has been machined true. To determine this you will have to make a pair of dummy pins about 100mm long to fit the little end and big end eyes. Measuring between both ends of the dummy pins will determine if the rod is bent or has the eyes machined out of parallel (FIGURE 7.2). Next check that the eyes have been machined in the same plane (i.e., not twisted). To do this set the big end of the rod up with a dummy pin fitted, on a pair of parallel V-blocks. Then, with a dial gauge, measure to see that both ends of the pin fitted in the little end are the same dimension from the surface plate.

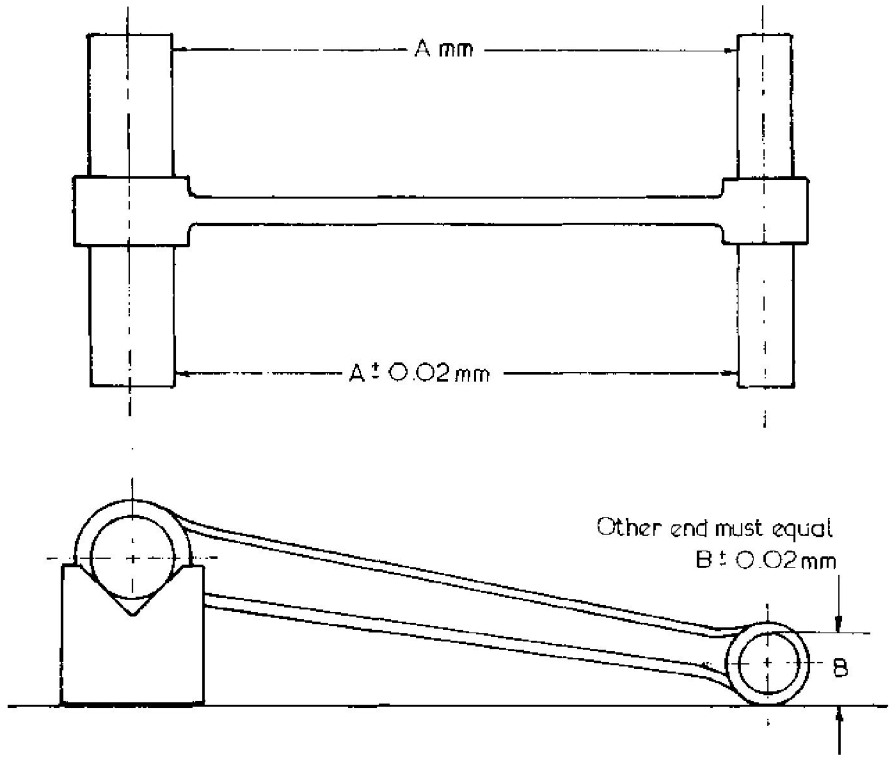


Fig. 7.2 Checking con rod trueness.

In multi-cylinder engines crankshaft balance must be maintained to avoid vibration damage to the crankshaft and bearings. This means that the weight of each big end assembly must be equal, and the weight of each little end assembly must be equal. The big end assembly is made up of the crankpin, big end bearing and thrust washers, and the rod big end. The little end assembly comprises the rod little end, the little end bearing and thrust washers, and piston pin. Unfortunately, few tuners have the equipment to do this balancing themselves, so this usually means that all these components must be sent to some automotive firm for balancing. If this is true in your case, be sure to pack each con rod assembly in a separate plastic bag and instruct the firm doing the balancing that under no circumstances are parts to be swapped from one assembly to another, otherwise all the time spent on obtaining proper big end tolerances will have been wasted.

The pistons, of course, will also have to be balanced, using either an accurate pair of laboratory scales or a simple beam balance. When the lightest piston is found, remove metal from inside the piston skirt and around the pin bosses to reduce the weight of the other pistons to within 1 gram of the lightest piston.

Whenever the big end or main bearings are replaced, don't just use any bearing which will fit. The loads experienced by the bearings in two-stroke engines demand the use of high quality parts if reliability is to be maintained. Therefore only those bearings equivalent to, or superior to the original components, should be utilised.

If you wish to use bearings better than those fitted as standard, you may be able to obtain a suitable replacement from the German INA bearing company. Their two-stroke bearings are the best available. Try to get main bearings with fatigue resistant plastic or fibre cages rather than riveted steel cages which seem prone to cracking up.

Main bearings with plastic cages demand plenty of lubrication to enable cool running. If the bearing overheats the plastic cage will distort or melt, causing bearing failure. To improve lubrication you may have to drill an oil feed hole in the crankcase to each bearing, similar to that illustrated in FIGURE 7.3. The hole should be about $\frac{5}{32}$in diameter, drilled from the transfer slot or the barrel spigot recess to the main bearing housing.

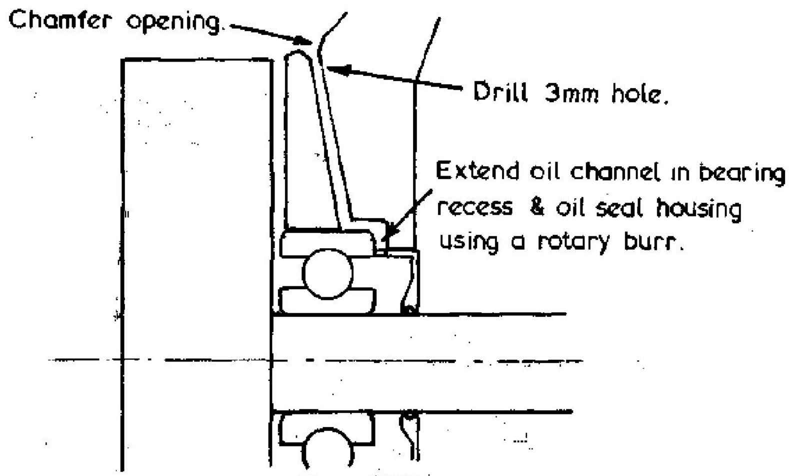


Fig. 7.3 Main bearing oil feed.

The big end bearing must be as light as possible, otherwise the inertia generated by the swinging of the con rod as it passes top and bottom dead centre will cause the rollers to skid and over-heat the bearing and rod big end. A lightweight bearing can be accelerated and decelerated quickly, but a heavy bearing will continue to rotate at a more constant speed, rather than staying synchronised with the relative rotational speed of the crankpin.

Most people are surprised to know just how much influence the angular swing of the con rod has on the rotational speed of the big end bearing. Normally, the bearing should rotate at half the relative crankshaft rpm. On the surface it would appear that a big end bearing in a motor spinning at 11,000rpm would be rotating at 5,500rpm. However, when you look at FIGURE 7.4 you can see that this is not so. At TDC the angular swing of the rod is in the opposite direction to rotation, but in the same direction at BDC. With a 2 to 1 rod length to stroke ratio (eg: engine stroke 54mm; rod length centre to centre 108mm) the instantaneous rotational speed of the rod in relation to the crankpin is 25% greater or less than the crank speed. Thus, at an engine speed of 11,000rpm the relative rotational speed will be 13,750rpm at TDC, and 8,250rpm at BDC. Remembering that the bearing rotates at half these speeds, we can see that its

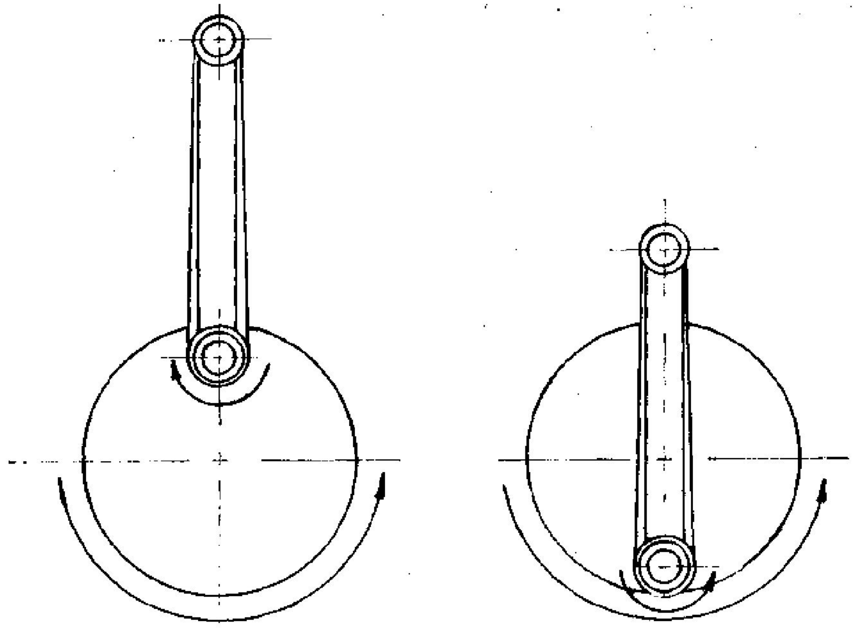


Fig. 7.4 Big-end bearing speed oscillates.

revolution rate must drop from 6,875rpm to 4,125rpm, and increase back again to 6,875rpm twice per crankshaft revolution. If the bearing has enough weight, it will resist this rapid oscillation, forcing the rollers to skid.

Most modern two-strokes have steel big end bearing cages plated with tin or copper, to provide a low friction surface. These can be beneficially replaced by very light INA bearings with a special lightweight silver plated cage. Such a move could raise the red line speed of a street engine modified for road racing by 2000rpm. The Yamaha RD400 is very popular for road racing, but its standard bearings are not up to the task. The simple solution is to substitute big end bearings out of Yamaha's TZ250 road racer. With these bearings, the RD400 will run reliably for hours at 10,000rpm.

From time to time a few tuners get hooked on the fad to lighten the crankshaft. They feel the flywheels should be machined to a 'T' or 'V' shape to reduce their weight and increase engine acceleration. Acceleration will increase, but you will have to change gears so much more that the machine will be slower around the track.

This is not to say all crankshaft lightening is a no no, as a very small number of engines will benefit from a moderate reduction in rotating mass. Generally, we can forget about the majority of Japanese engines, as these already have very light flywheels. The only exceptions would be some single cylinder 250cc motocross engines and also the Yamaha RD400 when these are modified for road racing. The amount of metal removed is quite small, usually not more than 6oz from the inside of each flywheel. This reduces the weight of the RD400 crank, for example, by 1.5lbs.

Before the crank is reassembled, the rods (even if new) must be crack tested. If the engine has a history of rod failures, the rods should be polished along the beams and then shot peened. Also polish and radius any corners formed by the oiling slots in the big-end (FIGURE 7.5).

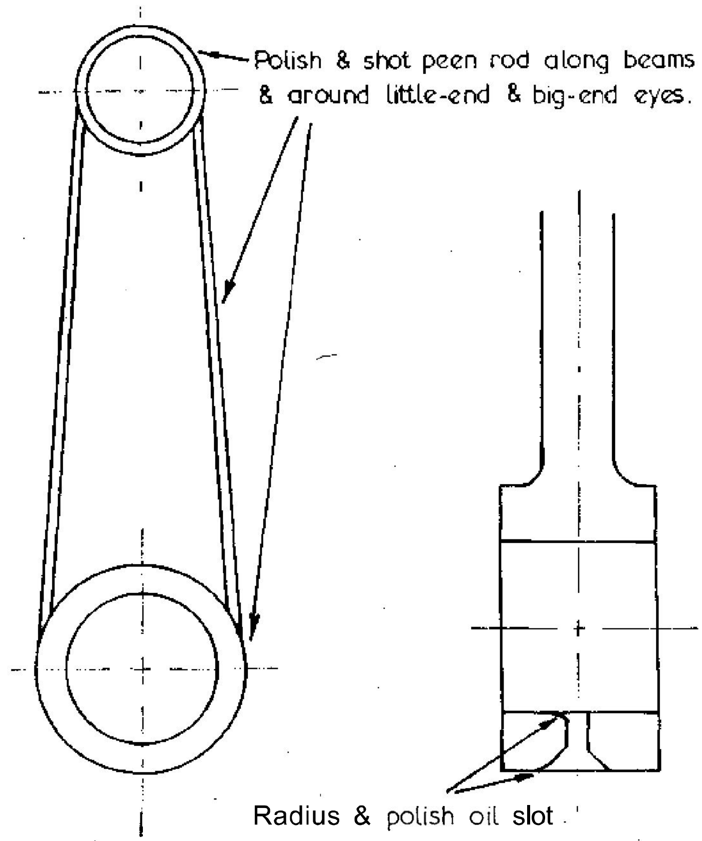


Fig. 7.5 Con rod modification.

The tough skin formed on the con rod by forging gives it much of its strength and fatigue resistance. Therefore the rod should never be polished unless you intend to follow up with shot peening to create another work toughened skin. I consider it a waste of time polishing the entire rod.

If you have a look at a rod you will see, along its edges, a rough band where metal appears to have been sawn away. That is where the excess metal called flash was squeezed out from between the forging dies when the rod was being made. Later, most of the flash is trimmed off, but a bead is left, as you can see. Of course, there is no hard skin along this ridge, in fact its roughness is a stress raiser, so the ridge should be removed with a sanding belt. Give the entire beam a polish with fine emery cloth and then follow up with buffing and shot peening.

After the rods have been prepared and the crankpins and bearings matched, the crankwheels can be reassembled. Scrupulous cleanliness is essential, and care should be taken to ensure the wheels are started on the pin as accurately as possible. Use a straight edge across the wheels to check this. If the pin is shouldered, the wheels are pressed hard home, but if a straight pin is used, the necessary rod side clearance must be maintained by inserting two feeler strips of the appropriate thickness, usually 0.25-0.3mm, on either side of the big-end and pressing until the strips are just ensnared.

The alignment should be checked between centres, using a dial gauge in contact with the bearing seat of a crankwheel axle. Any runout is eliminated by holding one wheel and striking the other with a copper or lead hammer. Runout must be kept to a maximum of 0.0015in, but 0.001in is preferable (FIGURE 7.1).

Some people weld cranks to keep them in alignment but I do not agree with this practice. Welding hardens steel and makes it prone to fatigue fractures. Therefore, I use Loctite on the fit between the crankpin and crankwheels. Apply a small amount of Loctite to the pin, and a larger amount in the crankwheel holes, before pressing the shaft together. Take care that you do not allow Loctite into the bearing.

Before the crankshaft is refitted, the crankcase will require some reworking to reduce friction in the engine and increase piston and ring life. At all times the piston should be perpendicular to the crankshaft, but production tolerances being what they are, this is seldom the case. For the piston to be 90° to the crankshaft the con rod must be straight, the cylinder bore axis must be at 90° to the base of the barrel, and the top of the crankcases must be parallel with the crankshaft centre.

To check the parallelism of the crankcase first loosely bolt the two halves together. Then fit the cylinder and evenly tension the cylinder retaining bolts to align the case halves. Next tension the crankcase bolts. With this done, the cylinder can be removed and the cases measured for trueness. The simplest way to effect this with accuracy is to fit a mandrel in the main bearings and take a measurement from the mandrel to a straight edge laid across the top of the crankcase (FIGURE 7.6). The dimension between the mandrel and straight edge must not differ by more than 0.001in from one side of the crankcase to the other. Usually, it will be found necessary to machine the top of the cases to bring them into line with the crank.

When the crankcase has been trued, the crank, together with new main bearings and seals, can be fitted. Take the time to lubricate the seals and bearings before fitting the crankshaft. A dry start will quickly wreck any engine.

Engines with horizontally-split crankcases can experience problems with the main bearings attempting to spin in their housings. The TZ250 Yamaha is particularly prone to this. The outer races do have little pips on them that fit into a small cavity where the case halves join, but this hasn't stopped the trouble. About the best move is to go over all the holes in the case faces and chamfer them. Studs tend to pull metal up around their threads and this can stop the cases mating tightly. When the shaft is fitted apply some Loctite to retain the bearings in the case.

Possibly the part of a two-stroke that takes unequalled abuse and gives the tuner the most trouble is the piston. Fortunately piston technology is constantly moving ahead, and piston related unreliability can, to a large degree, be eliminated by regular piston replacement and correct installation of the part in the first instance.

The largest improvement to be made to pistons came when the means were discovered for adding large quantities of silicon to the aluminium alloy. This has drastically reduced the piston expansion rate, minimising the incidence of seizure. Silicon also imparts more strength to aluminium at high temperatures and increases

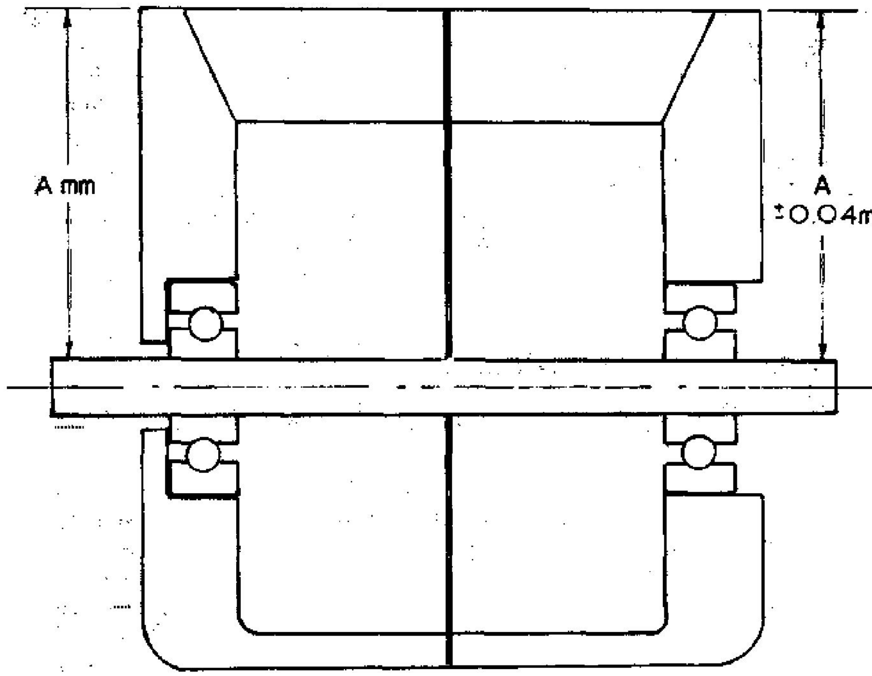


Fig. 7.6 Checking crankcase parallelism.

wear resistance.

Quality pistons for competition use generally contain around 18-22% silicon. Unfortunately, there are pistons being sold that do not contain very much silicon at all, even though the manufacturers claim they are racing pistons. This occurs because high silicon content pistons are difficult to manufacture and expensive to machine. Consequently, I stick with the manufacturer's original pistons if the engine was designed for road racing, enduro, motocross etc. in the first place. I have usually found original pistons to be of good quality. This particularly applies to Japanese pistons; they seem to be able to produce a very good product.

There is one area for concern with standard replacement pistons which must always be checked; a large percentage are cracked from new. The best insurance against this is to have all your new pistons Zy-Glo crack tested. If you can't find an engineering shop with Zy-Glo equipment, check out aircraft repair workshops in your area.

Another problem with standard replacement pistons is that many do not have any circlip extractor slots. This means that only tail-type wire circlips can be used and unfortunately this type of circlip wrecks engines. The constant rubbing of the gudgeon pin against the circlip wears through the tail, allowing it to drop into the cylinder, scoring the bore and possibly seizing the motor. If tail-type circlips are replaced regularly, say after every second race meeting, this kind of damage can be avoided.

A better solution is to machine extractor slots into the piston so that tailless circlips (or tail-type circlips with the tail cut off) can be fitted (FIGURE 7.7). The slot need only be ⅛in. wide to allow a small electrical screwdriver or the point of a scriber to fit under the circlip so that it can be flicked out. It should be cut in the position shown, using a small round key file or a ⅛in. dia. mounted grinding tip. Do not use a hacksaw blade or three cornered file to make the extractor slot, as the abrupt corner will form a stress point and eventually cause the piston to crack.

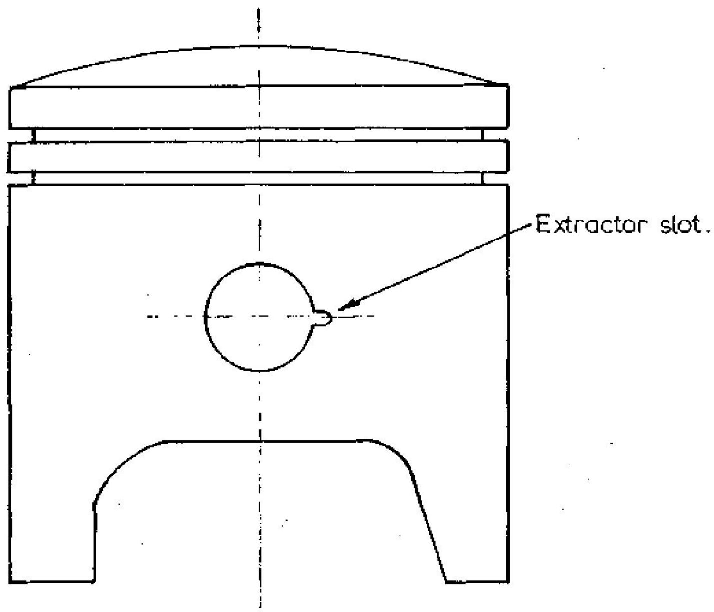


Fig. 7.7 Circlip extractor slot.

If you have an early model Yamaha you will not have to worry about machining extractor slots into the pistons. From the G model on, Yamaha pistons have been made with these slots, so when piston replacement is necessary substitute late model pistons.

We tend to think of pistons as being round, but actually the skirt is cam ground an oval shape. The piston also tapers from top to bottom (FIGURE 7.8). Both ovality and taper are necessary to prevent seizure. The top of the piston gets twice as hot as the bottom of the skirt, therefore it expands more and, due to the extra material around the pin bosses, more heat is directed to this area, elongating the piston across the piston pin axis. To compensate for this, the piston is also ground oval. Therefore you must be careful to measure piston clearance only on the thrust faces, and at the bottom of the skirt.

Before the piston is fitted, there are several clearance checks to be made. The first of these is the fit of the piston pin. It should be an easy slide fit, slipping through the piston under its own weight. A tight pin is to be avoided as this will overload the sides of the piston when the engine is running. At high engine speeds the crankshaft tends to whip and, if the pin is tight, this load will be transferred to the side of the piston, possibly distorting it and causing seizure.

A tight pin may also change the expansion characteristics of the piston and cause