From End to End

Joining piston to crankshaft, the connecting rod's small end bearing fits the piston's gudgeon pin, and the big end bears on the crankshaft

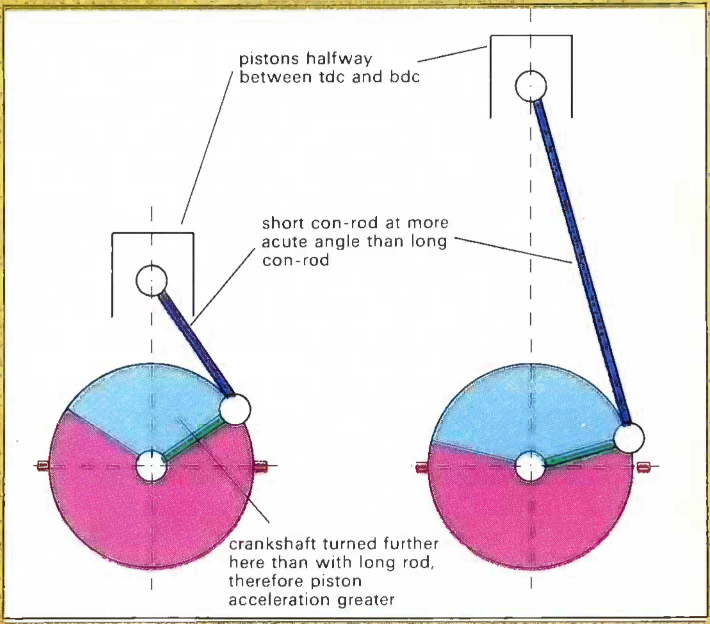
The conventions of engine construction (comprising piston, connecting rod, crankshaft and crankcase as the architectural essentials) were established as long ago as 1780 and have since been more supported by convenience than by authority, their invention having been patented by one James Pickard, a Birmingham button maker. As he arranged things, so the manufacturers of motor cycle power units have been content to arrange them ever since, with but few Wankels and other revolutionaries crying in the wilderness. Thus, while the motion of an engine's piston is strictly linear, that of the crankshaft is rotational; so that of the connecting rod (commonly abbreviated as con-rod or conrod) which links them is translatory. At the little end, the motion of the rod is in a straight line, but at the big end the motion is circular – and it is all very unfortunate, for this swinging action of the conrod is responsible for many of the piston engine's problems. Vibrations, bearing restrictions, difficulties in breathing and burning, even ignition and valve timing, are all sensitive to the proportions of this vital link.

In its construction, the con-rod has few mysteries. The little end (the bearing eye in which the piston's gudgeon pin is a very precise fit) has its dimensions more or less fixed by piston design. The big end (the portion which bears on the crankshaft) is usually split for convenience in manufacture and assembly, for the stronger and lighter one-piece rod demands the expense of a built-up crankshaft. There is a conflict of ideals involved in settling this very basic choice: the one-piece rod is better than the split rod, but a one-piece crankshaft is usually better than a built-up one, notably in its stiffness. So long as the conventional motor cycle engine had only one cylinder or, failing that, was a simple V-twin, a built-up crankshaft could be made amply stiff, and was actually cheaper than a one-piece forging in those days when metallurgy had not yet given us irons suitable for casting such a component. Today, beam and torsional stiffness requirements make a built-up crankshaft undesirable, and it is only adopted when the designer feels convinced of the desirability of observing the roller-bearing usage that is likewise a motor cycling tradition.

Rolling-element bearings for the main journals of the crankshaft are all very well; but their use as big-end bearings (which is what encourages the use of the one-piece rod) is erroneous, because the angular swing of the con rod as it moves makes the rollers skid and wear rapidly. Only in the more rudimentary of two-stroke engines (the only two-strokes found in motor cycles) where the crankcase is washed with

Above right: variations on a theme: from left to right, con-rods from an RD250 Yamaha, an FS1-E Yamaha, a 1910 Premier and a Suzuki 250 single. The length of the rod has a special significance as outlined below

Inset: with the same stroke in each case, con-rod angularity varies in relation to the rod's length

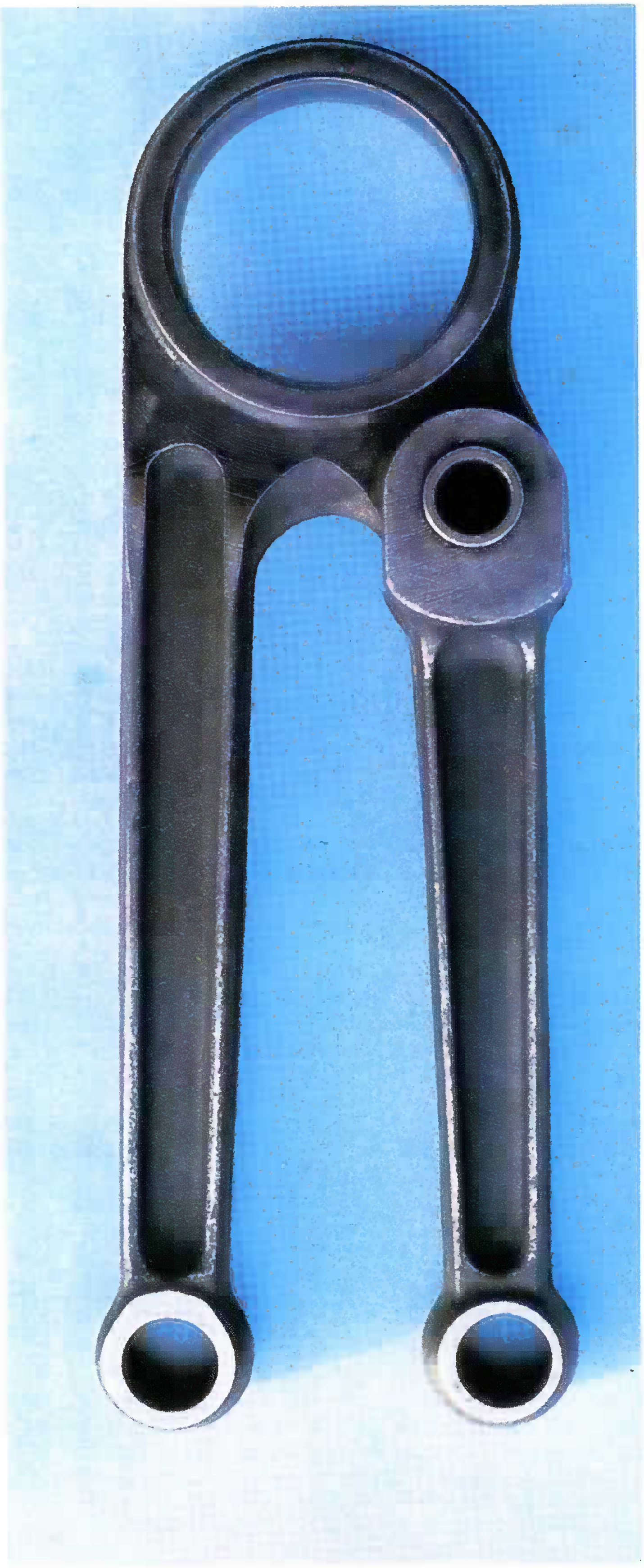


draughts of petrol and air, is the roller bearing a necessary evil. In fact, there is no need to associate the plain bearing with a split big-end eye; but if it is to be split, there are preferred ways of doing it. First, the split line should be perpendicular to the main axis of the rod, and in motor cycle practice this is usually so. Secondly, the faces to be mated at the split should be serrated rather than flat (and this requirement is less commonly met) in order not only to ensure true alignment of the bolt holes and preserve the bolts (or studs) from shear, but also to increase the mating area through which heat may be transferred from rod to cap, and also to reduce the danger of fretting corrosion between the mating surfaces. More important is the danger of the rod being weakened by the machined flat or counterbore which may locate the D-shaped heads of the bolts: these cause sudden changes in section that act as stress-raisers in what is already a very highly stressed area.

As for the shank between big and little ends, few designers bother to make it other than in the H-section which is easiest to forge. For a given cross-sectional area of metal (and therefore a given weight), the H-section beam is obviously stiffer than a symmetrically solid bar. Equally obviously, a tubular shank would be stiffer still, and such rods were once looked upon with favour; however, the con-rod is not a simple compressive strut but has a flailing action that imposes a greater need for rigidity in the plane of its motion than in the plane of the crankshaft, so the tube of the shank would have to be elliptical rather than circular. Scarcely anybody has bothered; but many makers of two-stroke engines, and BMW perhaps alone among the four-strokes until they adopted car-type conrods in 1969, reached a reasonable approximation with a solid shank of elliptical section. This distributes its mass so as to give the preferred orientation of beam stiffness, and it also minimises the aerodynamic drag of a rod moving fast through the air trapped in the crankcase – a form of parasitic loss that has recently been recognised as significant in high-speed engines. Quite apart from this drag, the highly tuned two-stroke cannot afford to have its carefully cultivated flow of fresh charge through the crankcase disturbed by a great paddle of a conrod, which is why the rods of high-performance two-strokes look so improbably sharp and slender. As it happens, the con-rod of a two-stroke has an easier task than its four-stroke equivalent: instead of suffering reversals of load from compression to tension in each complete cycle, it is virtually always in compression – and that is why it is safe to cut away the top of the little-end eye to improve the access of lubricating oil to its bearing.

Nevertheless, the knife-blade rod shank is not ideal, and if the loads imposed on the shank made an H-beam desirable, the web of the H should not lie in the plane of the rod's swing, even though that is how it is almost always made. Charles Lafayette Taylor and Roland Cross demonstrated decades ago, and the author several years ago, that the ribs should be extended generously in that plane and the web should lie between them in the plane of the crankshaft; very recently, some racing rods have been made in this style by Cimelli in Italy and have proved very successful – although much of their merit lies in the excellence of their manufacture, notably the overall shot peening of the surfaces of the forgings from which they are made, inducing a compressive stress in the surface of the steel.

Most con-rods are machined from steel forgings, although a few astute car makers cast them instead. During a long production run, forging dies wear so much that the dimensions and weights of the finished rods vary a lot, making large surplus masses of metal necessary if the danger of an undersized rod is to be avoided. Appropriate amounts may then be ground or milled away, to reduce all rods to equal weight, and it is common to see large projecting bosses of superfluous metal projecting from the big-end cap and the little end to allow for



Above: the 1948 EMC 350 used a split-single-cylinder, twin-piston, two-stroke engine. The two cylinders – one for exhaust and inlet and one for mixture transfer, sharing a common combustion chamber – were in a line at right angles to the axis of the crankshaft and parallel to each other, necessitating the use of this articulated, forked con-rod. Similar rods were used in radial and rotary engines.

such balancing operations. A cast rod is naturally more substantial because its metal is less strong, but it is dimensionally more consistent, so the big balancing lugs can be omitted, freeing all the material for useful work. Thus, a cast rod can actually be lighter than a forged one – and lightness is of tremendous importance, affecting engine performance and balance. Many motor cycle engines have successfully employed aluminium-alloy rods, and racing engines built regardless of cost may exploit titanium; in mass produced machines, weight saving is generally pursued in dimensional design rather than in choice of material.

One of the most vital dimensions is the length of the con rod. A short rod is obviously lighter and stiffer than a long one and, by allowing the whole engine to be reduced in height (or width, as in the case of flat twins such as certain racing BMWs), it encourages further savings in material. On the other hand, a long rod promotes smoother running and allows higher engine speeds to be reached for a given level of mechanical stress. This is because of the different amounts of angular swing of long and short rods in engines of identical piston stroke. If the con-rod could be infinitely long, its angular movement would be infinitely small, the motion of the piston up and down the cylinder would be perfectly harmonic (that is, sinusoidal) and there would be no problems of secondary vibration or bearing velocity change to impede the designer. As things are, the con rod must be of finite and practical length: the conventional distance between the centres of the little and big ends is twice the piston stroke, reducing to 1.8 times as the years pass.

This gives a fair compromise between conflicting requirements. Were the rod shorter, the whole engine could be shorter and lighter, and the piston would travel faster on its approach to, and recession from, top dead centre – with advantages in better breathing on the exhaust and inlet strokes, and better conservation of combustion heat during the expansion phase. If the rod were longer, the maximum piston acceleration at a given crankshaft speed would be less, inertia loads on the piston, rings, rod and crankpin would be correspondingly less, secondary vibrational forces would be reduced (because there would be less difference between the rates of piston acceleration on the upward and downward strokes) and frictional losses between piston and cylinder would be reduced (because of the reduction of conrod angular swing). The combustion process might also benefit from the more nearly constant volume of the combustion space when the piston was in the region of top dead centre. Note that mean piston acceleration is irrelevant, as well as virtually incalculable by direct methods; what matters is the maximum acceleration, which can be derived from the formula:

$$\frac{N^2 S}{2189}\left(1 + \frac{1}{2R}\right) \text{ ft per sec per sec}$$

where N is crankshaft rpm, S is the stroke in inches and R is the ratio of con rod length to stroke. This allows some interesting comparisons to be drawn – for example, between a couple of fairly coeval big twins, the Norton 750 Commando and the BMW R75. The Norton is a long-stroke engine with short rods ($R = 1.66$), the BMW a short-stroke engine with relatively longer rods ($R = 1.85$), and both of them generate maximum torque at 5000rpm. In doing so, the BMW's pistons reach peaks of about 40,330 ft per sec per sec and those of the Norton 52,057. It may be easier to think of these accelerations if they are expressed in terms of gravitational acceleration or *g*, in which case the BMW piston reaches 1253*g* and the Norton 1617*g* – in other words, it momentarily weighs 1617 times as much as when it is stationary, and that is how the gudgeon pin feels it, too. The R75 engine is safe up to 7000rpm, at which point the maximum piston acceleration climbs to 2456*g*, but if you run the Commando up to 7000, its piston reaches 3170*g*, which illustrates the importance of minimising the weight not only of the piston but also of the connecting rod itself. LJKS

Below: as well as the overall dimensions of a con-rod, its cross section is also an important factor in endowing it with the desirable characteristics of strength and lightness. The upper rod is of elliptical section and is from a 1969 BMW; the lower, H-section rod is from a later BMW