Crankshaft Construction and Engine Balance

Two Generations of Motorcyclists Have Known LAURENCE HARTLEY as a Tuner of High-performance Engines. Here, he Discusses Some of his Methods in an Interview with Bruce Main-Smith

Laurence Hartley busy in his Plumstead workshop.

THE basic purpose of a crankshaft is to convert the purely linear motion of the piston into rotary motion, and at the same time to return the piston for the start of a fresh operating cycle. The majority of motorcycle engines are either of orthodox single-cylinder or parallel-twin layout, and have the crankshaft disposed across the frame (the twins being, in effect, duplicated singles with pistons moving together). As interest will, in the main, be vested in these general types, I propose for this two-part article to confine myself mainly to experiences of them only.

Crankshaft construction can be divided into five broad classifications (see drawing):

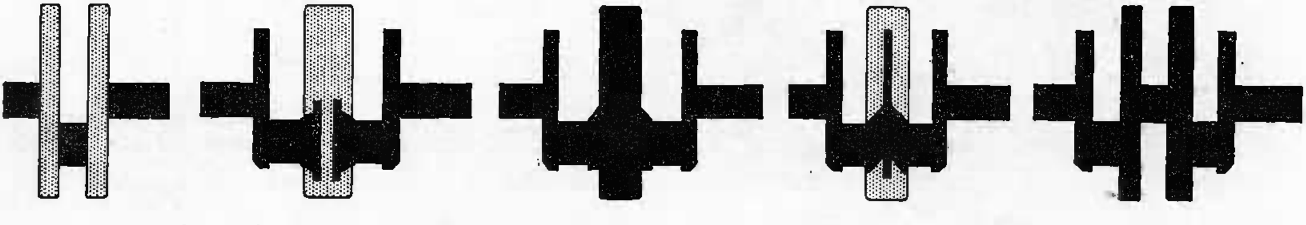
1. Built-up; two flywheels united by a single crankpin and serving either one cylinder, as in the case of the B.S.A. "Gold Star," or two, as in the Vincent "Rapide."
2. Built-up; two big-end journals, joined to mainshafts by webs and terminating in flanges, united by a central flywheel (e.g., Norton "Dominator").
3. One-piece; construction similar to (2), but with the central flywheel cast and machined with the remainder of the crankshaft (e.g., Royal Enfield "Meteor").
4. Built-up; similar in many respects to (3), but with a central flywheel ring bolted on between the two throws (e.g., B.S.A., Ariel and Triumph twins).
5. One-piece; similar to (3), but with the cast-in central flywheel "divided" to provide a centre main bearing (i.e., the webs joining the crankpins to the centre main bearing become, in effect, radically enlarged to take the form of twin flywheels). Examples: A.J.S. and Matchless twins.

The crankshaft is, of course, mounted in a crankcase and supported on bearings. The number, type and disposition of these bearings have no direct influence on my work in improving the accuracy of a crankshaft and connecting-rod assembly. I must assume, in preparing a customer-supplied assembly, that the bearings which support and take the load are in good, and preferably new, condition; that each is in line with the other, and also that the lateral plane of the crankshaft is parallel to that of the crankcase mouth. Errors under the first of these headings can be readily rectified at some small expense by the private owner willing to fit new parts; errors under the second stem from a faulty machining operation not detected during inspection, and from subsequent permanent deformation as a result of usage. If I have the crankcases, I check them and rectify by machining, if necessary.

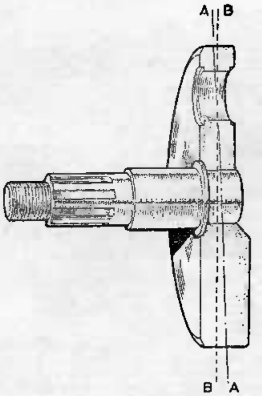
Consider first the case of the built-up flywheel and crankpin assembly, such as is used in the majority of present-day singles. A disc of steel is machined at its centre to accept a shaft. The shaft spins in one or two bearings, and to it is attached the driving sprocket. Similarly, another disc and shaft are used for the timing side of the engine. The disc obviously must be exactly at 90° to the shaft in all positions. If it is not, then at low revs. the shaft will run true and the disc will wobble. At speed, the enormous centrifugal and gyroscopic forces (and they are proportional to the square of the speed) will make the disc try to revolve perfectly true; then, either the shaft wobbles or, if both run true, the shaft, or its joint with the disc-shaped flywheel, must flex. Both types of flexure cause vibration and both will lead eventually to failure. An increase in the rigidity of the bearing housing may only aggravate the trouble.

Now, although a flywheel may be true with the shaft, it may still wobble when rotated. The forces set up when a steel disc weighing perhaps 8-11 lb. and 10 in. in diameter, is revolved at any speed above, say, 1,500 r.p.m. are tremendous; nevertheless, a flywheel which is geometrically true with the shaft will run smoothly *if the disc is a section of a "solid" cylinder.* But the normal motorcycle flywheel is not. It is recessed and strengthened at one point to take the crankpin, and opposite to the pin it is thickened to act as a bob-weight to help balance the mass of the connecting rod, piston, gudgeon pin, and rings. Such a disc will wobble when spun if the mean of its mass is not correctly disposed. Fig. 1 will make this clear. It is plain that the shaft should actually be at 90° to the line A-A, and not to the plane of the flywheel B-B. Often, it is not so. Only by disposing the masses equally about the geometric centre can smooth running of the disc and shaft be achieved.

It goes almost without saying, of course, that the shaft and the wheel must be properly attached to each other. In my opinion, the practice of pushing in a shaft with a flange on one end and then riveting the flange and the wheel together is barbarous in the extreme. The rivets are put into shear as power and idle strokes alternate and, as any first-year engineering student knows, rivets



Five classes of crankshaft assembly, as tabulated in column one, above. They are (left to right) Nos. 1, 2, 3, 4 and 5.



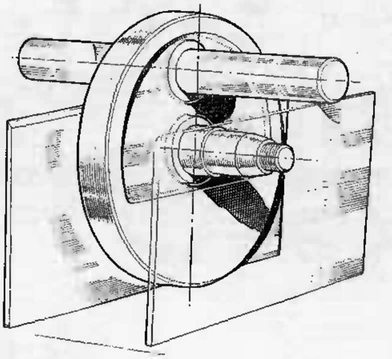


***Fig. 1** (left): Flywheel and shaft, showing plane of centres of mass (AA) and geometric centre line (BB). **Fig. 2**: Checking "vertical hang" of a flywheel.*

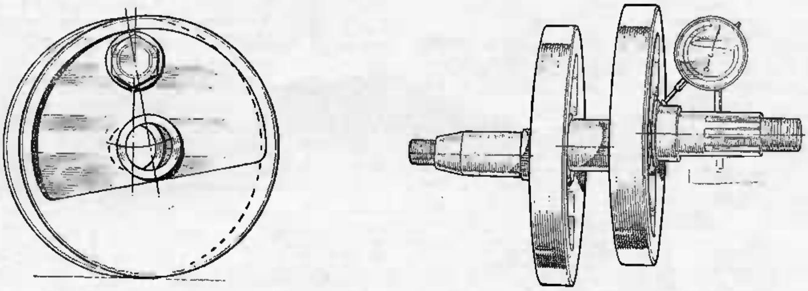
are not able to take this type of load. (Some mainshafts are pressed in only!) A more satisfactory method is to use a spindle, retained by a nut, and pulled into place against either a lapped taper or a flange (Fig. 5); no keys or keyways are necessary or desirable, given good workmanship.

Another fault that is sometimes found is incorrect "vertical hang," as I call it. The individual flywheel has a mandrel inserted into the big-end eye and this mandrel passes through for some distance, projecting on either side. The mandrel is then set on knife-edges, as in Fig. 2. Naturally, due to the built-in bob-weights, the part of the flywheel opposite the mandrel will sink to the bottom. Now, if the counterbalancing weight has been correctly disposed, the centre of the mainshaft will be vertically below the centre of the big-end eye. It is by no means unknown to find error here.

However, if trouble is found it must be corrected before the wheels are assembled. For instance, if one wheel has an error one way and the remaining wheel an error of the same magnitude in the other sense, placing the complete flywheel assembly on knife-edges will show no error. But in action each wheel will attempt to describe a circular path about its centre of gravity; the net result will be the two wheels desiring to follow separate circular paths superimposed on each other and on the normal mechanical rotation of the parts. The stresses and strains set up, and the vibration too, are very real and most unpleasant. No amount of change in balance factors can ever correct this trouble.

When two flywheels are "out" in this respect, the effects on the crankpin joining them are best left to the imagination. Anyone who has had a crankpin break at speed will know that the results can be unpleasant—and costly.

So the two flywheels can be rectified (if necessary) and then united by a common crankpin. From the foregoing, it is apparent that once the shafts have been set in the right relation to the wheels, the crankpin must join the wheels in such a manner that both shafts have a common centre line. Put simply, the two mainshafts are part of an interrupted mandrel, and should be as true! For instance, both crankpin holes must be at exactly the same radius from the main shaft centre. Error here is occasionally known; it is watched for especially when dealing with odd wheels (i.e., from two engines) which have been made into a pair.



***Figs. 3 and 4**: Two views of an exaggerated example of out-of-truth assembly; on the right, shaft displacement is being checked with a dial test indicator.*

Both shafts *must* be in line when the big-end nuts are finally tightened. An exaggerated example of out-of-truth assembly is shown from two viewpoints in Figs. 3 and 4. My aim is to reduce this type of error to an out-of-truth total run-out of .001 in. The correct figure is, of course, zero, but commercially this is seldom obtainable and is not even necessary for normal usage. The more bearings that are used, the greater the need for accuracy.

Keeping in mind the results that can stem from a physical mass not being disposed geometrically, let us examine what might happen in the case of a twin-cylinder crankshaft where the inner end of each crankpin terminates in a large flange that has a machined surface true with the journal and shaft. Between them is sandwiched a central cast-iron flywheel.

Rotational speeds higher than those intended by the designer may involve trouble where a cast-iron flywheel is used. But that is by the way. The central flywheel, obviously, is accurately machined to take both halves of that crankshaft and unite them so that they are true with each other. Dowels generally ensure that virtually no malalignment is possible.

Now if the cast surfaces of the flywheel are not machined properly all over, the mean of the mass centres and the geometrical centre will not coincide. Thus, high rotational speeds will produce a tendency for the flywheel to run out of truth, so causing vibration. On the other hand, careful casting and subsequent machining can make these two centres coincident and the flywheel, when speedily rotated, *will tend to stabilize the crankshaft instead of vibrating it.* Thus, not only can the cause of vibration be removed, but the once-offending part, now corrected, will promote smooth running.

In the case of parallel twins with three-bearing crankshafts, it is possible to arrange

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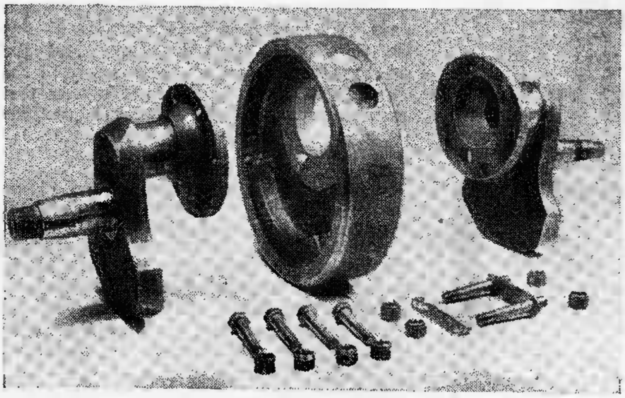


(Left) The bits that go round: rotating parts of a Norton 88 crankshaft assembly.

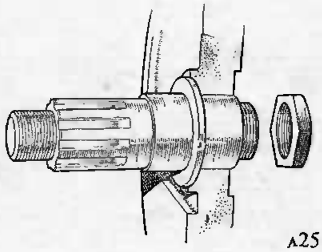


***Fig. 5.** (right): A sound method of retaining a mainshaft in its flywheel; the shaft is pulled into place against a flange.*

ENGINE BALANCE *Contd. from page 691.*

matters so that the centre main bearing becomes a convenient point for distributing the lubricant without oil bias to the big-end journals. It must not be forgotten that a correctly designed and manufactured crankshaft can be made to stabilize itself at high rotational speeds. Everyone knows how difficult it is to upset a rapidly spinning toy gyroscope. In the same way, a heavy rotating mass like a twin-cylinder crankshaft is very difficult to disturb, and if the original machining has been such that the centres of gravity of the various parts all lie along the mechanical centres, then that shaft will be next to impossible to move from its rotational path by other forces.

When the above rules have been applied to individual cases, it may be assumed that the components will run smoothly provided they are not upset by disturbing outside forces. Such forces will come from the reciprocating parts—namely the connecting-rod, piston, gudgeon pin, circlips, and rings. Resonance with the frame may also set up annoying vibration. It is not possible to remove *all* vibration, but only to minimize it to the point where it is of no consequence. Next week I shall explain how I achieve this and how I balance an engine to compensate for the reciprocating parts.

(*To be continued*)

Crankshaft Construction and Engine Balance

Concluding the Article in which Well-known Engine Tuner LAURENCE HARTLEY, Interviewed by Bruce Main-Smith, Describes Some of his Methods. Here, he Deals with the Reciprocating Parts

Mr. Hartley checks the run-out of a "Manx" Norton flywheel assembly. In this type of engine, readings are taken on the flywheel rims as well as on the mainshafts.

LAST week I explained how the purely rotating mass of a crankshaft assembly can be treated so that it revolves smoothly at all speeds providing it is not influenced by outside factors. Unfortunately, some outside factors always exist. The principal ones are the connecting rod(s) and piston(s) assembly, and resonant vibration from the frame. Let us examine their influences in that order.

What is a reciprocating part? The piston, gudgeon pin, circlips, and rings are purely reciprocating. They travel only in a straight line along the cylinder. The connecting rod is partly rotating and partly reciprocating. For instance, the small-end eye, which is generally located in the centre of the cylinder by the piston, has, to all intents and purposes, a purely linear motion up and down the bore. Therefore it may be said to be fully reciprocating. The big-end eye, on the other hand, describes a circular path; it is, therefore fully rotating. Obviously, at intermediate points along the rod the proportion of rotating to reciprocating mass varies.

At this stage, I depart from the conventional procedure of weighing both big and small ends of the rod. For convenience, I take only the small-end weight. This is assessed, as in Fig. 6, by placing the small end of a bare rod on a balance pan (or suspending it from a spring balance), whilst the big-end is supported on a convenient fulcrum (say, the crankpin), a line taken through the centres of the two eyes being substantially horizontal.

My next step is to determine the weight of the piston complete with rings, circlips and gudgeon pin—correct to the nearest 1/10 oz. for touring or sports purposes, or 1/20 oz. for racing work.

I then select the balance factor that I intend to use for the engine and frame in question, and for the use to which they are to be put, and arrive at a weight to be suspended from the small-end by the following formula:—

Balance factor × piston weight) — ([1 — balance factor] × small-end weight)

This method enables balancing for reciprocating parts to be carried out after the assembly and truing of the flywheel(s), which is most convenient.

For example, let us suppose that the balance factor is .6 and the weight of the piston components comes to 10 oz. Then the first part of the equation is (.6×10). If the small-end weight comes to 3 oz., the second part is ([1 — .6] × 3). Inside the brackets the values are 6 and 1.2 respectively, which, when subtracted, give the weight of 4.8 oz.—this to be hung from the small-end of the vertical rod for final balancing as in Fig. 7. The details of estimating the balance factor, about which so much hoo-ha is written, I will go into later; the balance factor plays an important part, but unless the crankshaft itself is dynamically smooth-running, correcting for reciprocating weight will not result in the desired absence of vibration.

The rectified and trued flywheel assembly, plus the connecting rod, of course, is then placed on its mainshafts on a pair of horizontal knife edges and allowed to come to rest. It will do so with the crankpin uppermost because, of course, the piston is not fitted. As the individual wheels have already been rectified during the vertical-hang process, there will be no heavy side to deflect the crankpin from the 12 o'clock position.

A weight is next attached to the small-end, equal to the amount to be balanced, as calculated above. The weight of the suspending medium is also taken into account, though a small piece of fine cotton weighs very little. (Another method is to slip a bolt and nut through the small-end eye and add washers to the bolt to make the desired weight.) If everything has been done correctly and the balance factor rightly assessed, the crankshaft will rest freely without any tendency to move in either direction. If it does settle, it will be with the pin at the top or the bottom, that is at 6 or 12 o'clock.

If the pin sinks to the bottom, obviously the bob weights are too light and the wheels will have to be drilled near the pin, care being taken to drill each wheel equally right through, and to get the holes disposed symmetrically on either side of the centre line. It goes without saying that if the pin rises to the top the bob weights are too heavy, and the weights themselves must be similarly drilled until the pin can come to rest in any position without bias.

Now it is apparent that, when the wheels have been drilled so that they will rest in any position, they have been balanced taking

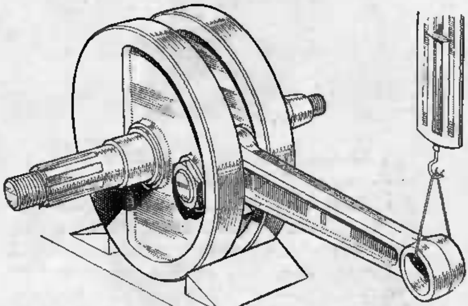
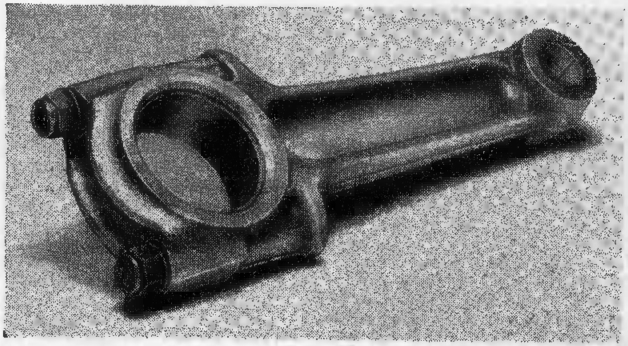
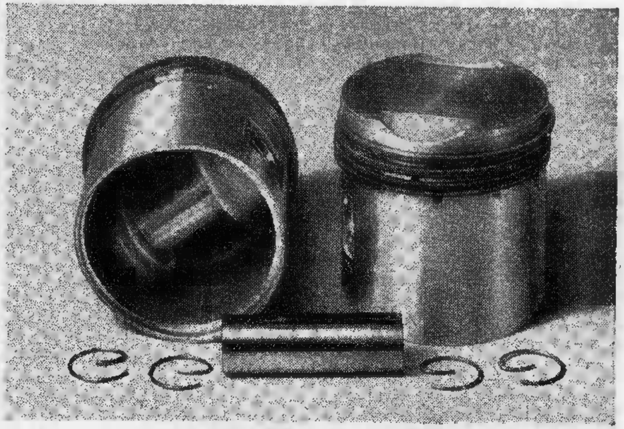
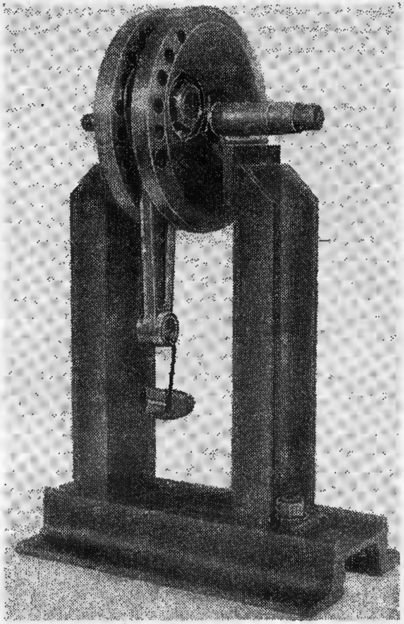


Fig. 6. *Weighing the small end with a spring balance (see text).*





On the right are the truly reciprocating parts of a Norton 88 engine; the con-rod (above) is "half and half." The "Manx" Norton flywheel assembly below ***(Fig. 7)*** *is balanced by weights suspended from the rod.*



into account the amount of weight hanging on the small-end in the process. It will be recalled that this weight was calculated by weighing certain components and then carrying out a simple mathematical calculation involving the use of a figure called the balance factor.

The magnitude of the balance factor is clearly going to affect the size, number and disposition of the holes drilled in the crankshaft assembly—hence the balance and smooth running of the power unit. Thus one's estimation of the correct balance factor for a given engine in a given frame is a vital one. After 35 years' experience, I find that I am reasonably proficient at this task, and am usually able to achieve first-time results. But not always.

For a given engine, irrespective of mounting, there is a definite, calculable, best balance factor. This, however, has often to be modified to suit the characteristics of a particular frame (on this account, it may vary between .55 and .8).

For instance, imagine a length of tubing supported at both ends, like a top frame tube. That tube can be vibrated at a definite frequency, and also at multiples of that primary frequency. If an engine has a vibration period which happens to coincide with that of the tube, the two periods will get into step. We all know how a marching body of troops, *all in step*, can bring down a bridge; so we can easily imagine the result of this coincidence. It is a violent vibration period—which the rider usually, and unfairly, blames entirely on the motor.

What the balancing specialist has to do is to estimate the tendencies of the frame and try to modify the balance factor to avoid this "in-stepness." Often I can arrange matters so that vibration occurs outside the usable r.p.m. band and then there is no trouble. The speed range of the engine, therefore, has a great deal to do with the selection of a suitable balance factor.

Nevertheless, it can be seen that a critical consideration is the balance factor and that its assessment calls for long experience. However, I would stress that its importance is more than overshadowed by the necessity for achieving smooth running of the crankshaft itself, as described in Part 1 of this article.

I could go on to elaborate on the folly of using too much ignition advance (and why too much causes a motor to "tire"); the advantages, for touring use, of a small-bore carburetter in conjunction with a highish compression ratio; the wisdom of making use of our British-made alcohol in a blend with petrol to make a greatly improved fuel for the average road user. I could also speak on two more of my favourite themes—the dangers of using tetra-ethyl-lead in fuel and the folly of restricting road-racing machines to ordinary pump fuels—but these points are outside my terms of reference in the present discussion. But I do hope I have succeeded in showing that crankshaft and engine balancing is neither a "black art" nor a hit-or-miss affair, and that its application, even to the modern motorcycle, can frequently pay dividends in terms of smoother running.