MOTORCYCLE ENGINEERING—26

The Problem of Balance

Part 1—Basic theory and the single

By PHIL IRVING

THE piston-connecting-rod-crankshaft mechanism which forms the basis of every reciprocating engine suffers from the inherent defect that it gives rise to unbalanced forces which may reach very high figures and cause either vibration which is externally noticeable, or internal deflection which may reduce the mechanical efficiency, or bearing-wear which limits the life of these components according to their ability to withstand the loads and speeds imposed.

This article is mainly concerned with the problem of vibration—or rather of reducing it to acceptable proportions, since it is an inconvenient but incontestable fact that one cannot balance completely a single-cylinder engine (or, for that matter, any form of twin or four-cylinder engine commonly employed in a motorcycle, unless it is built with two sets of opposed cylinders).

Of the lot, the single and the parallel-twin four-stroke, which mechanically is the same as a single since both pistons move up and down in unison, are the least well balanced. Yet, by skill in internal design and mounting in the frame, even these can be made to run very smoothly, except, possibly, at or above some critical speed at which vibration becomes annoyingly evident. That this is no mean feat will be appreciated from a consideration of the magnitude and complexity of the forces involved and how they are generated.

Once started on its way up the cylinder, the piston with its related components endeavours to keep on going, its motion being arrested and immediately reversed only by the action of the connecting-rod and the momentum of the crankshaft. In fact, if the latter component could be made absolutely weightless the piston would go up to top dead centre and *stay there*, however fast it might have been travelling on the up-stroke.

The force required to stop and restart the piston is at a maximum just at t.d.c. and if the con-rod were infinitely long, a force of the same value, but opposite in direction, would be required to perform the same function at b.d.c. The piston would then possess what is known as simple harmonic motion, or S.H.M., but this condition never exists because, for reasons some of which are obvious and some are not, the con-rod has to be a lot shorter than infinity and is usually somewhere around four times the crank radius or twice the length of the stroke.

Because of the angularity of the rod to the centre-line at or near mid-stroke, the points at which the piston attains its maximum upward or downward velocity, assuming, as is usual, that the cylinder is central and not offset, occur not at the 90° crank positions but at about 76° before or after t.d.c., the precise positions depending, of course, on the con-rod/crank ratio. This means that the piston has 152° of crank rotation to get from maximum speed down to zero and back to maximum during the upper half of the stroke, and 208° to go through the same sequence during the lower half; the upward inertia force must, therefore, be greater than the downward force.

With simple harmonic motion (which, incidentally, can be obtained in a swash-plate engine of the Michelle crankless type as well as by the impracticable scheme of an infinitely long rod) the inertia forces can be represented graphically on a crank-angle or time base, as a symmetrical cosine wave of which the maximum values are given by the formula

$$.0000142\ WN^2S$$

where W=weight in lb., N=r.p.m. and S=stroke in inches.

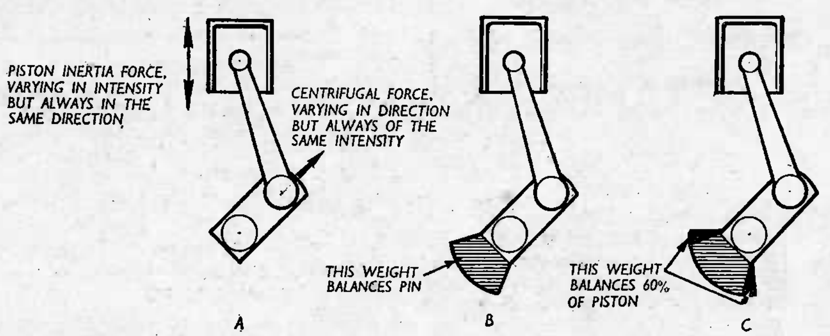
The effect of the necessarily short rod is to modulate this symmetrical curve by superimposing on it an infinite number of other harmonics, forming a Fourier series of increasing frequency and decreasing amplitude. Of these, only the secondary harmonic of twice the frequency and one-quarter the magnitude of the primary need concern us here, as the higher harmonics assume any importance only in engines with six or more cylinders.

This secondary harmonic gives rise to forces which act upwards at *both* t.d.c. and b.d.c., hence accounting for the increase in the actual force at the former point already noted; but it is also present at both 90° crank positions, at which it acts downwards, irrespective of the direction of piston travel. It has therefore within itself the possibility of initiating a secondary vibration at twice engine speed. This may become a reality in a four-cylinder engine where the secondaries from all four pistons act in unison although the primaries may cancel each other out; in a single, it is so small in relation to the primary out-of-balance forces that its effect may almost be disregarded (and anyway there is nothing one can do about it).

Arithmetically, the effect of a rod with a comparatively short length is to increase the primary force at t.d.c. and reduce it at b.d.c. by the factor $\frac{R}{L}$, where R is the crank *radius* and L is the rod length (both, of course, measured in the same units). As the rod is usually about four times the length of the crank radius, this factor can for our purposes be taken as $\frac{1}{4}$, the slight differences encountered in practice having only a minor effect.

To take a concrete example, in an engine with 3 in. stroke, a piston weighing 1 lb. complete with rings and pin and running at 6,000 r.p.m., the primary force, using the formula already quoted, works out at $.0000142 \times 1 \times 6{,}000 \times 6{,}000 \times 3 = 1{,}534$ lb. The effect of the rod is to modify this figure by $\frac{1}{4}$, or 383 lb., so the actual upward force at t.d.c. becomes $1{,}534+383=1{,}917$ lb., and the downward force at b.d.c. becomes $1{,}534-383=1{,}151$ lb.

Since these forces vary in direct proportion to the weight of the piston and to the length of the stroke, and also vary in proportion to the *square* of the speed, the figures given may be taken as basic ones for estimating the forces generated in any other size of engine very quickly. Thus, by increasing the piston stroke from 3 in. or 76.2 mm. to 80 mm., with all other things remaining equal, the maximum load goes



Simplified diagrams of a basic single-cylinder engine : (A) *completely unbalanced,* (B) *with weight added to balance the crankpin and big-end and* (C) *with further weight to balance the piston partially.*

up in the ratio of 80 to 76.2 and thus becomes 1,600 lb. in round figures which are easy to memorize; while at 7,000 r.p.m. it goes up in the ratio of 49 to 36, or nearly one-third more, and thus becomes 2,250 lb.

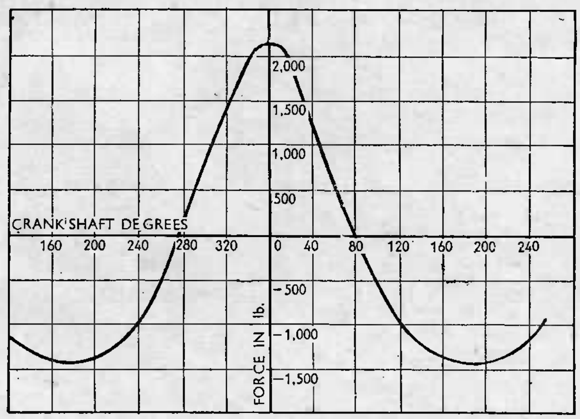
The last figure is rather staggering when one realizes that it is about five and a half times the weight of the complete machine and would, if it were to be applied continuously, give it an upward acceleration of $5\frac{1}{2}$ *g* and a vertical velocity of over 1,200 m.p.h. in a mere 10 seconds, by which time it would have attained an altitude of nearly a mile and three-quarters!

Fortunately, it is not applied continuously but only momentarily at the end of each stroke, so that an instant after the machine begins to rise under the action of the upward force, it is pulled downwards again by the force existing at the bottom end of the stroke. The net result is a vertical vibration of small amplitude—unless the frequency, which is equal to the rotational speed, happens to coincide with that of the whole machine or some part of it which is free to vibrate. The ends of the handlebars and the tail portion of an overhung rear guard are two common instances of parts which will vibrate in resonance and may develop quite a large amplitude, even though located at some distance from the exciting forces.

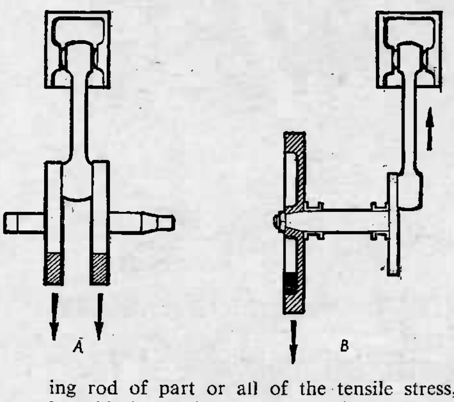
These forces are transmitted through the con-rod and crankshaft to the crankcase, and it is sometimes thought that the action of combustion pressures on the piston will counteract the inertia force and thus help to balance the engine, but this view is erroneous. It is true that the gas-pressure may be equal to, or even exceed, the inertia force and will therefore relieve the connecting rod of part or all of the tensile stress, but this is purely an internal effect and the out-of-balance forces are still present. Gas-pressure reactions *can* however make their presence felt in other ways, as noted in Part 16 of this series on "Torque Reactions" (December 17, 1959, issue).

Having seen that the piston gives rise to forces which alternate in direction but are unaffected by gas pressure, the question arises as to how their ill-effects can be minimized. The only component which offers any assistance is the crankshaft, which itself is unbalanced by reason of the weight of the crankpin and the big-end of the connecting-rod. This weight can be balanced quite easily by adding counterweights opposite to the pin, or, if disc flywheels are used, by removing metal from them on the same side of the pin until the centre of gravity of the assembly is brought exactly on the mainshaft axis.

(Right) Out-of-balance forces in a single-cylinder engine at varying degrees of crankshaft angle from t.d.c.



(Left) Counterweighting must be equally disposed in relation to the cylinder, as at A. *An extreme case of offset counterweighting is shown at* B.



If this result is achieved, the way in which weight is added or subtracted is immaterial so far as balance is concerned, provided that it is carried out equally on each side of the cylinder centre-line. If it is not, an unbalanced couple is set up which tends to make the crank assembly rotate conically instead of on its own axis, though the effect would not be very serious unless all the counter-weighting were done on one flywheel, especially if this happened to be an outside one and therefore some distance from the crank-pin.

Balancing the piston is not so easy—in fact it is out of the question, because if an additional counterweight were such that the centrifugal force it generated during rotation was equal to that generated by the piston at t.d.c., it would achieve balance at that point, but at that point only. It would be too great to balance the lighter inertia force present at b.d.c. and, worse still, at the 90° positions the centrifugal force would be acting horizontally, with no other force present to counteract it at all.

The final result would be merely to convert an engine which was unbalanced in the vertical plane into one which was just as badly unbalanced in the horizontal plane (assuming that the cylinder itself was vertical). While the effect might be, in certain circumstances, to confer greater smoothness of running, it would be by no means a solution to the problem and would do nothing towards lightening the loads imposed on the main bearings.

The best that can be done is to effect one of those compromises with which all forms of engineering are liberally sprinkled, and counterweight the crankshaft by an amount equal to some percentage of the reciprocating weight which inspired guesswork and previous experience indicate to be adequate, then to alter it in the light of experimental results until the desired degree of smoothness is obtained.

This may seem to be a light-hearted way of describing a lamentably haphazard procedure. Nevertheless, there is no magic formula to enable one to do anything else and, despite that lack, excellent results can be obtained in quite a short time by knowledgeable people, especially when working on familiar ground so far as the frame and the method of installing the power-unit are concerned.

Frame and installation affect the issue partly because of the aforesaid resonant effects, partly because the method of mounting the engine has a bearing on its ability to vibrate in relation to the frame, and partly because the selected balance-weight percentage, commonly termed the "balance factor," alters not only the amount of unbalanced force unavoidably remaining, but also the plane in which it reaches its maximum value and which lies somewhere between the cylinder axis and the 90° crank positions.

Usually, a balance factor of 60-65% will be found to be correct, but it may vary widely; the factor for the MOV Velocette was as high as 85%, while some engines have been down to 40%. The reciprocating weight to which this factor applies consists of the complete piston plus the top half of the connecting-rod, as measured with the rod horizontal and with the big-end supported in some frictionless manner.

Because the eventual smoothness obtained depends so much upon the frame, a balance factor which suits one installation may not suit another, using the same engine —a fact which used to plague proprietary engine manufacturers, some of whose customers insisted upon some specified percentage which others would not accept.

An instance of this was found in the original Vincent-H.R.D. singles. These engines ran very smoothly in the standard spring frame with a factor of 66%, but the speedway versions, of which a very few were made, vibrated badly in a cobwebby dirt-track frame until the factor was reduced to 61%, this not-very-large reduction making all the difference between a machine which was passably smooth and one which shook itself out of your hands.

The 66% factor was still employed on the post-war "Comet," with an inclined engine forming part of the frame structure. This model was notably smoother than the diamond-framed edition and, in fact, was free from any definite vibration periods of the kind which are occasionally found and are so difficult to eliminate. Whether this virtue was due to the type of mounting or the angle of inclination is, however, difficult to assess and impossible to prove, because in this case the construction was such that you could not have the one feature without the other. It was, however, possible to employ without ill-effects pistons varying by as much as two ounces in order to obtain some desired compression ratio, whereas

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some models of different construction are very sensitive to piston weights.

When a periodic vibration is very persistent and altering the engine balance only chases it up and down the speed range, the best solution is to arrange matters so that it occurs at a speed as far removed from normal usage as possible. On a touring model, vibration at the highest engine speeds will rarely annoy most riders, and when it does set in at least it constitutes a warning against over-driving, while on a racing model any resonant period below 4,000 or 5,000 r.p.m. will never be noticed, except momentarily when getting away from the start.

Sometimes it is simpler to eradicate undesirable resonant vibrations by attending to the offending component rather than by altering the balance of the engine. This does not necessarily mean simply stiffening it up, though that is the procedure which is most likely to produce results. Softening the component or its attachments may be just as effective, as witness the occasional use of rubber-mounted handlebars.

Resonance can also occur in the crank shaft assembly, sometimes with destructive results, and this condition must be specially guarded against in engines with more than one cylinder. These will be the subject of the next instalment, though it must be appreciated that the balancing problem of the fundamental single-cylinder has been dealt with only in a very sketchy manner. Anyone who is interested in the complicated mathematics involved will find an excellent chapter on it in N. W. Judge's " Automobile and Aircraft Engines."

NEXT WEEK

Balancing multi-cylinder engines