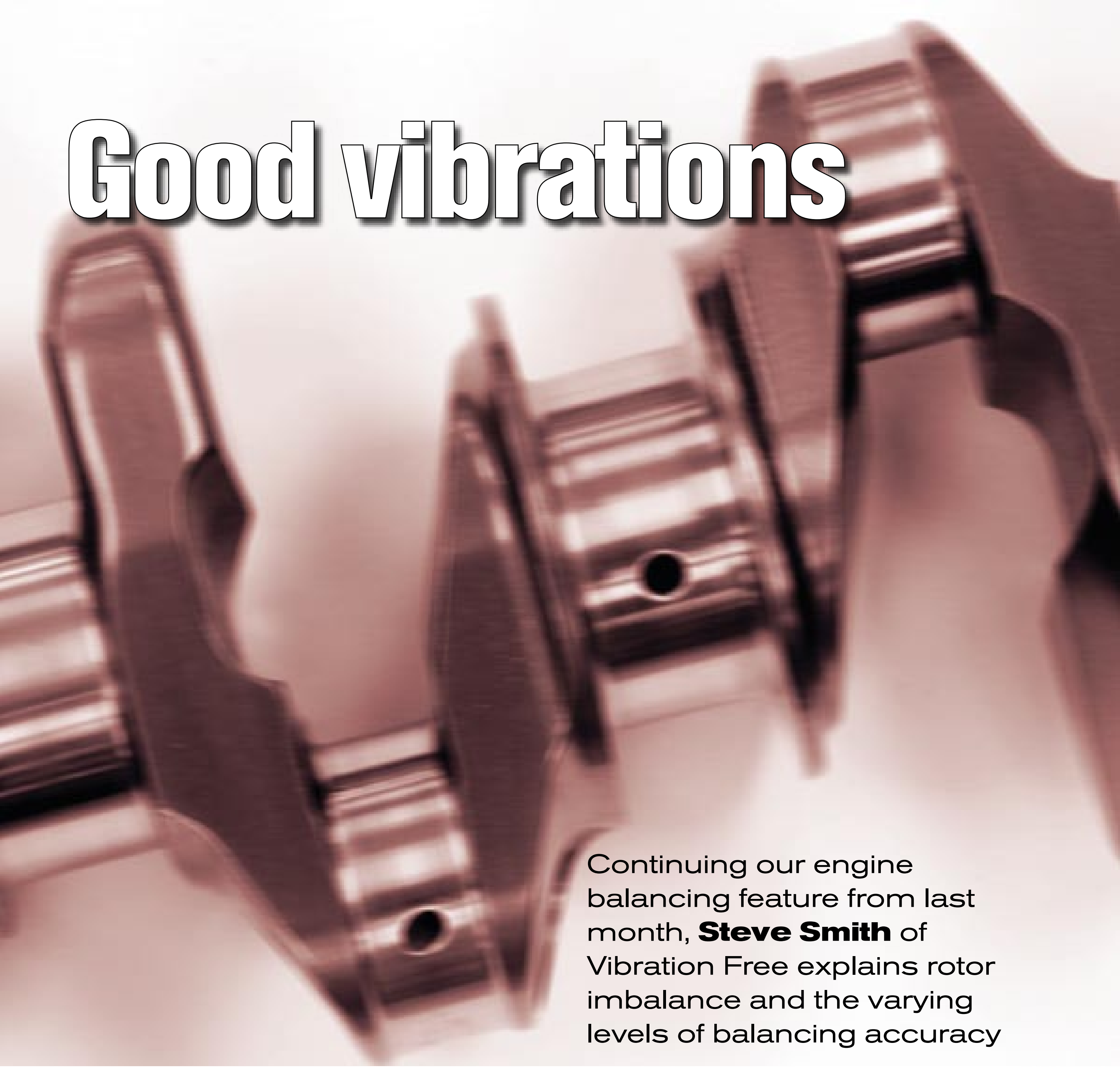
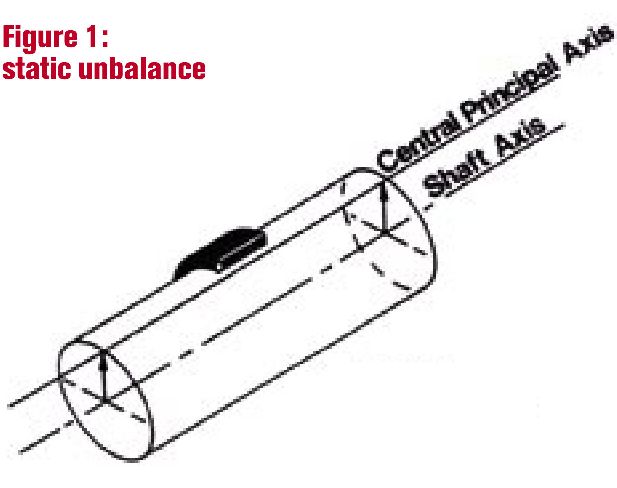
Good vibrations



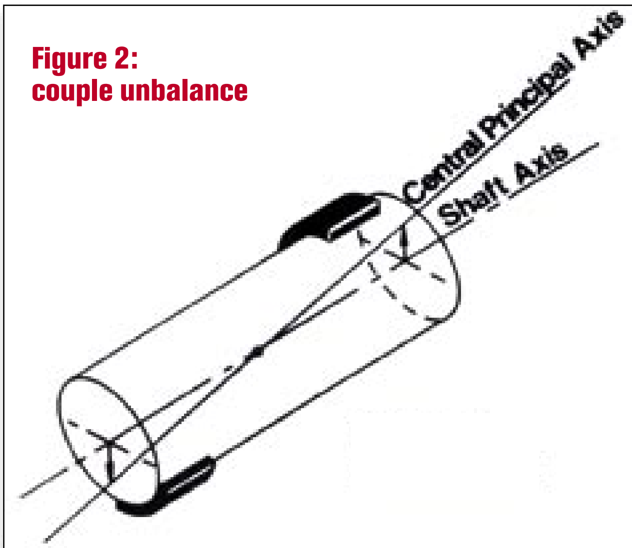
Continuing our engine balancing feature from last month, **Steve Smith** of Vibration Free explains rotor imbalance and the varying levels of balancing accuracy

Figure 1: static unbalance

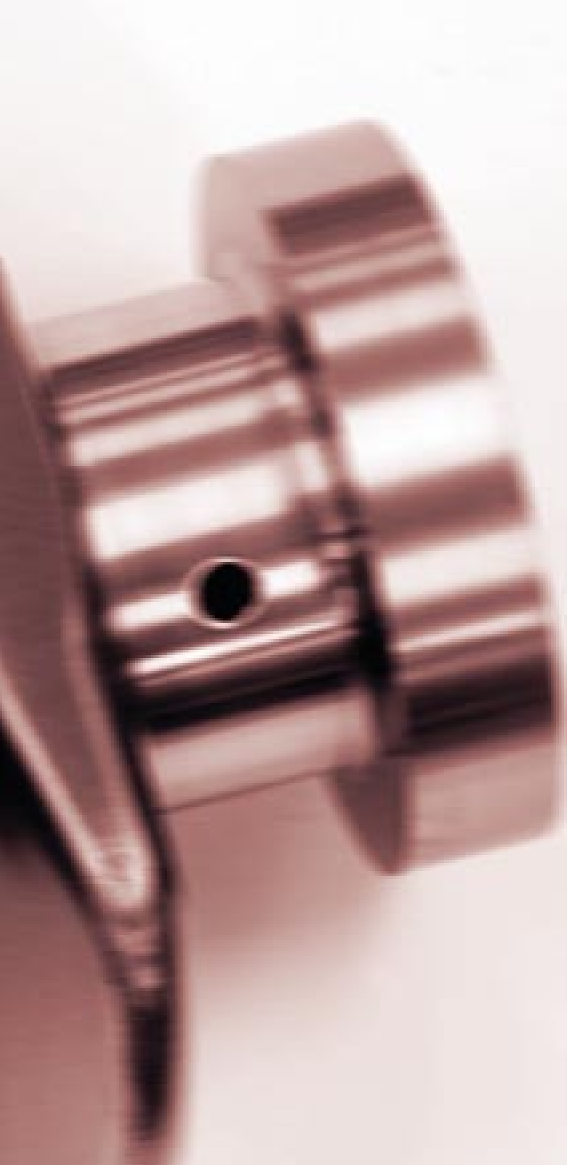


An offset of the principal axis parallel to the rotating centreline

Figure 2: couple unbalance



An offset equal in amplitude and opposite in direction at each end of the rotor



All rotors, especially crankshafts, have some degree of flexibility, depending on their service speed

Part one of this article, published in V17N3, reviewed the importance of engine balancing and its beneficial effects on reliability and efficiency. It outlined the many potential causes of unbalance, introduced the concept of rigid and flexible rotors and then described units of unbalance and how unbalance gives rise to vibrational forces.

Now we move on to consider the four different types of rotor imbalance, issues relating to the effective balancing of both rigid and flexible rotors, stiffness issues with alloy blocks and ISO grades of crankshaft balance.

Types of rotor imbalance

The four characteristic unbalance cases are termed static, couple, quasi-static and dynamic. Static unbalance (Figure 1) produces an offset of the principal axis parallel to the rotating centreline. It is recognised through showing equal amplitude and phase in two planes. Couple unbalance (Figure 2) produces an offset of the principal axis equal in amplitude but opposite in direction at each end of the rotor. The principal axis intersects the rotating centreline at the centre of gravity. It is recognised about two end planes of a rotor by having values of equal amplitude but 180 degrees out of phase. Quasi-static unbalance (Figure 3) is similar to couple except the principal axis here intersects the rotating centreline at a point other than the centre of gravity. It comes about when a static unbalance has the same angular position as one of the components of couple unbalance. It is recognised at two end planes by having 180 degrees phase difference and unequal amplitudes. Dynamic unbalance (Figure 4) produces a condition where the principal axis and rotating centreline do not coincide at all. It is recognised by there being no relationship between amplitude or phase about two planes of measurement.

Dynamic unbalance, which is a combination of static and couple unbalances, is the most common form of unbalance. The displacement between the principal axis and the rotating centreline is dependent on the position and value of the unbalances as seen at any two arbitrary balancing planes. These values will be different transversely along the length of any given rotor.

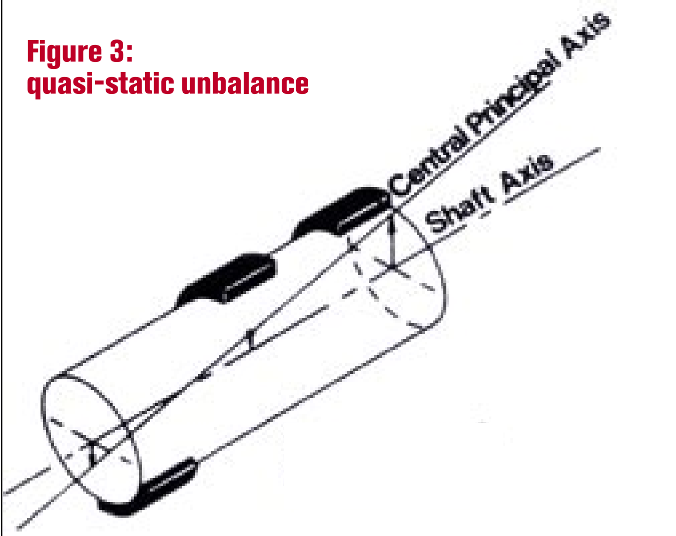
It follows that any rotor of significant length will require dynamic balance correction about two chosen planes to minimise forces at the bearing journals. Generally, it will not be possible to determine the positions of the unbalances at every point along its length, rather to correct for a resolved value as measured at two chosen planes.

This method of balance correction is based on the presumption that the transverse offset between actual unbalance position and the unbalance correction does not create sufficient forces at service speed to cause the rotor to bend. It is this factor which decides between a rotor being rigid or flexible. Ultimately, all rotors are flexible – it is purely a matter of service speed.

Effective balancing

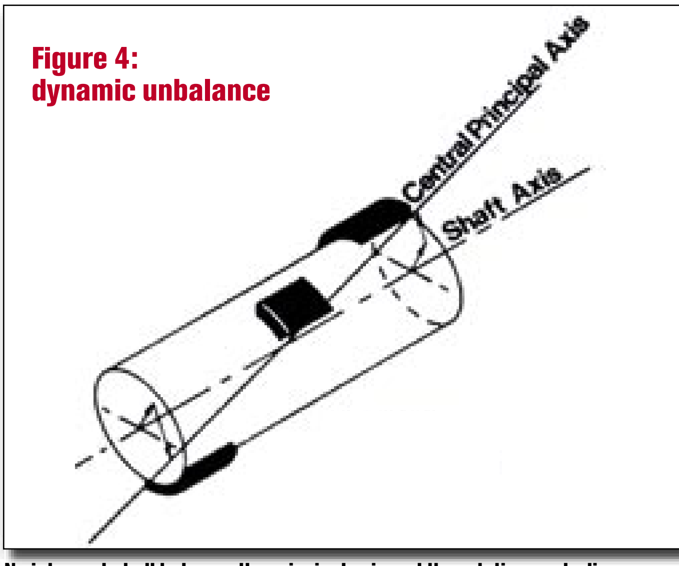
If a rotor is effectively rigid – that is, if it operates at up to 70 per cent of its first bending moment (critical speed) – it can be balanced about any two chosen balance correction planes. These will usually be as far apart and at the largest correction radius that can practically be achieved, so as to minimise the correction required. As a rigid rotor, it will not deform along its length due to internal bending forces throughout its →

Figure 3: quasi-static unbalance



A similar offset to couple unbalance but with the intersect not at the c of g

Figure 4: dynamic unbalance

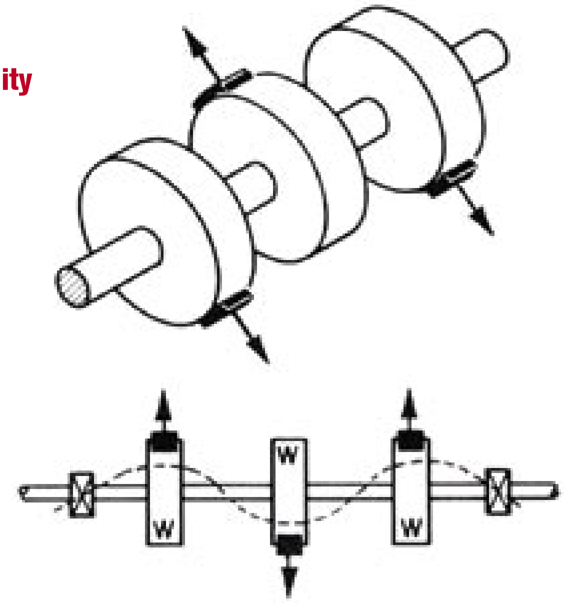


No intersect at all between the principal axis and the rotating centreline service speed range and will consequently be vibration free about its bearings. However, should the same rotor have a large initial unbalance somewhere about its length then this may not be true. Likewise, if the service speed is increased, then again the rotor may become flexible due to increases in internal bending forces.

Rotors that operate at a speed above 70 per cent of their first natural frequency are considered flexible (Figure 5). That is, they will deflect along their length due to the internal bending forces created from unbalance. Flexible rotors have to be balanced with care to minimise these internal bending forces by carrying out balance corrections at several transverse planes close to or at the initial unbalance positions.

If an understanding exists of how the rotor bends then low-speed balancing can achieve a satisfactory result, as long as the chosen balancing planes meet the above criteria. Otherwise, initial balancing is carried out at high speed to reflect on how the shaft deforms due to its unbalances. Further corrections then prove out a satisfactory result. This type of high-speed balancing is very specialised.

Figure 5: rotor flexibility



If the internal bending forces in a crank are understood, it can be balanced satisfactorily at low speed

Cost-effective design

If all rotors were treated as flexible and balanced adequately along their shaft length, they would remain straight at all speeds. But this is too costly an exercise to consider in most cases, so the majority of rotors are designed to operate at a speed that allows them to be balanced as a rigid rotor. Crankshafts fall into this category and are typically manufactured with extra unbalance weight present in their end counterweight planes. This is to assist the drilling machine process during balance correction.

Crankshafts are also designed cost effectively to utilise the minimum material required to get the job done. This typically equates to a crankshaft with fewer counterweights than ideal to minimise bending forces along its length. For this reason alone crankshafts cannot be run at faster operating speeds, loads or swapped from iron to alloy blocks without due consideration.

Similarly, a crankshaft's response to bending forces will be sensitive to rotating inertias about its end planes and may not tolerate a reduction in flywheel mass. The stiffness and damping present on the rotating assembly is also critically affected by bearing clearances and shell width.

In most instances, automotive engines are quite tolerant of modest increases in power and operating speed. But with performance upgrades reaching new levels and the increasing use of alloy cylinder blocks, crankshaft design is becoming more critical with respect to suitable counter weighting.

In order to minimise internal bending forces and reduce bearing loads, crankshafts must be designed with counterweights along their entire length. The sizing of which is critical, so as to match the unbalance created by the big end journal and its associated rotating parts opposite the counterweight. As such, the crankshaft is being treated as a flexible rotor. Even if it truly doesn't reach 70 per cent of its first natural frequency, the benefits are evident through reduced bearing loads, refinement and performance.

The residual unbalance tolerance for any given rotor is usually decided by the manufacturer of the shaft. Where no tolerance value is available, the deduction of a suitable tolerance can be calculated by following guidelines set out by the International Standard Organisation. For crankshafts the document is ISO standard 1940 (available for purchase from www.iso.org).

Types of crankshaft and their corresponding quality grades

G Grade	Type of crankshaft
6.3	Racecar crankshafts, and generally car vehicle shafts definitely requiring very fine balancing.
16	Sportscar crankshafts and generally car requiring good balancing. Six or more cylinder engine shafts requiring accurate balancing.
40	Small car and truck crankshafts with limited balancing need. Four stroke, six or more cylinder crankshafts, elastically suspended, piston speed exceeding 9m/sec.
100	Diesel engine crankshafts with six or more cylinders, piston speed exceeding 9m/sec. One, two or three-cylinder engine shafts.
250	Diesel-engine crankshafts with four cylinders. Rigidly suspended, piston speed exceeding 9m/sec.
630	Big four stroke engine, diesel marine engine, elastically suspended crankshafts.
1600	Two stroke crankshafts of larger engines, rigidly suspended.
4000	Diesel marine engine crankshafts, rigidly suspended, no matter how many cylinders, piston speed not exceeding 9m/sec.

Quality grades

The ISO table shown left categorises crankshafts for different requirements into various quality grades. The lower the quality grade number, the lower the residual unbalance tolerance for that crankshaft. Crankshaft quality does not extend further than grade 6.3 but the grading system for other rotors requiring very fine balance, such as turbines and gyroscopes, extends through grades 2.5 and 1.0 to 0.4. The quality grade number directly reflects the level of vibration that would be expected at the rotor's bearings when running at service speed. The lower the quality grade, the lower the residual unbalance and running vibration levels. RE

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