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# Racecar engineering



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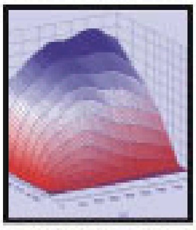

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## Engine balancing

Taking out the shake


# Bad vibrations

**Steve Smith** of Vibration Free explains the theory of unbalance and the factors that affect the degree of unbalance within an engine

Engine balance is of considerable importance in both road and racecars because it influences both the level and nature of engine vibration. While driver comfort is rarely a prime consideration in racecars, other potential effects of this vibration certainly are. In addition to compromising the reliability of the engine itself, it can degrade the performance and/or longevity of engine ancillaries and any other components that are mounted on or near to it. An engine with minimised vibration is also more efficient.

This article aims to provide a useful introduction to engine balance and crankshaft tolerance discussion. It also hopes to provoke consideration of other sources of vibration in an engine assembly and the limits imposed by crankcase and block stiffness.

## Interpreting unbalance

There are several ways of interpreting unbalance. The most common definition is that it is 'the uneven distribution of mass about a rotor's axis'. This induces a rotating force which is transmitted to the bearings and produces vibration in the rotor housing or system. The level of vibration produced from an unbalance is dependent on the system mass, stiffness and damping. The manufacturer of the assembly usually decides the acceptable unbalance limits, based on the bearing forces and resulting vibration. In setting these limits they may also be guided by authorities and standards organisations, or influenced by end user requirements.

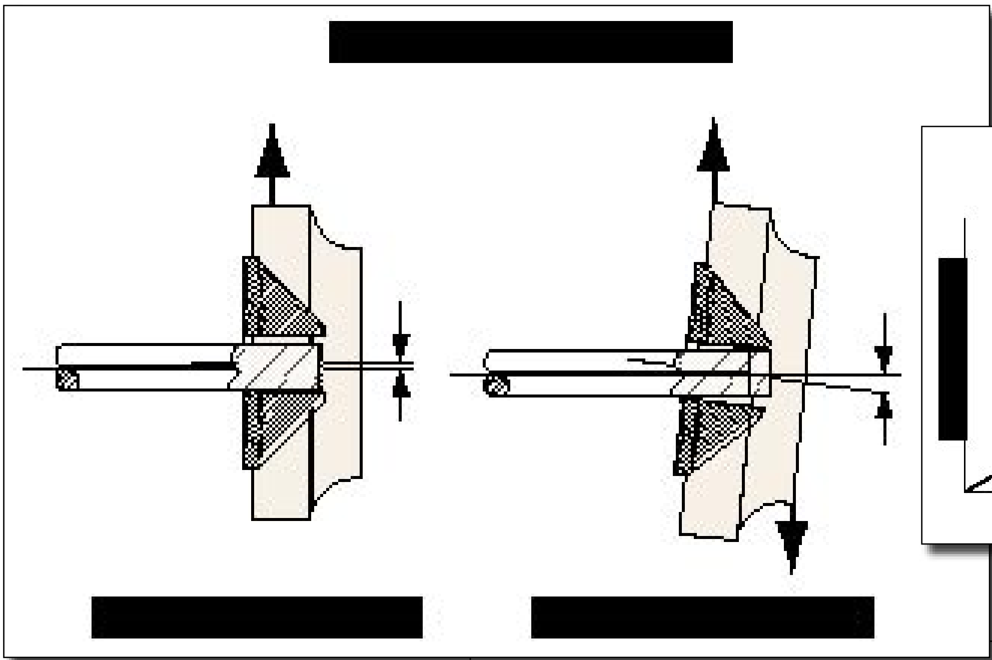
In the case of car engines there are also reciprocating elements (pistons and little ends of con rods) that have the potential to impart vibration, at the same frequency as the rotational unbalance of the engine. They can also have a direct influence on the rotational unbalance. For this reason reciprocating balance tolerances have to be taken into consideration, too.

> “UNBALANCE FORCES USE UP ENERGY”

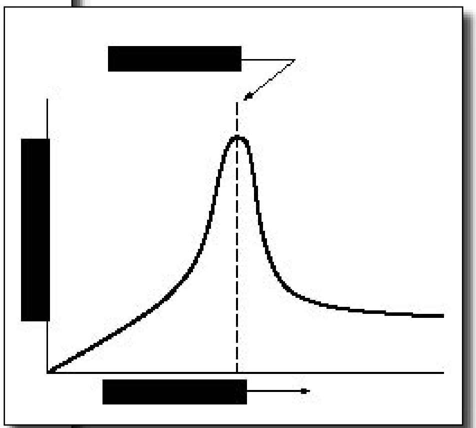
Crankshafts – the most common engine part to be balanced – can be quite complex in their shape and function. They are usually produced from forgings, castings or billets of steel that require several machining operations. The uneven distribution of mass from their rotating centre and along their length requires that they be dynamically balanced to minimise rotating unbalance forces.

The unbalance forces that remain in a rotating assembly use up energy that would be better spent producing rotational momentum. Also, the bearings that support rotating assemblies are designed to cope with a given cyclic load. If the residual unbalance uses up some of that capacity then the bearings have less ability to cope with process loads (ie the combustion forces in the case of an engine). Further, the casing structure that supports the bearing housings also has an operational stiffness designed to cope with process loads. If the assembly is close to its operational design limits then a small increase in residual unbalance forces can cause dramatic changes to the stability of the system assembly, with the possibility of structural failure.

The automotive industry has always treated crankshafts as being stiff enough to resist bending forces along their length when suitably supported within the crankcase bearings. This is a function of design, which is greatly affected by crankshaft process loads, rotational speeds and the degree of unbalance along the length of the crankshaft.

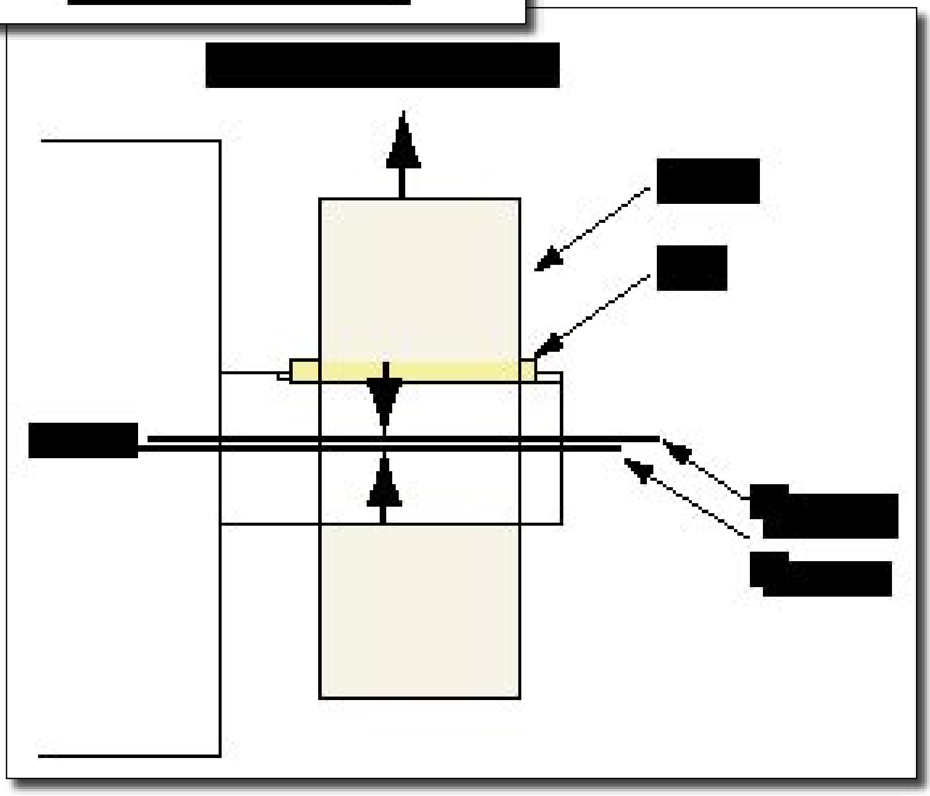


Imbalance occurs when the centre of mass of an assembly does not coincide with its rotational axis



Vibration amplitude increases dramatically at imbalance's critical speed

> **"THE UNBALANCE TOLERANCE IS A FACTOR OF OPERATIONAL SPEED"**



Acceptable tolerances of fit or balance on indvidual components can add up to an unacceptable imbalance on the complete assembly if all biases are orientated in the same direction

As crankshaft loads and speeds rise, the use of alloy crankcases with limited stiffness can lead to deflections of crankshaft and crankcase. The crucial factor could be simply the degree of unbalance along the length of the crankshaft.

Residual unbalance can therefore be a very sensitive issue in some applications. Tolerances must be established for component parts and whole assemblies, ensuring that the assembly clearances or processes do not lose the required accuracy. Take, for instance, the case of a four cylinder engine whose connecting rods are not match balanced accurately. If only one of the four connecting rods has a big end that is heavier by just 1g at a typical radius of 70mm on the crankshaft, this will produce a force of over 2.5kg at 6000rpm.

When balancing crankshafts in a dynamic balancing machine, it is usual to run the crankshaft about the main journals on open roller supports. Balancing speeds can be anywhere from 200 to 1200rpm, depending on the balancing machine being used. It is vital for the balancing operation that the crankshaft is both straight and stiff enough to resist whipping at the machine's rotational speed. The chosen balancing speed has nothing to do with the operational speed of the crankshaft, as the balancing machine is designed to measure the unbalance at the lower speed. It is the unbalance tolerance which is a factor of operational speed.

## Causes of unbalance

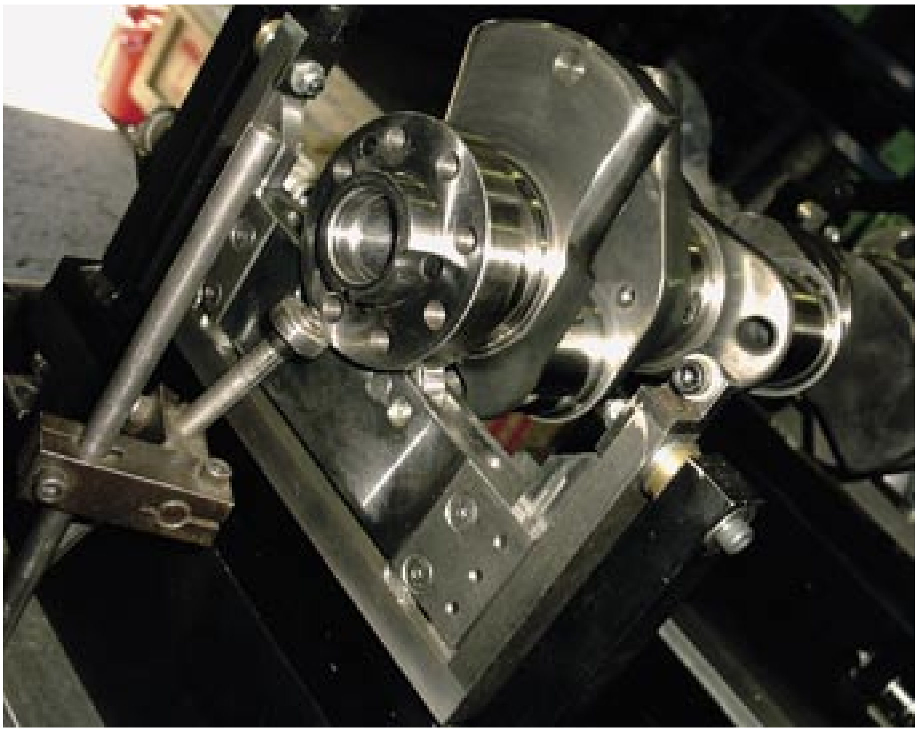
When considering the many components that make up an engine assembly, some of the most common causes of unbalance in rotating and reciprocating parts are as follows: castings and forgings of inconsistent density; cast and forged parts with un-machined surfaces; components with uneven symmetry about the rotational axis; assembly clearance tolerances producing eccentricities; fasteners (bolts, nuts, washers etc) of differing weights; heat treatments causing bending and deformation; uneven process coatings; thermal expansion during engine operation; unquantified influences such as oil entrapment and circulation in moving parts; tolerance stack up (eg a heavy side of a pulley in line with the heavy side of a damper) and tooling and calibration errors during balancing operations. Bearing all this in mind, it is always preferable to correct for unbalance of components as sub assemblies as this minimises internal forces within the completed assembly.

The cyclic stress imposed on the →

> **"RECIPROCATING ELEMENTS HAVE THE POTENTIAL TO IMPART VIBRATION"**

components of an assembly by the rotational unbalance force is a function of the unbalance level: the rotor mass, the rotational speed and the damping present in the system. The forces can be detrimental to an engine's useful life and performance in the following areas: bearing load and associated wear; fatigue in component material through cyclic stress; mechanical fretting at joint interfaces; increased wear in gear teeth; increased power consumption; reduced efficiency through wasted energy; reduced power capacity; transmission of vibration to other key components, affecting their efficiency; increase in maintenance; increase in transmitted noise; reduction in durability, quality and refinement.

Rotors requiring balance corrections are divided into two categories: rigid rotors and flexible rotors. Rigid rotors are those that operate at or below 70 per cent of their first natural (resonant) frequency, while flexible rotors are those that operate at or above 70 per cent of their first natural frequency (ie 70 per cent of their critical speed). As a result, flexible rotors need to be balanced differently to rigid rotors. Rigid rotors can be suitably balanced dynamically about any two chosen balancing planes, whereas flexible rotors need to be dynamically balanced about several planes along their length, but essentially it is the unbalance along the length of a rotor that causes it to bend or flex. When a rotor operates at exactly a sub harmonic of the critical speed, it will be more sensitive to unbalance effects than at non-harmonically related speeds.



**Dynamic crank balancing is usually done at between 200 and 1200rpm, yet in racecars operational speeds can be 10 or more times this. The vital unbalance tolerance is a factor of an engine's operational speed**

## Units of unbalance

The most common reference unit for unbalance is the product of the residual unbalance mass and its radius from the shaft axis, usually expressed in metric units eg g.mm. A flywheel having an unbalance mass of 5g at a radius of 150mm, for example, will have a unbalance of 750g.mm.

There is a direct relationship between the weight of the rotor and the eccentricity produced by the unbalance such that:

**unbalance (g.mm) = eccentricity (µm) × rotor weight (kg)**

So if a crankshaft weighs 50kg and has an unbalance about the centre of gravity of 400g.mm then this would produce an offset of the crankshaft mass about its principal axis of 8µm (0.008mm). This offset is also referred to as the mass centre displacement (MCD). Likewise, if a flywheel of 15kg was offset on the crankshaft mounting flange by as little as 0.04mm (40µm) it would produce an unbalance of 600g.mm, equivalent to a force of 15kg at 5000rpm.

## Understanding unbalance forces

The force ($F$) generated by rotational unbalance can be calculated from the formula

$$F = mr\omega^2$$

where $m$ is the unbalance mass, $r$ is radius from the rotational axis and $\omega$, the angular rotor speed in radians per second, is given by

$$\omega = 2\pi N/60$$

where $N$ is the rotor speed in rpm. This translates to approximately

$$F = 0.01(N/1000)^2\, mr$$

where $F$ is in kilograms, $m$ is in grams and $r$ is in centimetres.

For example, if a crankshaft has an unbalance of 1g at 1cm then at 1000rpm it will produce an unbalance force of 0.01kg. At 2000rpm this will increase to 0.04kg, so twice the rotational speed produces four times the unbalance force.

The inertia forces created by unmatched reciprocating weights can be calculated from

$$F = ma$$

where $m$ is the unbalance of reciprocating parts in kilograms and $a$ is the maximum acceleration of the reciprocating parts in m/s².

For example, if a multi-cylinder engine having a typical stroke of 100mm and a con rod ratio of 1.5 has one piston heavier by 0.5g then at 5000rpm it will produce an inertia force of approximately 1kg.

## Contact

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