"The Balancing Act"

Balance the design, or design for balance?

Steven Smith
Vibration Free

Vibration Free
Unit 18
Green Farm
Fritwell
Bicester
Oxon
OX27 7QU

Tel: +44 (0) 1869 345535
Fax: +44(0) 1869 346565
Email: steve@vibrationfree.co.uk
www.vibrationfree.com

1. SYNOPSIS

Development and manufacturing costs can be cut dramatically by adopting a Balancing Strategy at the design stage. This will greatly reduce the risk of discovering problems at the acceptance-testing phase.
In this paper the proven method of Dynamic Table Tooling is explained. A range of motor sport applications are presented where the traditional balancing difficulties have been tackled and successfully overcome.
In prototype areas the method provides speedy results, producing valuable feedback in the design process. Whereas in production, with fixed design the method provides an answer to a difficult balancing issue. In both cases it is a cost effective and time efficient result.

Race Tech 2004 invite the presentation of topics pertinent to motor sport, where aspects of controlling costs, increasing quality and utilising calculated simulation methods or practical means could be shown to have benefit to the industry.

This paper will present an approach to using sensitive dynamic balancing machines to assist in the development of crankshafts. It provides a test procedure which supports and exceeds present simulation methods, enabling better crankshaft design to reduce crankcase bearing loads. The procedures are quick and easy to set up, providing reduced development time for any engine configuration. It has a direct relevance to any reciprocating engine.

1.1 Introduction

Increasing needs for lower vibration and noise levels on rotating machinery; reductions in bearing and shaft loads and improved performance and efficiency, demand finer tolerances of residual dynamic imbalance.

"Unbalance is still the predominant cause of failure in rotating machines."

1.2 The predicament

At the design stage a mathematical approach is simple to dictate, but sometimes proves difficult to implement. In practice the residual imbalance can rise to an unacceptable level due to fitting tolerances, system response or stability of the rotating assembly.

Careful consideration must be given not only to the desired balance tolerance, but also perhaps more importantly, to the means of effecting correction of the rotor to guarantee a vibration limit is met.

Some rotating machinery components frustrate the efforts of traditional balancing methods.
This paper discusses by example, a proven tooling approach to handle awkward rotors or rotating assemblies. The tooling arrangement is a "Dynamic Table". This allows completely free movement of the rotor or assembly regardless of how it is clamped or supported.
The linear dynamic response of the table and suspension, allow for optimum dynamic balancing or dynamic assessment of the assembly. Using this approach to balancing can resolve many of the inherent difficulties.
The benefit of balancing an assembly is in this way is that it provides a qualified result, where component balancing can still fall victim to fitting tolerances.

1.3 The applications

The rotors or assemblies that dynamic table tooling can be applied to and are discussed within this paper are;-

- Wind tunnel model wheels
- Composite, single and two piece drive-shafts
- Centrifugal supercharger and turbocharger assemblies
- Mazda and Norton rotary type engine assemblies
- Single, multi or 'V' cylinder engine assemblies

1.4 The companies

Vibration Free located near Oxford in the UK, specialise in Vibration Analysis and Dynamic Balancing, providing services to;-

- Racing Teams
- Racing Engineers
- Automotive industry & component suppliers
- Development Consultants
- Manufacturing industry

The motor sport applications are centred on Engine and drive train requirements. Support is provided with vehicle vibration analysis and component or assembly dynamic balancing.

The vibration analysers and dynamic balancing machines are manufactured by IRD Balancing Ltd, located at Chester UK. Menard Engineering Ltd (MEL) are leading suppliers of world class race engines and services to the motor sport industry and are located near Oxford UK.

2. BACKGROUND

2.1 What is unbalance?

Unbalance is often defined as the unequal distribution of mass within a rotor about its rotating centreline. A condition of imbalance exists in a rotor, when vibration forces or motion is imparted to its bearings as a result of centrifugal forces.

2.2 Units of measurement

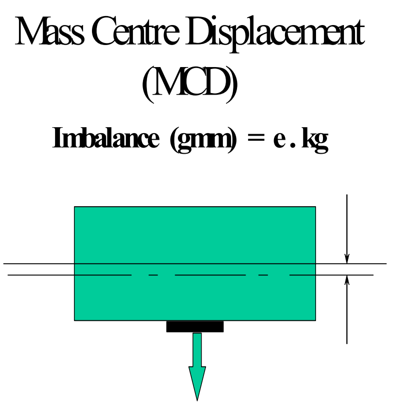
The amount of unbalance in a rotating body is normally expressed as the product of the residual unbalance mass and its distance from the rotating centreline. Therefore, the general units for expressing unbalance are gram-millimetres, (g.mm).

2.3 Balance tolerances to ISO 1940.

ISO 1940 (Balance quality requirements of rigid rotors), is one of the most widely used International guides for recommended balance quality grades. The standard categorises rotors, based on worldwide experience, according to their *type, mass* and *maximum service speed* into a quality grade (G). The number given (e.g. G0.4, G1, G2.5, G6.3, G16 etc.) relates to the allowable level of vibration (mm/sec) measured on the bearing housing at rotor service speed. It is the product of specific unbalance and the angular velocity of the rotor at maximum operating speed and is a constant for rotors of the same type. The quality grade determined by ‘G’ is related to permissible residual unbalance measured in g.mm and the allowable mass centre displacement (MCD) measured in micrometers, (µm).

In general, the larger the rotor mass, the greater the permissible residual unbalance. It is therefore appropriate to relate the value of the permissible residual unbalance, ***U***, to the rotor mass, ***m*,** in terms of permissible rotor mass centre displacement***,*** ***e*** and equate this to maximum rotor service speed and the ISO quality grade, ***G***. (ref fig 1)

Figure 1.



$$e = \frac{U}{m} = \frac{G \times 1000}{2\pi n / 60}$$

e = Mass Centre Displacement (µm)

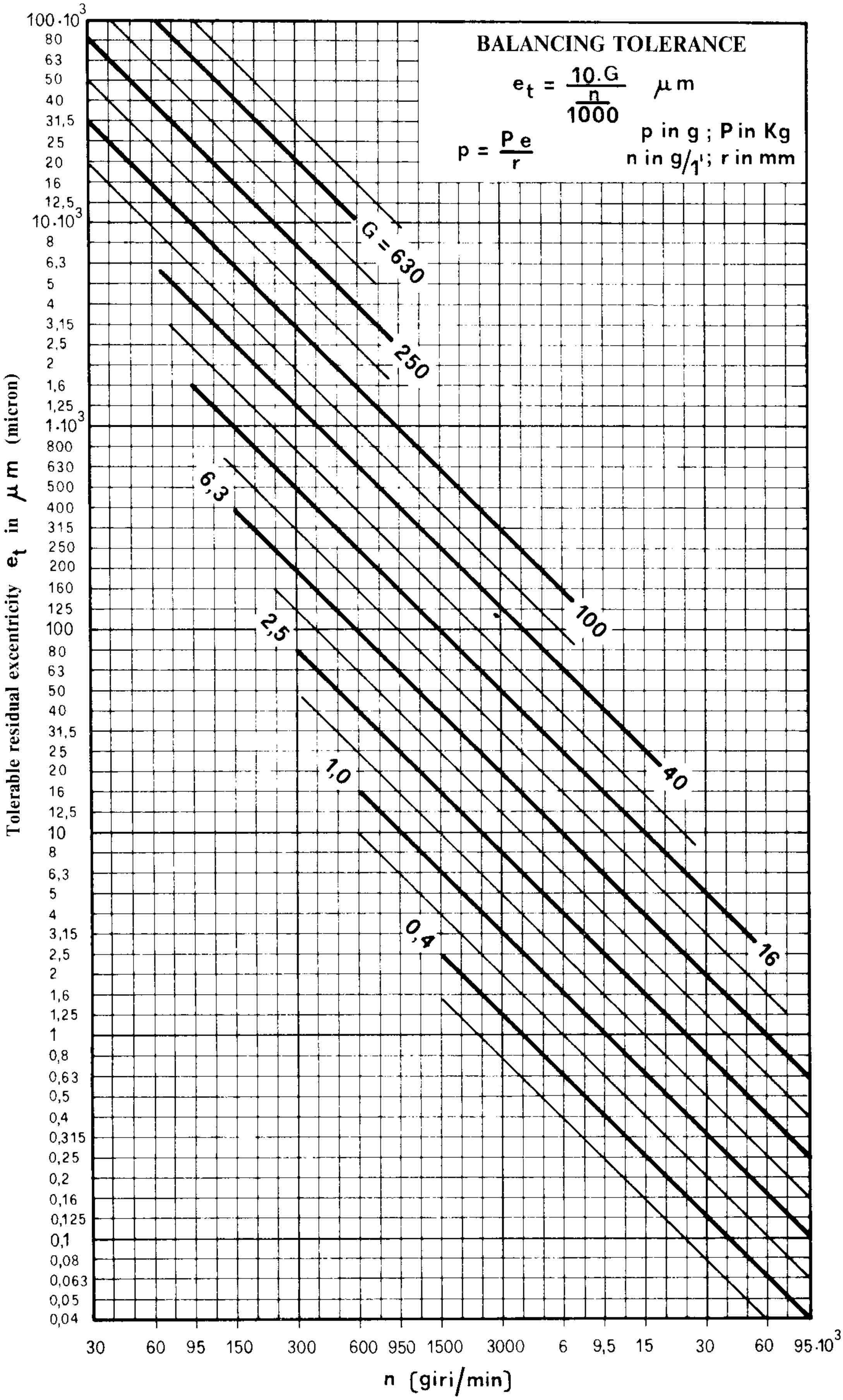
U = Permissible Unbalance (g.mm)

m = Rotor mass (kg)

G = ISO Balance quality grade value (mm/sec)

n = Maximum rotor service speed (RPM)

2.4a Crankshaft Balancing Standards ISO 1940



2.4b Crankshaft Balancing Standards ISO 1940

G Grade	TYPE OF CRANKSHAFT
6,3	Racing car crankshafts, and generally car vehicle shafts definitely requiring very fine balancing.
16	Sports car crankshafts and generally car requiring good balancing. 6 or more cylinder engine shafts requiring accurate balancing.
40	Small car and truck crankshafts with limited balancing need. 4 stroke, 6 or more cylinder crankshafts, elastically suspended, piston speed exceeding 9 meter/sec.
100	Diesel engine crankshafts with 6 or more cylinders, piston speed exceeding 9 meter/sec. One, two or three cylinder, engine shafts.
250	Diesel engine crankshafts with 4 cylinders. rigidly suspended, piston speed exceeding 9 meter/sec.
630	Big 4 stroke engine, diesel marine engine, elastically suspended crankshafts.
1600	2 - stroke crankshafts of larger engines, rigidly suspended.
4000	Diesel marine engine crankshafts, rigidly suspended, no matter how many cylinders, piston speed not exceeding 9 meter/sec.

2.5 Force calculations

One important reason for balancing is that the forces created by unbalance are detrimental to the life of the machine, the rotor, the bearings, and the supporting structure. The amount of force created by unbalance depends on the speed of rotation and the amount of imbalance.

Force (F) generated by imbalance can be calculated from the formula:

$F\ (kg) = 0.01 \times W \times R \times (RPM / 1000)^2$

W = Imbalance mass in grams

R = Radius in centimetres

2.6 Influence of fitting tolerances

One of the most common causes of imbalance is the stack-up of fitting tolerances in the assembly of a machine. Murphy and Sod’s law conspire together to guarantee the tolerances work against the finished quality levels.

Obviously, the heavier the rotor component is, the more critical the tolerance of fit becomes to its shaft for a given speed of rotation. If we consider a symmetrical rotor of weight 1000 Kg and having an offset on its shaft of 0.025mm.

Then at 1000 Rpm, F = 12.5 kg / Bearing and at 3000 Rpm, F = 112.5 kg / Bearing.

The following chart produced by the German Standards Institution (2), distinguishes between quality levels, rotor speed, bearing loads and acceptable eccentricity. It is useful in providing focus on machined fitting tolerances and the associated bearing loads. Whereas the majority of quality guidelines, relate to the vibratory motion produced by the residual imbalance level.

Permissible Mass Centre Displacement and centrifugal force against rotor speed and quality grade (VDI 2060)

Quality class	Speed ranges rpm	Permissible centrifugal force as % of rotor weight	Residual eccentricity in µm
0	10,000	-	0.2
I	7,500 - 10,000	2.5	0.2…0.4
II	5,000 - 7,000	3	0.45…1.0
III	3,000 - 5,000	4	1.44…4.0
IV	1,500 - 3,000	5	5.0…20.0
V	750 - 1,500	6.5	25…100
VI	1,000	-	50…250

3. METHODS

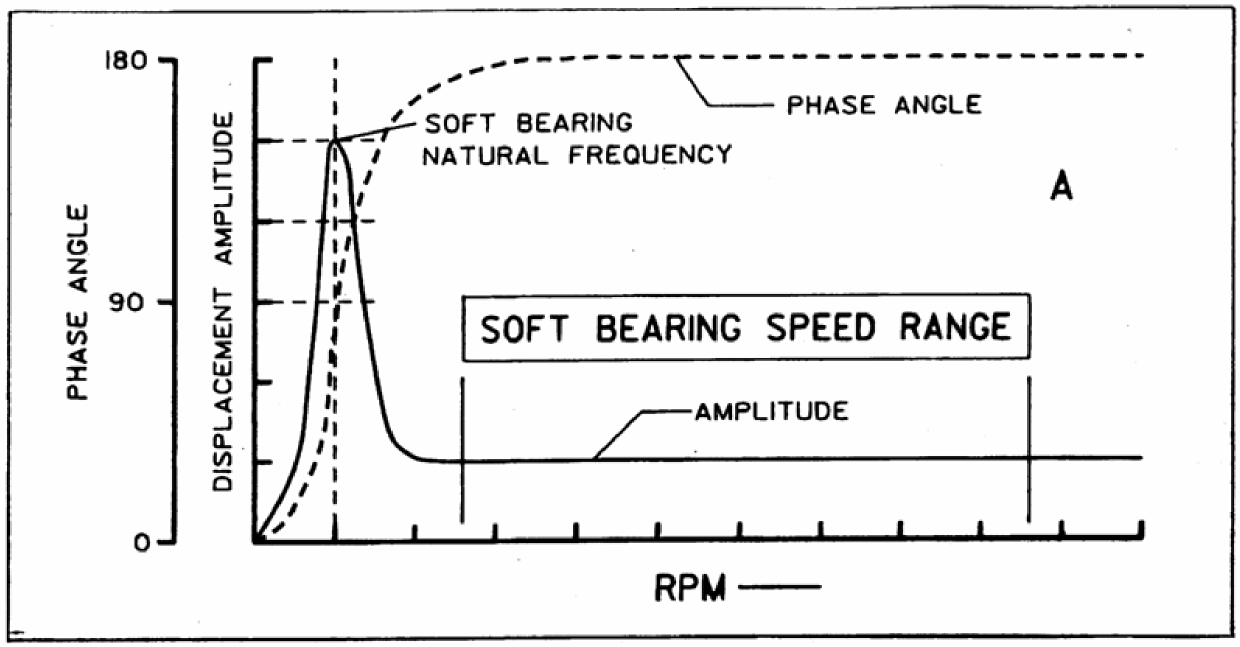
3.1 The traditional approach

Most rotating assemblies or rotor component parts are dynamically balanced in a balancing machine. Balancing machines are generally divided into two types;

- the **“soft”** or flexible bearing machine, measures displacement
- the “**hard**” or rigid bearing machine, measures force

The soft bearing balancing machine derives its name from the fact that it supports the rotor to be balanced on bearings that are free to move in at least one direction, usually horizontally and perpendicular to the rotor axis. The resonance of the rotor and bearing system occurs at one half or less of the lowest balancing speed, (typically 250 rpm). By the time the balancing speed is reached, the measurements of vibration displacement, amplitude and phase have stabilised and can be measured with accuracy. The aim of the soft suspension is to provide a linear response above the minimum balancing speed. (ref fig 2) The suspension supports the rotor as if in free space, to provide unchallenged movement in the direction of displacement measurement. (ref fig 3)

Figure 2.

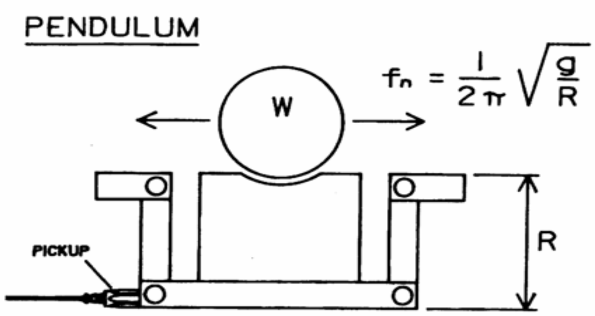


Soft bearing suspensions provide a linear response from low balancing speeds

3.2 Dynamic table tooling

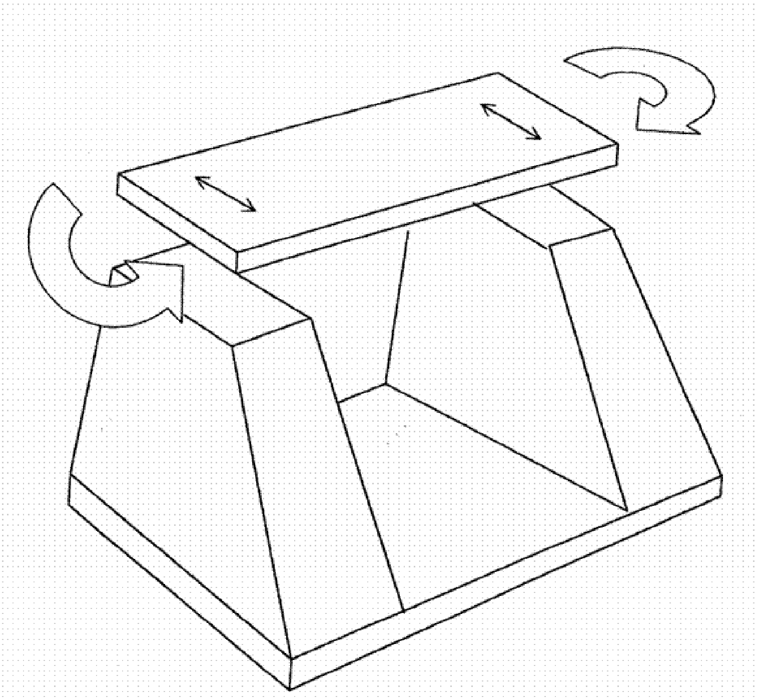
The displacement measurement comes extremely close to the mass centre displacement (MCD) value derived in the balance tolerance calculations. Difficulty arises when the rotor cannot be supported in open rollers on the balancing suspensions without impeding this vitally important "free space" motion. For this reason special tooling has to be designed to support the rotor, allow it to spin and impart its "free" motion to the suspensions. The dynamic table is an arrangement that allows a complete or sub assembly to be mounted onto the suspensions. The design provides for completely free movement of the rotating assembly, regardless of how it is clamped or supported. The dynamic table is an invaluable tool to achieve balancing limits otherwise unachievable because of fitting tolerances, or awkward handling of component parts. The protected linear response of the suspension allows close and accurate scrutiny of the rotating assembly, knowing that the balancing machine is not influencing the motion. (ref fig 4)

Figure 3.



Natural frequency of the pendulum suspension is not affected by the weight of the rotor

Figure 4. Principle motions of the Dynamic Balancing Table



4. CASE HISTORIES

The following examples show 5 applications for the dynamic table and then discuss the reasoning behind this approach.

The cases discussed are: -

• A wind tunnel model wheel (4500rpm)	C & B Consultants
• A high speed drive shaft (9000rpm)	MCL
• A Volvo 5 cylinder racing engine (8,500 rpm)	TWR *Arrows F1*
• A high-speed automobile supercharger (75,000 rpm)	*Force One Superchargers*
• V8 race engine (9600rpm)	MCL

4.1 Case history; Wind tunnel model wheels

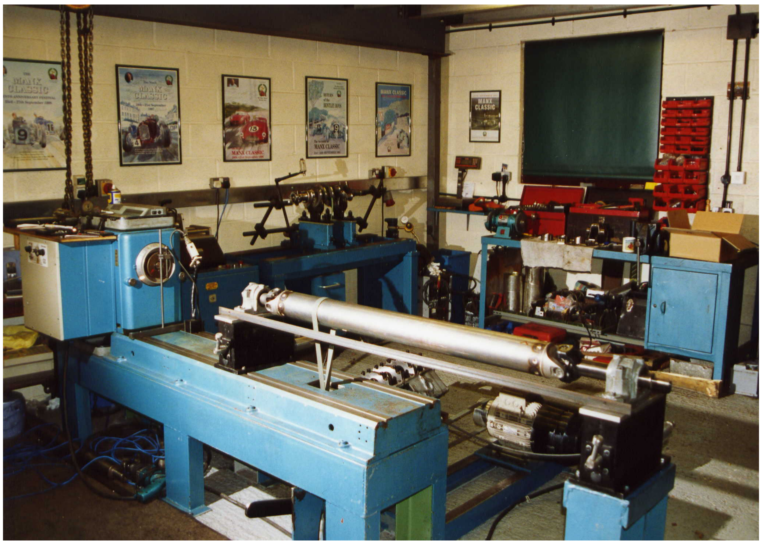
The demands of wind tunnel testing in modern formula have brought about larger scale models and higher test speeds. In a field where measuring movement is critical data, the input from the wheels and suspension can do without interference from imbalance induced motion.

The wheels are very sensitive to imbalance due to their lightness and high speed. The tyres are often made of carbon fibre about an alloy rim. The wheel bearings are relatively small with little separation; this makes the wheels very sensitive to imbalance at their periphery.

The acceptable tolerance is often within 0.1g at the correction radius. The sensitive nature of the wheels requires them to be balanced about their fitted bearings, and check balanced at full speed. It has been found that the bearing clearances and level of carcase deflection are critical to the stability of the balance tolerance.
Without a dynamic table approach to balancing, the sensitive levels could not be achieved. The fitting tolerances and the instability of the rotor between normal balancing speeds and final user speeds would confuse traditional balancing methods.

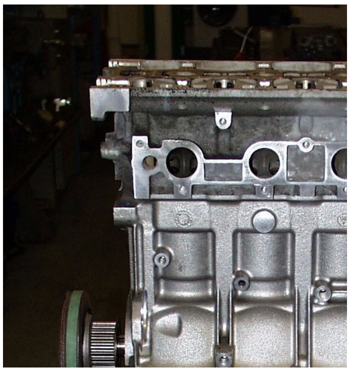
4.2 Case history; High speed drive shaft

The difficulties of balancing drive-shafts effectively for all speeds are not a new problem. Multi-piece drive-shafts are an even greater challenge. Within motor sport the quest for the lightest and stiffest shaft drive is a constant development.



By using a dynamic table approach to the support and testing of the shafts when balancing, two things can be guaranteed; the balance level achieved at any one speed and the stability of the shaft throughout the speed range.
Normally the shaft is balanced at a modest 1500 or 3000rpm, and then the shaft is progressively spun faster throughout its full operating speed. If the shaft is prone to deflect at any specific speed, then this can be monitored by a tracking shaft rider. Balance corrections can then be made accordingly along the shaft length to "hold" the shaft true throughout its driven speed range.

4.3 Case history; Group II crankshaft balancing in engine assemblies



Crankshafts are often complex in shape, part machined and in part left as they were from forging. Inevitably, imbalance that is present generates centrifugal forces that are proportional to the square of the speed of rotation. From a force point of view, consider a crankshaft that has a typical 85mm stroke and a speed range to 6,500 rpm. If the imbalance condition relates to just 1 gram at the stroke radius, it will generate a detrimental cyclic force of 3.5 kilograms. The vibratory forces produced are using energy, which would be better employed doing useful work rather than causing wear and tear. Reducing the vibration will increase power, acceleration and speed; engine parts can have their life extended by anything from 25% to 100%. The automotive industry have balance tolerances that vary from make to make, typically 70 g.mm to 300 g.mm is the allowance on a passenger car crankshaft. The motor racing industry tolerances are much lower, approximately 1/10th of that found in passenger cars.

In practice a mass produced crankshaft may attain a balanced standard of G6.3, relating to 2 grams at the crank throw. It follows that all the pistons and connecting rods must be match-weighed equally well; otherwise the balance tolerance will be lost. Not all crankshafts however are treated equally when it comes to balancing. Many IC engines use crankshafts, which are inherently out of balance. Their axis of inertia does not coincide with the axis of rotation and are classified as group II crankshafts.

These group II crankshafts rely on part of the reciprocating weight to balance the rotating masses. Traditionally, the crankshafts have bob-weights bolted onto their big end journals to simulate the percentage weight of the reciprocating parts. Then they would be balanced by traditional methods on open roller balancing machines. This approach is flawed by the percentage error in the calculation, manufacture and fitting of bob-weights. Additionally, inherent fitting tolerances stack-up on engine assembly i.e. flywheel to crankshaft, clutch to flywheel etc. The resultant levels of vibration, noise, wear, and load attributed to imbalance can be significant.

The relentless quest for power and performance in motorsport is resulting in a new approach by top race teams. The TWR 1988 BTCC championship winning Volvo 5 cylinder engine was one such application.

The engine’s typical balance levels were measured during dynamometer tests and found to be considerably higher than should be expected on a race built engine. Furthermore the resulting vibration movement of the crankcase was found to be exaggerated through critical speed ranges due to the aluminium block, which lacked inherent stiffness. Balancing the whole crankshaft assembly to more accurate and definitive levels, not only reduced power consumption, wear, vibration and noise, but also greatly improved system response throughout the critical speed ranges.

It should be noted that for this engine that rotates up to 8.500 rpm, initial build would leave approximately between 500 and 800 g.mm of imbalance at each end of the crankshaft. Consequently, the respective bearing loads were increased by 40 – 60 kg at full revs and crankcase movement varied between 70 – 100 μm depending on the rpm.

Following a full assembly balance using the dynamic table approach, crankshaft bearing loads were reduced by up to 87 % and crankcase vibration displacements from dynamic unbalance were reduced to below 5μm at all speeds. The largest gains were noticed at speeds around 5000 rpm where critical crank case movement was reduced by 94 %, and produced power levels were measurably improved.

Other similar exercises carried out on race built ‘V’ engines, have shown initial unbalance levels typically to reach in excess of 3.4 kg.mm. This can only endorse the inaccuracy of common ‘bob-weight’ practice used for group 11 crankshaft balancing and highlight the difficulties of mass balancing con-rods accurately.

4.4 Case history; Automotive centrifugal supercharger

Balance quality grades are established against rotor weight and maximum rotor service speed. It follows that the permissible specific imbalance (e) relates to residual unbalance (U) per kg of rotor mass (m), and is equivalent to the rotors (Mass Centre Displacement) at the centre of gravity. For rotors balanced within a specific quality grade, an allowable MCD exists for that speed range irrespective of rotor weight. To double the speed range would effectively half the allowable MCD. This imposes extremely tight fitting tolerances on high speed rotors, such as the example given here with an automotive supercharger.

A new design of centrifugal supercharger runs at speeds of up to 75,000 rpm, requiring tight balance tolerances to be met. Traditionally the majority of turbochargers produced, run in hydrostatic bearings to speeds in excess of 150,000rpm. The supercharger is by its very nature mechanically driven and has to run in ceramic rolling element bearings.

This arrangement running at 75,000 rpm has a permissible specific imbalance (e) of 0.127 µm. No machining tolerances however acute would protect the level of fitting accuracy required.

An assembly balance was the only option available to guarantee finished product quality. Fortunately, since the compressor wheel and shaft were running in their own bearings, a small dynamic table easily supported the arrangement. The rotor balance tolerance was so sensitive to compressor wheel fit, that tests showed if the compressor wheel lock nut had to be retightened for any reason, the balance tolerance would be lost by a factor of 10.

Balancing the assembly on a dynamic table provided the finest balance tolerance achievable and proved its assembly status. The result was a very quiet, vibration free supercharger with longevity to match.

4.5 Current case history; NASCAR V8 Engine crankshaft re design

The present day NASCAR USA racing series run V8 engines close to 10,000rpm. Operating at or close to this level typically produces main bearings failures.

The challenge has been set to produce a crankshaft with lower bearing loads. A solution is being sought by treating the crankshaft as a flexible shaft and balancing it about each cylinder bank. This will effectively balance the crankshaft as four separate twin cylinder engines.

Figure 5. Fully Counterweighted V10 Crankshaft

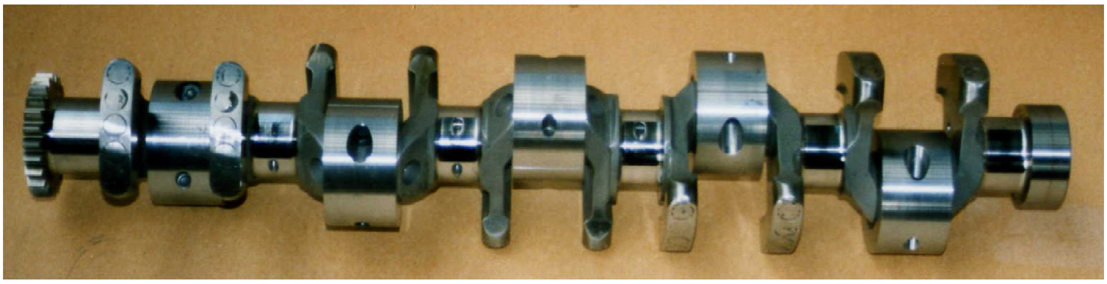


Figure 6. Multiplane Flexible Rotor

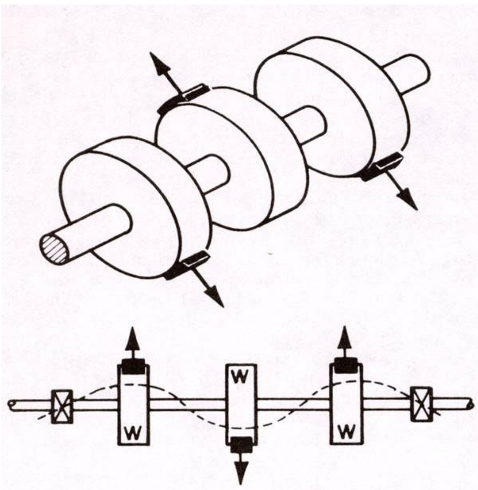
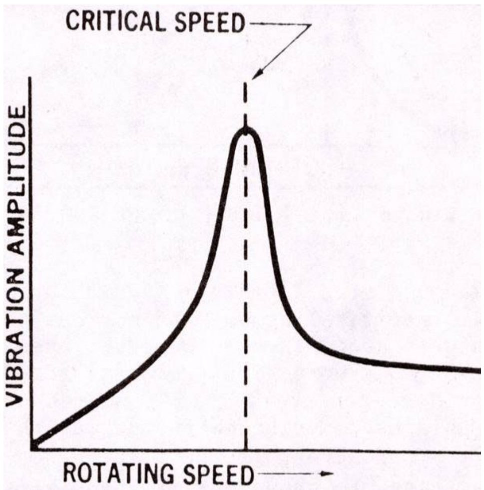


Figure 7. Critical Speed point



4.6 Tradition

Crankshafts are treated as rigid rotors, e.g. rotors that operate below their first critical speed. Due to the manufacturing process, most crankshafts are made with the end counterweights over "heavy". This is to ensure that the drilling of the crankshaft at the balancing station is always in the end counterweights.

In the past, engines in general have had relatively heavy flywheels and clutch assemblies, with equally heavy pulleys and dampers. Modern trends are moving towards higher engine speeds, lighter vehicles, lighter flywheels, clutches and alloy engine blocks.

Consequently, the engines ability to accommodate imbalance is reduced and tighter balancing tolerances are needed. Likewise, if the crankshaft has any deflection forces along its length, then the lack of stiffness in alloy blocks and high engine speeds, promote crankcase deflections and detrimental bearing excursions.

4.7 Crankshaft balancing tolerances

Imbalance is measured typically as the product of a known weight at a known radius, e.g. 1gm at 100mm radius produces 100gmm.

Crankshafts have a certain length to them, so tolerance is measured at each end.

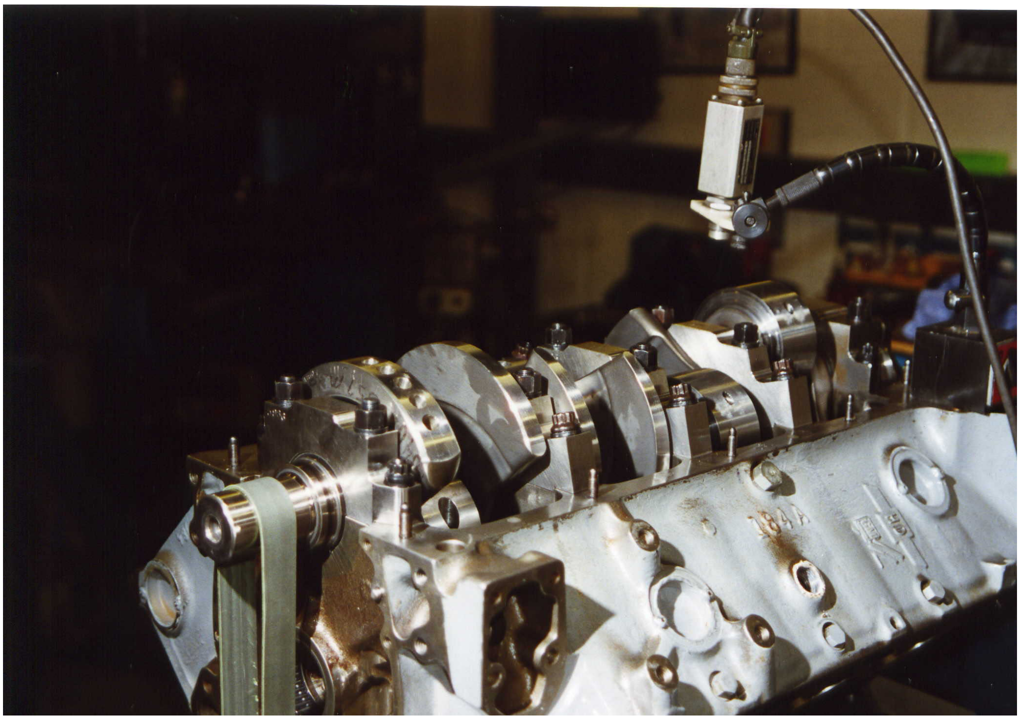
Typical values would be;

- 80gmm – 300gmm (production car engines)
- 60gmm – 100gmm (sports car engines)
- 40gmm (touring racing car engines)
- 26gmm (F1 racing engine)

What actually decides the tolerance for any particular engine can be dependant on several factors;

- Historical limits
- Vehicle refinement
- Engine performance
- Noise limitations
- Reliability issues e.g. electrical connections and components.

Figure 8. V8 Engine on Dynamic Table



The traditional crankshaft balancing machine uses “open” rollers to run the crank journals on. It measures values of either force or movement to give a calibrated degree of imbalance. Certain crankshaft types require pin – weights to be fixed on the big end journals during the balancing

operation. These pin weights have to be “calculated” and represent the portion of the conrod and piston weight needed to balance the crankshaft.
It follows that the pin – weights must be accurately calculated; manufactured and fitted otherwise an imbalance will result. (ref case history 3)

The balancing machine can be adapted to measure imbalance of an assembled engine. This has “real” benefits in that it proves an assembly tolerance and that it is significantly more cost effective and accurate than pin weight methods.

The balancing machine features which enable this method are as follows;

- The dynamic table facilitates all aspects of imbalance motion
- The suspension is linear in its response
- The suspension is soft, i.e. measures displacement
- Instrumentation has accurate filtering and high resolution

Vibration Free have used these methods for many years to balance awkward rotors and assemblies. They have been used to calculate; pin–weights by empirical means, prove the need for accurate con–rod mass centring, balance satellites at 50 rpm, turbochargers and high speed assemblies at many thousands of rpm.

One of the first interesting applications within current motor sport engine building was when one leading team asked for an assembly check balance on a simple four cylinder engine.

This should have been unnecessary as all parts had been component balanced with conventional methods.

However, there were still some significant imbalances within the engine. These proved to be from the con-rod mass centring.

It is quite common for modern billet con-rods to be fully machined and match balanced as a total weight, yet not match balanced end to end. The manufacturers believe it to be unnecessary. Tests at Vibration Free have shown that on leading con-rods, a difference of up to 8gms end to end can be present. **To prove this out Vibration Free developed a method to measure con-rod mass centre to within 0.001&m and end to end imbalance within 0.001g.**

The next most significant issue was assembly balancing V8 and V6 engines and discovering that small changes in piston set weights greatly affected overall balance levels. This suggested that the counterweight sizing was critical.

Previous tests on the Volvo touring car five cylinder engines showed that fully counterweighted crankshafts, although heavier than the standard shaft, gave reduced bearing loads and increased power. Therefore the single crank throw test was carried out on crankshafts to establish the accuracy of counterweights.

A few years ago the Rover 1.8ltr “K” series engine was challenged to find out that it had a significantly under counterweighted crankshaft. **The imbalance present represented a force of 968kg at 8200rpm about each cylinder throw!**

A fully counterweighted crankshaft was produced albeit heavier than the original and matched to an equally lighter flywheel and clutch assembly. This produced a much improved engine in performance and refinement.

One of the current projects concerns the NASCAR V8 engines. The performance and reliability issues are being challenged at engine speeds approaching 10,000 rpm.

Main bearing failures are the common failure mode. MCL at Leafield, Oxon, UK is challenging the engine further and has enlisted Vibration Free to carry out the counterweight tests accordingly.

The V8 crank has a symmetrical layout about its centre mains, although the two end throws have larger counterweights than the centre ones.

Figure 9. Single throw crank with pin-weight

Utilising a slave V8 block supported on the dynamic balancing table. The initial test qualified the residual imbalance at the end crank throws, using a bob weight to represent the portion of con-rod and piston weight required. This showed that the counterweights were under weight by 300gms at a radius of 65mm.

The pistons and rods were then fitted in place of the bob weight. This produced an unbalance level of 280gms, showing the bob weight was inaccurate by 20gms. However, this level of imbalance was significant producing forces close to 1800kg at 9600rpm!

A similar test was then carried out on the centre counterweighted throws, this produced levels of 425gms unbalance at 65mm radius. This would generate a cyclic force of 2,500kgs at 9600rpm!

The next stage will be to spin test a full size crank with tungsten heavy metal added. With the counterbalancing more accurately compensated, the spin test should show the power required to drive the crank within the block and bearings.

Now that the empirical test has proved out the forces about the main bearings, the modelling software showed a significant reduction in bearing loads by 20%.

At present the existing crankshaft is not capable of providing the required counterbalancing along its length. The next generation of crankshaft will be suitably redesigned to carry the required level of tungsten counterbalancing within the engine.

This will produce a heavier crankshaft and increase its moment of inertia. Therefore weight savings must be made else where on the crank line to compensate. Fortunately on the NASCAR engine, the working rev range is relatively small, so it is not so critical. This would not be so for an engine with a broad speed range.

4.8 Summary

- Empirical tests can prove out simulation and calculated modelling.
- Fast results are obtained by relatively simple tests.
- Refinement of design facilitates accurate balance methods.
- Performance gains are made through reduced vibration, bearing loads, power consumed and increased reliability.

5. CONCLUDING DISCUSSION

The need for a dynamic balancing table solution has often been driven by production directives that had no alternatives. The table has proved to have a variety of uses. The ability to handle complete assemblies leads to many in-service applications that save on costly strip-downs and rebuilds. This combined with linear response suspensions, allow for optimum balancing or dynamic assessment of the assembly.

There are many benefits: -

- It can provide a qualified balance unaffected by fitting tolerances.
- It may be the only way to balance an awkward rotor arrangement.
- Pre-production tooling costs can be kept to a minimum.
- It allows low speed balancing of an assembly.
- The finest balance tolerances can be achieved.
- It can be the most cost effective solution to a difficult task.

However, the underlying reason for its production need is brought about by a lack of timely attention to balance criteria, leading to focus on manufacturing tolerances with consequent costs to quality and value of the finished product. It would be better to allow the manufacturing process to dictate manufacturing tolerances and plan for final balancing of the prototype or production item after assembly. This strategy requires good provision for balancing efficiently and economically. It has a better track record than chasing machined tolerances which may well turn out to be unnecessary.

It is false economy to balance sub-assembly components to small tolerances, if the assembly will not hold the required limits. Too often, expensive and time consuming rework is carried out in an attempt to chase unbalance, vibration, noise and failure issues on the finished product.

It is common for balancing tolerances to be specified on a manufacturing drawing without due thought or for the wrong reason. Tolerances are frequently read across from other similar jobs or projects. Alternatively the balance tolerance is tightened up with the mistaken idea of producing a better product (with less noise and vibration). It does not necessarily follow that setting a tighter tolerance limit will achieve the objective; what really matters are the ability to achieve a limit consistently and as cost effectively as possible during the production run.

Balancing considerations should commence in the design office. It is therefore surprising how little attention is being paid to balancing at the design stage. Production costs are reformed if the compensating process is addressed in the balancing strategy at the design stage. The act of balancing a component efficiently can only be carried out swiftly when the means of correction has been purposefully designed.

It is well known that measuring the magnitude and position of unbalance takes only a few seconds, whereas compensation frequently requires minutes or hours. Although, with thought and planning compensation times can be achieved which only fractionally exceed the measurement time.

Drawings need to specify the method and position of balancing corrections. They need to state clearly the tolerance allowed. Ambiguous instruction or none at all is a costly practice and makes it almost impossible for the balancing engineer to understand what is required. Advice or consultation from a balancing engineer when drawing up specifications will reap rewards at this stage.

All drawing specifications for balancing need to contain the following information;

- The number and position of the balancing planes.
- The method of mass compensation.
- The permissible residual unbalance ***U*** in g.mm or a quality grade ***G*** with associated rotor speed and weight.

If the component to be balanced is part of a subsequent assembly, then the fitting tolerances have to be taken into account. It would be futile to specify a permissible residual imbalance of e = 5µm for a fan impellor if the tolerance between bore and spindle is 30µm.

In such cases imbalances can appear after assembly, which are larger than a multiple of component tolerances, thus the time consuming procedure of fine balancing was wasted. It follows that balancing of the complete assembly is essential where high precision must be achieved and the designer must make the necessary provision for this.

Consideration for levels of product vibration, noise and performance should incorporate balance criteria at the design stage. Not, as often is the case, be finalised once the manufacturing process has started. Early evaluation tests can conclude the influence that balance grades will have on finished product quality. Should a particular product or design benefit most from a final assembly balance, then this should be accounted for throughout the manufacturing process to reduce costs and qualify build quality. It is therefore necessary to consider the economics of each individual case.

6. CONCLUDING SUMMARY

- Consultancy support aids design and product economies and effectiveness
- The Dynamic Balancing Table contributes to efficiency and sensitivity
- Empirical tests can defy software modelling and calculation
- IRD Balancing machines and instrumentation allow accuracy and resolution

7. ACKNOWLEDGEMENTS / REFERENCES

7.1. ISO 1940/1-1986, "Balance Quality Requirements of Rigid Rotors." International Standards Organisation.

7.2. VDI 2060 – 1966, "Criteria for assessing the state of balance of rigid rotating bodies." German Standards Institute.

The Balancing Act

Steve Smith

Vibration Free

www.vibrationfree.com

Design for Balance
Or
Balance the Design ?

www.vibrationfree.com

Vibration & unbalance is detrimental to power, performance and longevity

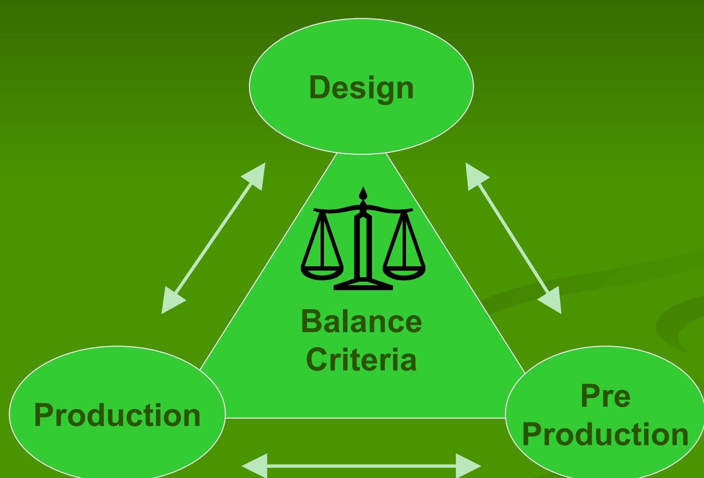
www.vibrationfree.com

The Manufacturers Dilemma

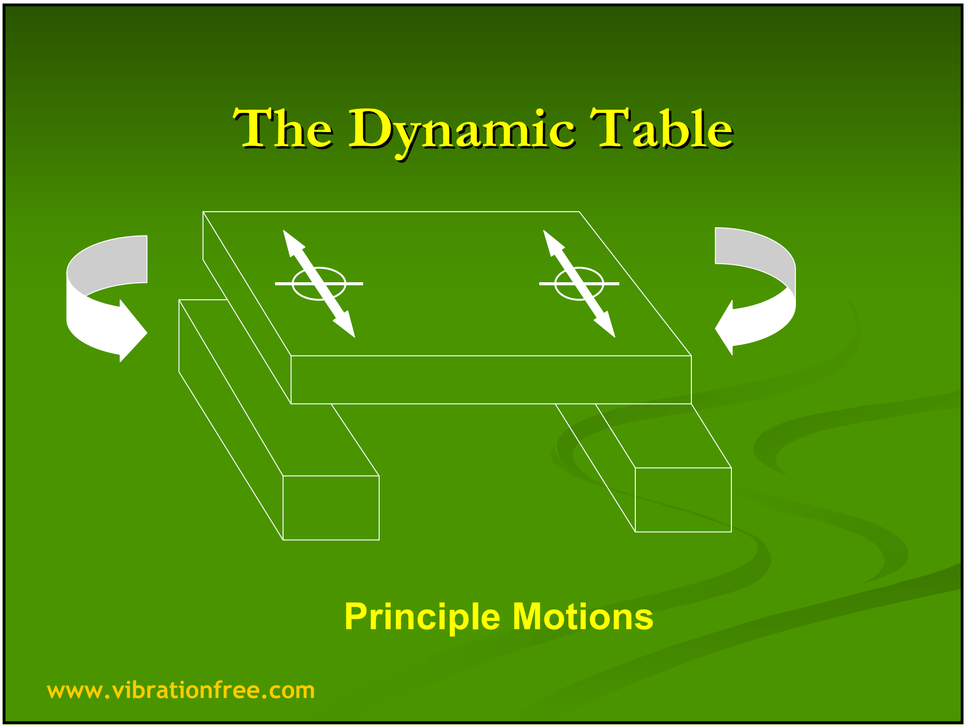
- SPEEDS
- EQUIPMENT MANUFACTURERS
- EXPECTATIONS

- COSTS
- TIME
- TOLERENCES

The Manufacturing Process

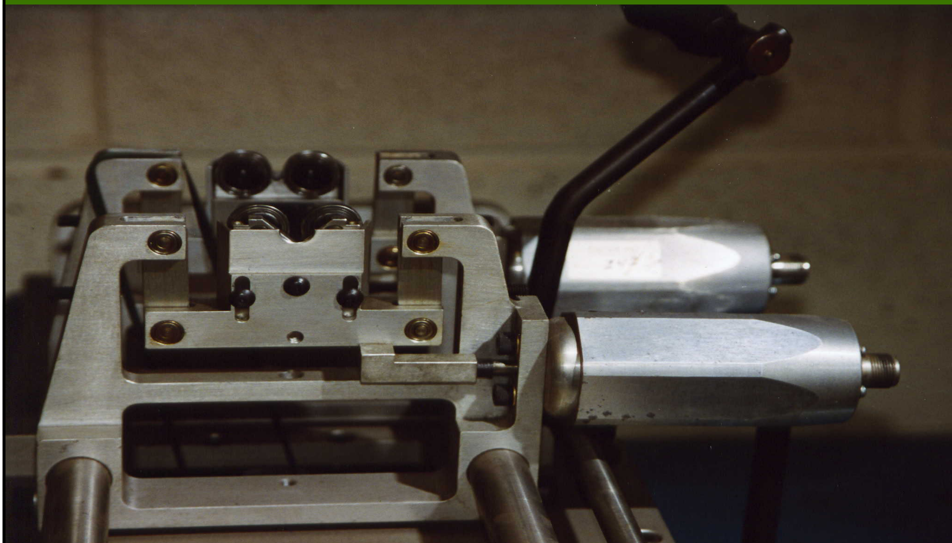


www.vibrationfree.com


The Dynamic Table
Principle Motions
www.vibrationfree.com

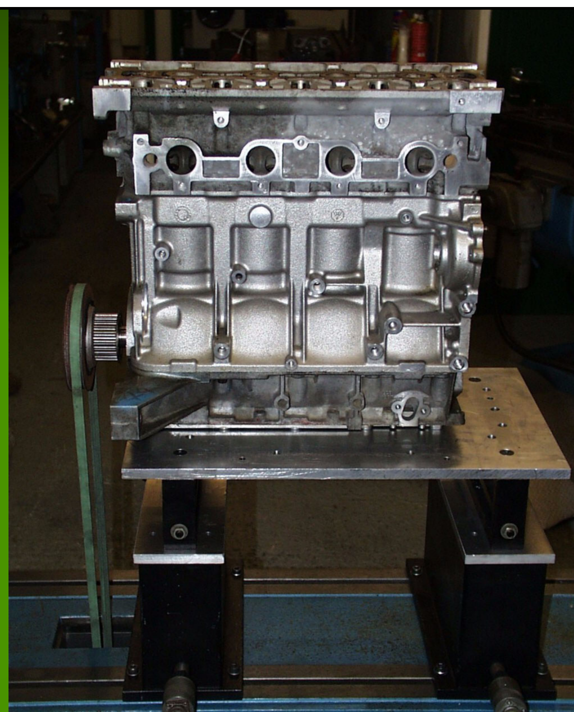

www.vibrationfree.com

Linear Displacement Suspension



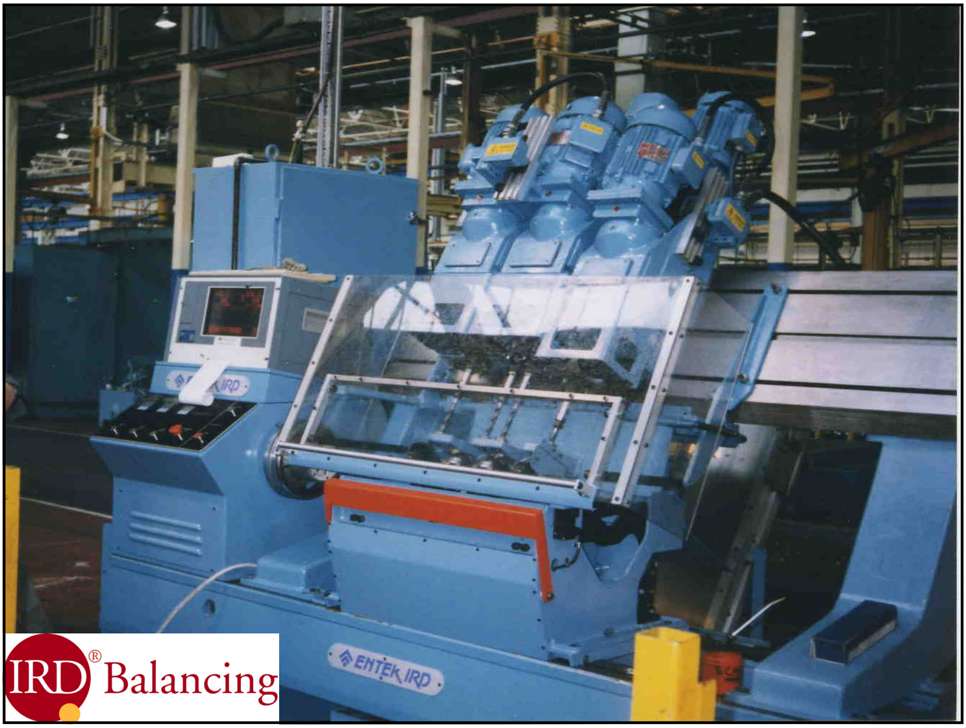
www.vibrationfree.com

Applications



- Turbochargers
- Model wheels
- Drive-shafts
- Multi-cylinder & Vee engines
- Any assembly

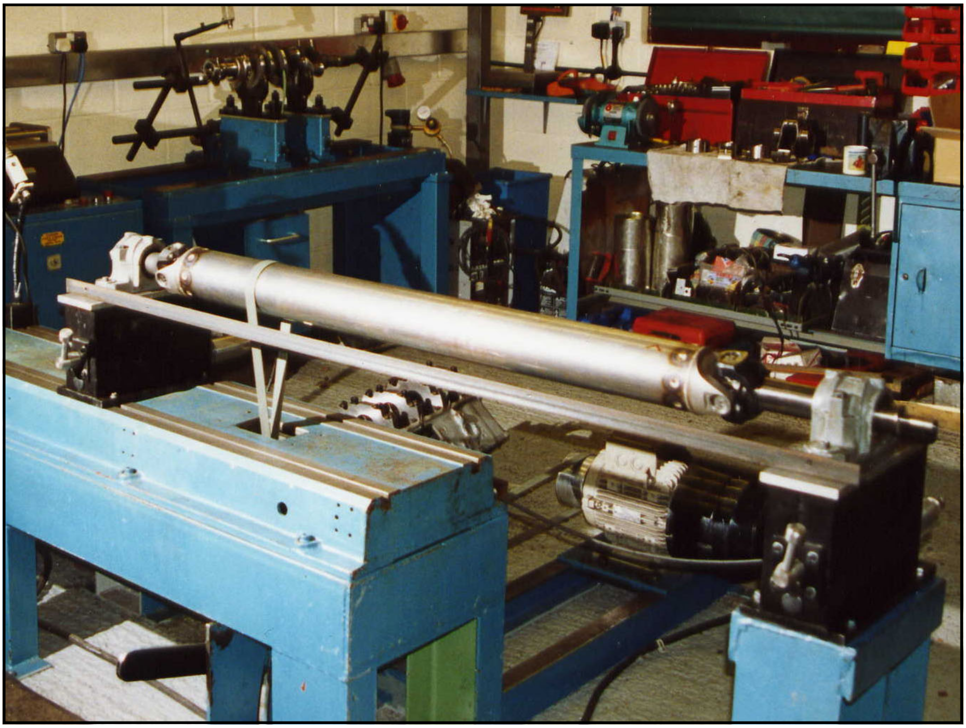
www.vibrationfree.com

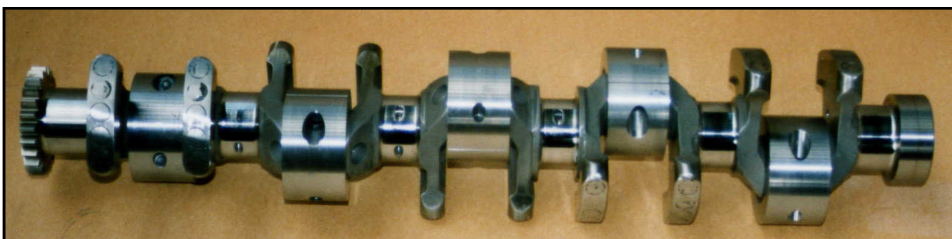

IRD® Balancing
ENTEK IRD

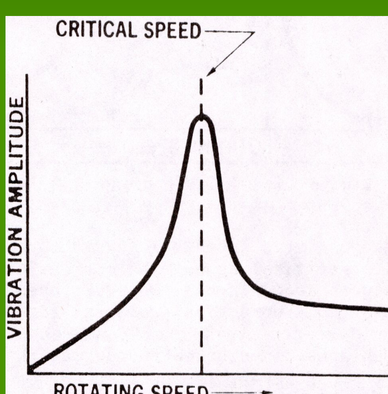

DUPONT
24
8
Budweiser
www.vibrationfree.com

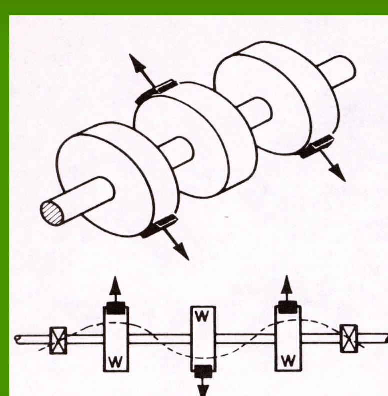

MEL


Vibration Free







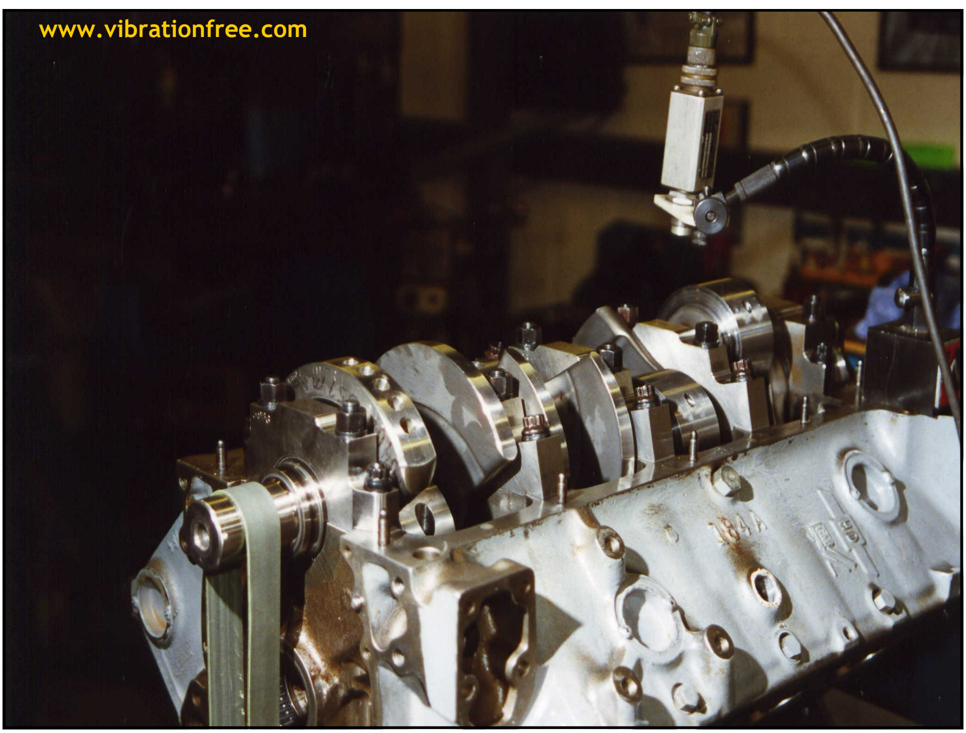


www.vibrationfree.com

Crankshaft Balancing Tolerances

- Typical values for race engines
- 80gmm – 300gmm (Production cars)
- 60gmm – 100gmm (Sports cars)
- 40gmm (Touring car race engines)
- 26gmm (F1 race engines)

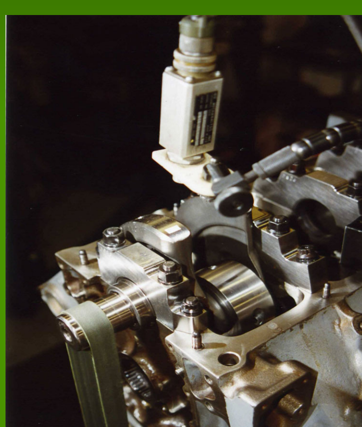
www.vibrationfree.com

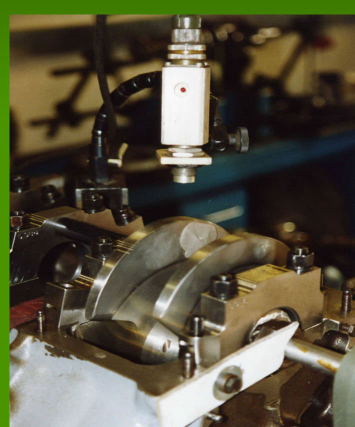


Calculated Pin Weights

www.vibrationfree.com

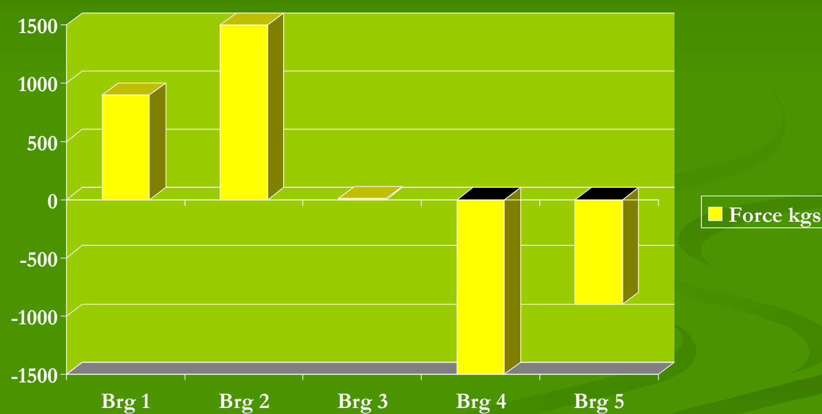
Single Throw Tests





www.vibrationfree.com

Crank bearing forces at 9600rpm

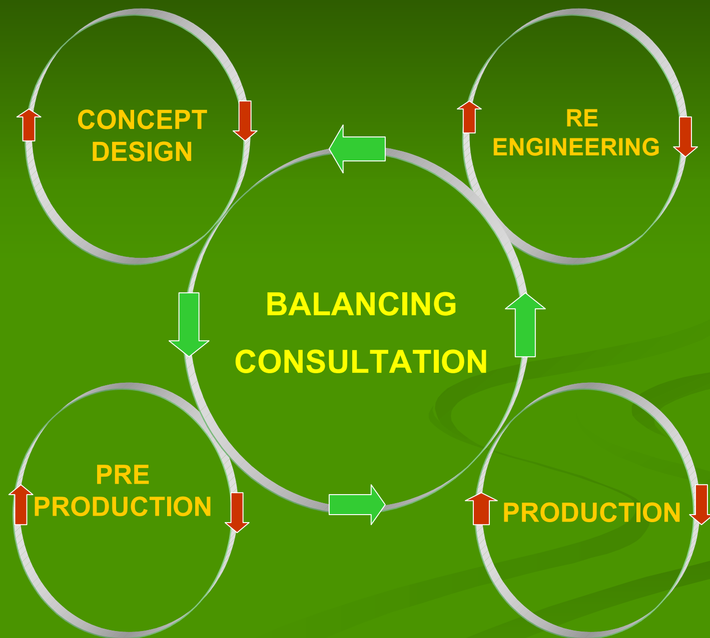


www.vibrationfree.com

Balance Influence

- Design
- Production
- Performance
- Cost

www.vibrationfree.com



Question Time