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Contents

**The Fundamentals of Industrial Balancing Machines
and their Applications**

by J. F. G. Wort 3

News from the Factory29

The Fundamentals of Industrial Balancing Machines and their Applications

by

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ABSTRACT

An introduction to the principles of rigid rotor balancing facilitates description and comparison of dynamic balancing machine types. Many factors influence the choice of machine to suit a given task and are discussed briefly. The extension of the technology to field balancing and to flexible rotor balancing is also outlined.

SOMMAIRE

Une introduction aux principes d'équilibrage des rotors rigides, facilite la description et la comparaison des différents types de machines d'équilibrage dynamique. Plusieurs facteurs influencent le choix d'une machine pour un travail donné; ils sont discutés brièvement. L'extension des techniques d'équilibrage sur le terrain et d'équilibrage de rotors flexibles est également soulignée.

ZUSAMMENFASSUNG

Eine Einführung in die verschiedenen Wuchtmethoden mit Beschreibung und Vergleich verschiedener Wuchtmaschinentypen. Viele Faktoren spielen bei der Wahl der Maschine eine Rolle und werden kurz beschrieben. Ebenso ist die Erweiterung der Technologie auf Betriebswuchten und flexible Rotoren erwähnt.

Introduction

Unbalance is a major source of vibration in all types of equipment with rotating elements. Typical examples are crank-shafts, cardan shafts, electric armatures, turbo-machinery, domestic equipment printing rollers, centrifuges, and gear wheels. Some common causes are machining error, cumulative assembly tolerances distortions due to heat treatment, blow holes or inclusions in castings, material non-homogeneity,

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and asymmetric components (e.g. camshafts). The increase in operational speed of many machine designs in recent years has exacerbated the problem, and has made a careful evaluation of unbalance factors vital.

The prime advantages to be gained by improving the balance quality of a rotating element, or rotor, in terms of its consequences on the assembled product can be summarised:

- Reduced internal wear of machine seals and bearings,
- Reduced likelihood of fatigue failure in ancilliary housings and brackets, or loosening of fastenings,
- Reduced transmission of vibrations to the foundations, and consequent radiated noise
- Reduced requirements for structural sturdiness, with consequent material saving
- Lengthened product life
- Improved customer relations in the face of increasingly severe consumer protection laws and environmental pollution legislation

Modern electronics and transducer techniques have greatly improved the efficiency and precision of industrial balancing machines, but the selection of appropriate equipment for a new balancing installation, or for updating existing facilities, can be bewildering. This article attempts to give an introduction to balancing technology, and to explain the functions of common types of industrial balancing machines used for dynamic balancing.

Defining unbalance

The classic unbalance model comprises a perfect disc, homogeneous and isotropic, on a slim axle. Any small uncompensated mass located in the plane of the disc, but offset from the shaft axis, represents unbalance. In practice, permissible unbalance is related to the mass of the disc, so it is convenient to introduce the term specific unbalance, \bar{e} , expressed as:

$$\bar{e} = \frac{m\bar{r}}{M} \quad \text{g mm/kg} \quad (1)$$

where \bar{e} , \bar{r} are vectors and m = mass of uncompensated mass in g
 r = radius of uncompensated mass in mm
 M = mass of disc in kg

The uncompensated mass, m , produces unbalance forces when the shaft rotates, such that for a rotation speed of ω rad/s the centripetal force required to resist the unbalance force may be expressed as:

$$\bar{F} = m\bar{r}\omega^2 \quad \text{g mm/s}^2 \quad (2)$$

Considering expression (1), (2) can be rewritten in terms of \bar{e} , which is also seen to have units of length, μm , and represents the displacement of the centre of mass. Thus:

$$\bar{F} = M\bar{e}\omega^2 \quad \text{g mm/s}^2 \quad (3)$$

Expressions (2) and (3) illustrate a duality in the description of unbalance in a rotating body or *rotor*.^{*} Unbalance effects may be reacted by elastic forces in the support bearings, by inertia forces in the rotor itself, or by some combination of the two. The unbalance force acts radially and always occurs at the rotation speed. It can be seen from (2) that the unbalance can be measured directly as a force or from (3) indirectly by the resulting displacement at the centre of mass, under suitable constraint conditions discussed in section "The Rotor Support System".

The simple unbalance model, illustrated in Fig.1(a) is called *static* unbalance. In principle its presence can be detected by placing the *rotor* shaft on a pair of knife edges to locate the heavy spot.

Most practical rotors are not discs, but have mass distributed along a length of their shafts. Such rotors may appear balanced when placed on the knife edges, yet produce unbalance forces in service. Pure *couple* unbalance produces a wobbling effect about the centre of mass, and may be considered as an inclination of the inertia axis as shown in Fig.1(b).

The general condition when both static and couple unbalance are present is called *dynamic* unbalance, as shown in Fig.1(c). The principal inertia axis is now inclined to the geometric shaft axis but there is also an eccentricity at the centre of mass. This condition requires correction in two planes. *Balancing* describes the procedure by which the mass distribution of the rotor is adjusted so that its principal inertia axis is aligned with the axis of its shaft.

* Terms in italic type are those recommended in ISO-1925 "Balancing Vocabulary".

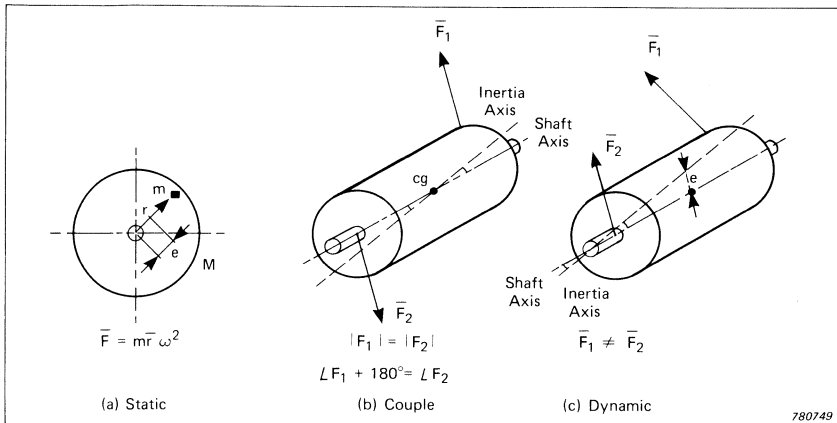


Fig.1. Unbalance types

Recognising the empirical observation that the physical consequences of unbalance are proportional to speed, it may be deduced further that for any given unbalance severity:

$$e\omega = \text{const.} \quad (4)$$

This relationship, also based on the specific unbalance or centre of mass eccentricity concept allows the requirements for similar rotor types to be grouped together in different balance quality grades. ISO-1940 "Balancing Quality of Rotating Rigid Bodies" is typical of current

Quality grade G (ISO-1940)	Example with service speed (RPM)	Equivalent CG eccentricity (μm)
0,4	Reference gyro 24 000	0,15
1	Precision electric motors 6 000	1,6
2,5	Fractional HP electric motors 3 000	8
6,3	Larger electric motors 1 500	40
40	Vehicle engine crankshaft 6 000	60

Table 1. Concept of unbalance quality*

* As quoted in ISO-1940, VDI-2060. Many national standards have been based on ISO-1940, either directly or in the language of the respective country. Examples are ASA std. 2-1975 (ANSI S2.19-1975) and BS 5265, Pt. 1.

standards using such a categorisation: in this standard the grades are given the designation G. The requirement for balance quality differs widely for different types of machine. Some examples are shown in Table 1, related to typical service speeds.

The concept of a balancing machine

A dynamic balancing machine consists essentially of a base plate assembly and an associated console or display unit. Instrumented support pedestals carry the rotor which is driven at constant speed by a motor and drive system.

Unbalance effects are sensed at the support bearings and the electrical signals derived are transmitted to the console unit. Here the state of unbalance is analyzed to enable calibrated correction values to be displayed on the front panel. The console unit is the nerve centre of the system: it should give the operator sufficient information to enable the *residual* unbalance to be reduced within target unbalance *acceptability* or *tolerance* limits. Typical recommended residual unbalance levels for common types of rotating machines are listed in VDI-2060 and ISO-1940 "Balance Quality of Rotating Rigid Bodies".

For a *rigid* rotor, measurements made on any two transverse *measuring planes* are sufficient for derivation of the information required for unbalance correction by addition (or removal) of mass in any two convenient *correction planes*. If the rotor can be balanced to within the required *tolerance* at any convenient balancing speed, it can be expected to remain within these tolerances up to maximum service speed. (The requirements for "balancing" or "dynamic straightening" of *flexible* rotors operating at service speeds above 50% of their first critical speed is introduced briefly later — in this state the balance condition becomes speed dependent due to bending deformation of the rotor itself.) There is not usually any merit in separating components of static and couple unbalance, thus the description *dynamic balancing machine* is used to describe a machine with two support bearings to enable two-plane balancing; it can also be used for single plane balancing. Many two-plane machines operate with the rotor axis horizontal, and are described as "universal". The support pedestals can be locked at any convenient position along the baseplate to suit different rotor geometries within the mass range, and the console is designed to accommodate any likely combination of measuring and correction planes presented to it.

For practical rotors the correction planes selected will depend on the physical form and on constructional features of an individual rotor. Thus the range of adjustment available for the support pedestals, and the way in which the *amount* and *angle* of unbalance is displayed will, to a large extent, govern the application and utility of a balancing machine.

Vertical axis balancing machines are also in common use for single-plane balancing of disc-shaped rotors, such as brake-drums, brake-discs, flywheels, clutch assemblies etc. The basic measuring philosophy is as discussed for dynamic balancing machines, whilst the analysis is simplified considerably as couple effects are not considered, so this type is not discussed specifically.

The advantage of constructing a balancing machine of these types is that the measurement and correction of unbalance can be optimised for one (or more) rotor type using a standard procedure. A high work rate can be achieved, particularly where all rotor handling and correction (e.g. drilling, milling) is fully automated. Such installations are to be found in car production plant for crankshafts, cardan shafts and brake components, and mass production of electric motors. Rates of up to 40/hour might be achieved for smaller rotors such as turbocharger elements. At the other end of the scale there remains the individual order and series production situation for which a simpler manual or semi-automatic system is quite adequate.

The rotor support system

The two pedestals which support the rotor *journals* must incorporate some kind of *bearing*. Typical types in use include V-blocks, open rollers, or plain half-bearings; alternatively, the supports may be designed to accept the rotors own bearings. The supports have a defined radial direction, usually vertical or horizontal, in which the measurements are made to derive the unbalance information.

Considering the mass acting at one bearing, the rotor/support assembly may be modelled as a single degree of freedom system. The differential equation of motion can be expressed:

$$(M + m_p)\ddot{x} + C\dot{x} + Kx = m|r|\omega^2 \quad (5)$$

For sinusoidal motion the displacement x can be written:

$$x = X \sin (\omega t - \phi) \quad (6)$$

which gives the solution:

$$X = \frac{mr \left(\frac{\omega}{\omega_0} \right)^2}{(M + m_p) \sqrt{\left[1 - \left(\frac{\omega}{\omega_0} \right)^2 \right]^2 + 4 D^2 \left(\frac{\omega}{\omega_0} \right)^2}} \quad (7)$$

where

$$\omega_0 = \sqrt{\frac{K}{(M + m_p)}} \quad D = \frac{C}{2 \sqrt{K (M + m_p)}}$$

and: m_p = *parasitic* mass (support system, bearings, transducers, etc.)

ω_0 = natural frequency of rotor

K = stiffness of support system in N/m

C = damping of support system in N/ms⁻¹

X = zero to peak motion in μm

ϕ = phase lag between $|r|$ and X .

The rotor motion in the bearings can be expressed as a proportion of the specific unbalance to give:

$$\frac{X}{|e|} = \frac{M \left(\frac{\omega}{\omega_0} \right)^2}{(M + m_p) \sqrt{\left[1 - \left(\frac{\omega}{\omega_0} \right)^2 \right]^2 + 4 D^2 \left(\frac{\omega}{\omega_0} \right)^2}} \quad (8)$$

$$\tan \phi = \frac{2 D \left(\frac{\omega}{\omega_0} \right)}{\left[1 - \left(\frac{\omega}{\omega_0} \right)^2 \right]}$$

This solution indicates that there are different regimes of interest, depending on the relative values of the rotor rotation velocity (ω), and the natural frequency of the suspension system (ω_0).

Where normal balancing speeds are much lower than the support system resonance ($\omega < \omega_0$), the balancing machine is described as having *hard* bearings. The unbalance forces in the rotor are reacted elastically

at the pedestals and are in phase with the displacement, whilst the linear motion is restrained to such an extent that inertia and damping terms may be neglected. In a hard bearing machine sensors are installed to give an output proportional to the force required to resist the motion of the rotor. Equation (8) is simplified to give:

$$M |e| \omega^2 = KX \quad (9)$$

Where normal balancing speeds are much higher than the resonance frequency of the rotor and suspension system, the machine is described as having *soft* bearings. The inertia forces in the system are now much larger than the elastic restraint exerted by the suspension, which may be neglected. The unbalance force in the rotor and the inertia force $m \times \omega^2 = m \times a$ are in **anti-phase**. In a soft bearing machine, sensors are installed to measure the linear motion of the support bearing whilst offering the minimum restraint to radial motion. The lower limit on the measuring capability of this type of machine is governed by the *parasitic mass* (m_p), which comprises all the moving elements of the support pedestal, including springs, bearings and transducers, with the exception of the rotor mass itself. Neglecting the damping term (D), equation (8) can be expressed as:

$$\frac{X}{|e|} = \frac{1}{\left(\frac{M + m_p}{M}\right) \left[\left(\frac{\omega_0}{\omega}\right)^2 - 1 \right]} \quad (10)$$

This can be further simplified assuming $\omega_0 \ll \omega$ to give:

$$\frac{X}{|e|} = - \frac{1}{\left(1 + \frac{m_p}{M}\right)} \quad (11)$$

It can be seen that if the sensors measure bearing motion (X), as representative of centre of mass eccentricity ($|e|$), then the ratio between parasitic mass and the rotor mass (m_p/M) should be minimised to achieve the highest sensitivity.

Where the balancing speed coincides with the rotor/suspension natural frequency ($\omega = \omega_0$), the elastic and inertia forces which are in anti-phase are of equal magnitude. The magnitude of the radial displacement is now governed by the damping term (D). Where damping is low, the dynamic magnification effect can produce large radial displacement.

ments. In the past, *resonance* machines have offered the advantage that these displacements can be measured directly using mechanical measuring instruments. The considerable difficulty in ensuring stability and repeatability in this regime, together with the improved performance and stability of electronic instrumentation, has focused modern design on the more stable above- or below-resonance designs. Note that the descriptions of the various machine types refer to the dynamics of the suspension system supporting the work-piece and not to the type of journal bearings used. The different operating regimes are shown in Fig.2.

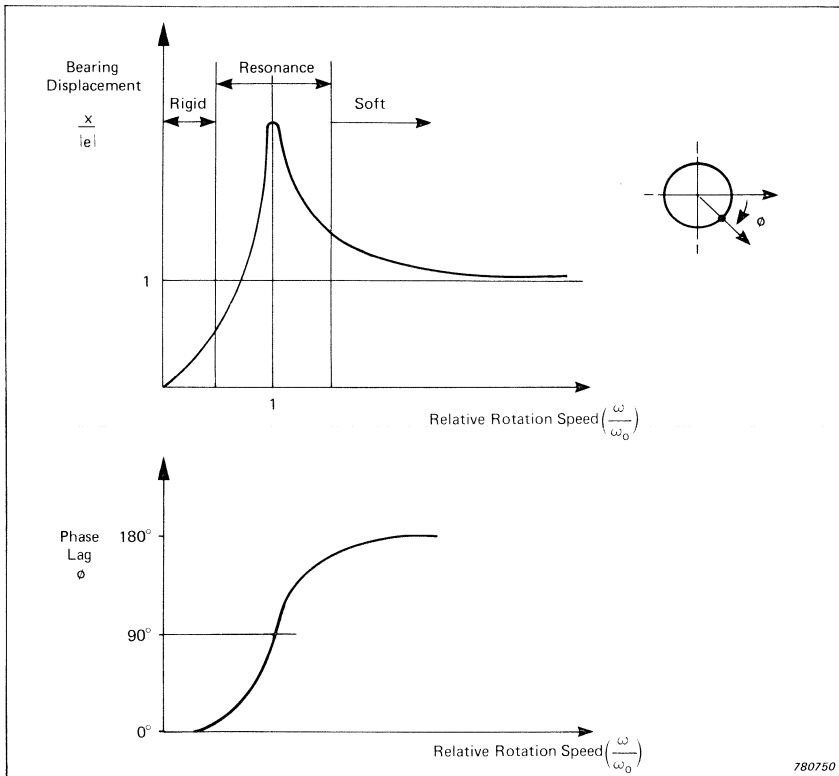


Fig.2. Operating regions of balancing machines

The measuring system

The type of transducer available, the type of rotor to be balanced, the work rate, and required residual unbalance are all factors which have influenced the development of modern soft- and hard-bearing machines.

The types of transducer used have been various. The first electronic machines of the 1930s used moving coil pick-ups, developed from loudspeakers, mounted on soft bearing support systems: these machines were themselves developments of existing balancing equipment. The large displacements could be measured relatively easily by connecting the pick-ups to modified wireless amplifiers. The development of the soft-bearing machine has been a logical one from the first patent registered by Henry Martinson in Canada in 1870.

There remained the problem of devising support systems which were sufficiently strong to support large rotors, yet sufficiently flexible to cope with the large amplitude excursions when balancing large rotors: these problems are particularly severe in cases of high initial unbalance and low rotation speed. Design focussed on hard-bearing construction, though the velocity pick-up remained widely used: a lever ratio was used to give mechanical amplification to the linear elastic motion of the journal bearing support members. To ensure that the motion of the rotor was restrained, and the unbalance force measured, a stronger, heavier machine evolved.

The high levels of precision required in industry, currently, taken with the increased availability of refined transducers and low-noise electronic circuitry, has fostered successive generations of balancing machine, of both types, which offer exceptional ruggedness and reliability in the industrial environment.

In the modern soft-bearing machine, the use of modern design techniques and lightweight materials enables the parasitic mass of the moving elements to be minimised for high sensitivity. Such a machine is shown in Fig.3. In this machine piezo-electric sensors measure absolute rather than relative motion, and are decoupled from external vibration effects through the support system. These elements have a wide dynamic range, have no moving parts, and are stable over a wide range of operating conditions. A secondary effect arising from the decoupling of the rotor system is that the baseplate and support posts can be of relatively lightweight construction, resulting in a transportable machine offering high sensitivity over its mass range.

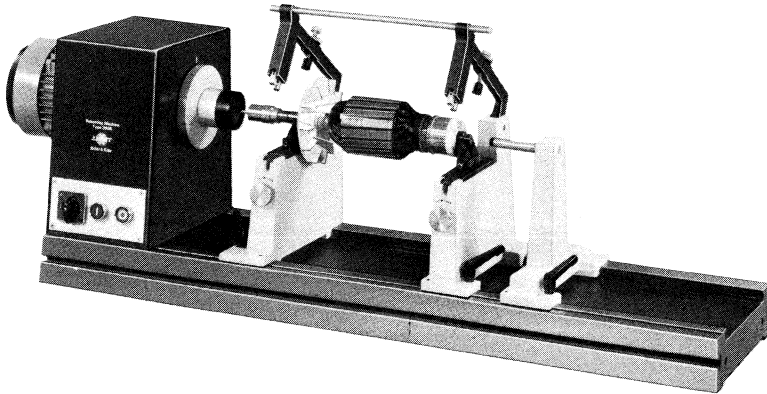


Fig.3. Brüel & Kjær direct drive soft-bearing production Balancing Machine Type 3905 — capacity 10 kg

For ease of construction, the support system allows for essentially unidirectional motion, usually in the horizontal, whilst being relatively stiff in the orthogonal direction. Special conditions for fine balancing, of gyros for example, may require an isotropic suspension to be used: this type of rotor is balanced in its own bearings, and unnecessary loading of the bearings by the use of a hard support systems would affect the delicate balance of the rotor system.

Correspondingly, modern hard bearing machines use sensors which measure force at the bearing directly, or else the linear elastic motion of the support members. Piezo-electric elements or electrodynamic devices are commonly used to produce the output proportional to force. To restrain the rotor adequately a stronger, heavier form of construction is necessary for the baseplate and support members, thus offering the high mechanical impedance required at the bearings. To minimise the transmission of extraneous effects through the base bed to the sensors, a concrete foundation block is frequently included in the installation as a seismic mass.

Drive methods

Belt drive is often used for small rotors, because individual drive connections are not required, rotors are changed quickly, and a high work rate achieved. The drive motor is typically mounted below the rotor with some form of overslung or tangential belt-drive arrangement to turn the rotor. Reference positions for the unbalance are obtained rela-

tive to marks on the rotor itself, using a stroboscope and *angle datum marks*, or by using a tachometric pulse derived from a suitable probe unit. The penalty of using this system is that positive control of the rotor, and absolute fixing of the unbalance angle, is forfeited; a qualified decision is required by the operator to determine the correct location for unbalance correction.

Direct drive through a connecting shaft and some form of universal coupling, with the component elements arranged axially, is also a common configuration. It is well-suited to the balancing of heavy rotors which can then be run up to speed under the positive control of the motor to minimise overall measuring time. For all rotor types this system allows direct recognition of the angle for unbalance correction and lends itself to automatic control with minimum supervision. In modern machines, careful attention to the design of the drive/coupling assembly ensures that machining tolerances and component misalignment do not restrict the capability of the machine, whilst allowing rapid change-over of rotors in a production series. Axial symmetry of screws and clamps on the coupling is important: by indexing the rotor by 90° or 180° , depending on the coupling design, it is possible to check that it is the rotor itself, and not the coupling elements whose unbalance has been compensated. These drive types are shown in Fig.4.

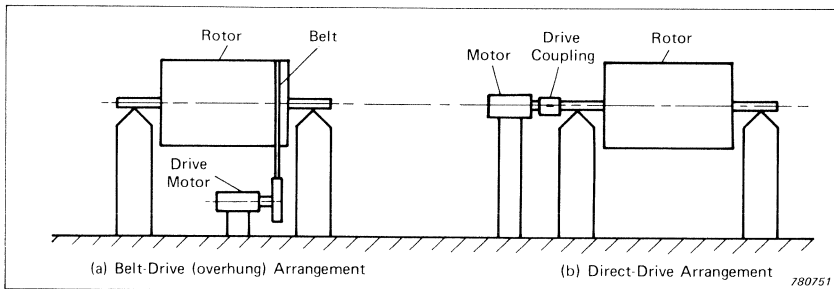


Fig.4. Drive configurations

For special applications, it is sometimes preferable to use other methods. Gyros which may require to be balanced successively at speeds up to their service speed — typically 20 000 — 24 000 rpm — to achieve the precision necessary (e. g. G 0,4 as recommended in ISO-1940), can be run up using their own drive. Electric or air-driven types are common. Similarly certain types of fans and turbocharger rotors are best

driven by air to simulate the service condition more closely. This criterion becomes increasingly important for balancing to fine quality grades better than G1, due to the increasing significance of "quasi-unbalance" effects transmitted through the bearings which must also be reduced.

Instrumentation console

Whichever type of transducer is used to measure unbalance, the task of analysis and display remains. The balancing machine console may be considered as a "dedicated" computer which processes the raw unbalance signals to yield calibrated correction values. The balancing bench and the console unit operate as complementary items of equipment; used correctly an *unbalance reduction ratio* of 80 — 90% should be achieved regularly in a single run.

In practice, the signals from the sensors do not have the simple sinusoidal form arising from cyclic effects of unbalance at the fundamental frequency, but also contain spurious signals from extraneous vibration, eccentricity of journals, air turbulence, etc. The measuring system must suppress higher order effects and noise to allow the unbalance to be analysed. Frequency discrimination is an important feature, and is effected using electronic filters or multiplier devices. The finer the balance quality required, the better the rejection characteristics of the filter must be. The resulting unbalance signal is processed through a network of sensitivity- and mixing potentiometers so that appropriate calibrated correction values for the correction planes to be used, appear on the front panel. These corrections should be selectable by a switch as to mass addition, or mass removal, to suit individual applications.

In the production situation, simultaneous display of the unbalance corrections for the two planes is desirable; the use of a selector switch necessitates note taking as an intermediate step. The displays should also incorporate a memory facility so that they are maintained when the machine is stopped. Various forms of *unbalance indicator* are in use: these include all analogue displays, digital/analogue displays and all digital displays, as well as vector displays which give a direct polar representation of the unbalance. Analogue displays always involve some interpretation by the operator, and have meter movements which are relatively vulnerable and sensitive to environmental vibration. Digital displays offer the potential for unambiguous interpretation of readings, periodic update of display to verify consistent operation, and high levels of repeatability and stability through the use of digital electronics.

In more specialized applications where components of unbalance have to be separated into vertical and horizontal components, or where a pictorial display is an advantage, the vectormeter display is useful; again the precision of the readings is limited by the divisions on the display graticule.

Calibration and operation

Both soft- and hard-bearing machines must be calibrated to suit different rotor types. The procedure is one of *plane separation* or *setting*. The front panel controls are adjusted to enable the unbalance measurements made at each bearing to be related directly to a mass value required on an associated correction plane, without this mass affecting the display relating to the second correction plane. Basically, the controls are adjusted to mix the signal from one sensor, with part of the signal from the other, to derive a signal which is referred to its correction plane rather than to its measuring plane. In this way the operator can obtain independent readings for each correction plane, and the confusing *correction plane interference* or cross-effects, as shown in Fig.5, are minimised. The details of calibration differ for the two types, and reflect their fundamentally different operating principles.

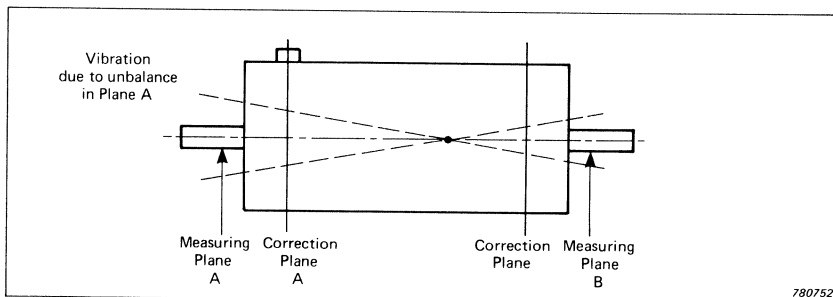


Fig.5. Interaction between planes for unbalance on Plane A

In the soft-bearing machine, the measuring system senses motion at the bearing journal as developed in (11). The machine must be calibrated dynamically for each rotor type as the motion detected in the measuring plane is consequent upon the location of the centre of mass, the moment of inertia, and the mass of the rotor, as well as on its geometric configuration. The calibration procedure is carried out on a prototype rotor, usually the first in a production series; it consists of running

the rotor up to speed to investigate the effect of adding a trial mass to each of the two correction planes to be used, in turn, at the radius to be used for correction. Thus:

1. The interaction between mass addition (or removal) on one correction plane and the display referred to the opposite correction plane can be minimised. *Plane separation* controls are used.
2. The displays for each correction plane can be adjusted to read directly in *practical correction units*, or convenient units for the production workshop. Typical units might be mass of a correction weight (g), length of a standard section of steel stock (mm), depth of a bore hole (mm), number of rivets, etc. These adjustments are made using calibration potentiometers.

Calibration is greatly accelerated when the console unit is equipped with *compensators*, as a balanced rotor is not required as the prototype. Compensator controls enable rotor balance to be simulated electronically before calibration: the compensators are switched out when calibrated measurements are to be made.

With the panel controls set, unbalance corrections can be found for any subsequent rotor in the series in a single *balancing run*. Where the calibration potentiometers are fitted with counters, the settings can be filed, and recovered for future balancing on rotors of the same type.

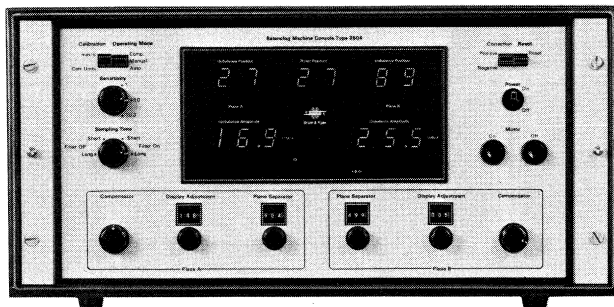


Fig. 6. Brüel & Kjær Balancing Machine Console Type 2504 with two channel all digital display

In fact, the requirement for preliminary calibration runs is little penalty in series production situations. The runs are performed just once, and thereafter correction values are displayed in terms of any locally convenient units to be used in the production workshop. The calibration achieved is unique for a given rotor configuration, and has been made at the level of unbalance anticipated in production. Alternatively, the unbalance might be displayed in values of vibration velocity for direct comparison with unbalance quality grades recommended in ISO-1940 or VDI-2060. A console with these features is illustrated in Fig.6.

The same console unit, when connected to appropriate vibration transducers, is equally suitable for equipping in-house constructed balancing machines. The operating principle requires only that bearing response is linear over the range of unbalance experienced by the machine, and that the characteristics of the two supports is similar. Where these characteristics do not pertain, the effect of damping will be to introduce a complex response coefficient between support bearings and correction planes such that the *correction plane interference* circuits can no longer adequately eliminate cross-effects.

For a hard-bearing machine, the measuring system senses force at the bearing journals as developed in (8). In this configuration the rotor system is statically determinate, and the usual rules of statics apply. By measuring the forces at the bearings, these can be referred to any other planes by geometric calculation. Thus the machine is *permanently calibrated* within its capacity and speed range, but the panel controls must be pre-set according to the geometry of the rotor, and the relative location of measuring and correction planes. The dimensions to be set might include:

1. Distance from left measuring plane to left correction plane.
2. Distance between measuring planes.
3. Distance between correction planes.
4. Left and right plane correction radii.

The unbalance corrections are obtained on the first run; the accuracy achieved now depends on the accuracy with which the dimensions have been measured and the controls set. This procedure is particularly convenient in one-off balancing situations, or where balancing small numbers of large rotors at low speeds: the possibility of introducing unexpectedly high unbalance effects during a trial run is also eliminated. However, for series production there remain advantages in calibrating in a similar mode to the soft-bearing machine to ensure precision re-

sults in a particular plant installation. The hard bearing system is generally preferred for balancing of heavy rotors and in cases where the calibration by geometric setting gives a significant time saving. This is particularly useful when many different rotors are balanced as one-offs or in small numbers and handling time becomes an important proportion in overall balancing time. This system is also suitable in cases where aerodynamic effects are significant (e.g. certain types of large fan) and for balancing supercritical shafts where the bearings might provide a better simulation of operating conditions.

Additional factors

The selection of balancing machine and display unit is clearly governed by many engineering factors relating to the intended application. Some of these are considered briefly here.

Where individual components are balanced on a mandrel (e.g. small fans and flywheels), prior to final assembly on a shaft, concentricity of diameters must be ensured, as well as adequate balance quality for the mandrel itself. Equally important is that a consistent approach is adopted to items such as locating keys and pegs: in US it is common to balance using a 1/2-key (flush with shaft) whilst in Europe a full-key standard has been widely adopted. If possible all set screws should be arranged with axial symmetry such that the possibility of introducing mass eccentricity of the assembled element is minimised. The importance of evaluating the cumulative effects of the maximum eccentricity of machining tolerances on an assembly of individual components can be seen from Table 2. This shows typical and obtainable tolerances for radial cast obtained by various machining operations. It is advisable, where practical, to balance at successive stages of assembly to prevent a build up of unbalance.

	Typical	Obtainable
Turning	250	30
Boring	60	30
Precision Drilling	40	20
Grinding	35	15

Table 2. Examples of typical and obtainable radial cast tolerances (μm) for various machining methods under ideal conditions

Fans and compressor elements often suffer greatly from aerodynamic effects influencing the indicated correction values. For cylindrical fans it is sometimes convenient to use a paper shroud around the circumference, or to tape a paper disc over the exposed intake, to reduce these effects: turbine discs may be spun up in the contra service direction to ensure the blades adopt a similar position as when in service. On larger elements a vacuum chamber may be necessary to achieve good quality grades.

Certain rotating elements may be subject to plastic deformation in the early stages of service, until suitable work-hardening has occurred. To avoid the necessity to strip down and to rebalance a second time, it may be convenient to overspeed the rotor on the balancing machine so that the component is work-hardened and may be balanced before being put in service.

Wherever temperature, magnetic, or reciprocating phenomena affect the unbalance in the assembled machine, every attempt should be made to ensure that either the unbalance tolerances achieved on the balancing machine are adequate, or that these physical conditions can be duplicated in the balancing workshop. Experience shows that due to the complex behaviour of rotor-bearing systems, empirical work is usually necessary to ensure that the balance quality achieved on the balancing machine is maintained with the support conditions experienced in service.

The production engineer must also consider the human factors relating to the acceptability of a balancing machine. Amongst the features which might be considered important are:

1. Ease of use after minimum instruction.
2. Clear, unambiguous display of unbalance.
3. Rapid interchangeability of rotors.
4. Security system to minimise the danger to the operator during normal operation and to prevent mechanical damage in cases of unsuspected high initial unbalance.

By consideration of these criteria it is most likely that the full advantages of improved balance quality of a product will be obtained. Guidance as to a formal evaluation of balancing machines is outlined in ISO-2953 "Balancing Machines — Description and Evaluation".

Portable instruments

Many manufacturers offer *field balancing* equipment of the type illustrated in Fig.7. This equipment is particularly useful for in-situ balancing of assembled machines at the commissioning stage, or for use as a maintenance tool: it can also find application in the production shop.

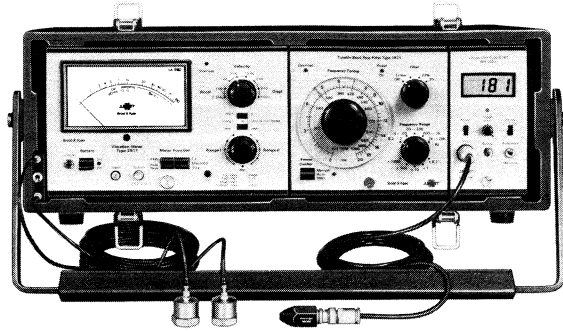


Fig.7. Brüel & Kjær Type 9500. Typical battery operated portable vibration measuring set for field balancing

The method of measuring unbalance is very similar to that described for the soft-bearing machine in Section "Calibration and Operation", as for retro-fit of measurement sensors, vibration transducers are the logical choice. The technique was first described by Thearle and Schectaly in their paper "Dynamic Balancing of Rotating Machinery in the Field" in 1956. (APM-56-19). The dynamics of the rotor system and its associated parasitic masses must be evaluated by introducing known "trial" unbalance in each correction plane, and observing the effects in the two measuring planes. This is necessary as the characteristics of the supports of an assembled machine are almost invariably unknown: in principle they could be anywhere between the "hard" and "soft" regimes shown in Fig.2 as well as being different end for end. In fact, the unfortunate case of resonance on a lightly damped support may make balancing impractical until the service speed or support characteristics are changed.

Measurements of vibration amplitude, and phase relative to a geometric trigger reference point on the rotor (using a stroboscope or phase meter) for two measuring points, are taken. Two test or trial runs with a known unbalance placed in turn in each of the selected correction

planes enable the characteristics to be established. Analysis of the resulting twelve values (six vector quantities comprising both amplitude and phase) is very readily performed by the current generation of programmable pocket calculators on the market, such as HP-67 and TI59. Where full portability is not necessary, any available computing facility might be used to enable the two correction values to be calculated.

This equipment tends to require more know-how on the part of the user than the balancing machines mentioned earlier, but its key feature is its versatility. Not only is it, with a pocket calculator, fully portable, but transducer sensitivities and analysis program can be selected appropriate to the given measuring task. It is also used for analysis of other potential sources of vibration, major culprits being misalignment at twice rotational speed, and loose elements or drive belts which may cause disturbance at various multiples of the fundamental.

Typical uses for this equipment are:

Extremely low *production rate* of large or specialized rotors which have to be run up in their own bearings and housings.

“Control” of rotors where balancing is only necessary infrequently.

Extremely large rotors, or where use of a conventional balancing machine would be impractical.

Assembled machines whose mode of operation due to air-, hydraulic- or magnetic influences could not be simulated adequately on a balancing machine.

Specialized investigations where these instruments, with their associated filters, calculators etc., offer features not available on a conventional balancing machine console.

Very wide range of maintenance balancing where corrosion, erosion, wear, and relative re-location of mating parts introduces unbalance in service.

Flexible rotors

A rotor is said to be “flexible” at operating speeds above 50% of its first natural bending mode. Whilst the balancing of a rigid rotor may be achieved by arbitrary selection of two axially separated correction planes, the unfortunate selection of these planes in the flexible condition can cause excitation of one or more natural bending modes. Stated simply, the rigid- or low-speed balancing operation does not allow for point by point axial correction of the unbalance. The forced vibrations associated with initial unbalance and lack of straightness are very important, and balancing machines are often used in this application.

For rotors with their mass concentrated at discrete axial positions (e. g. simple turbine assemblies), adequate balance quality can, in many cases, be achieved using low-speed techniques. In these cases the unbalance can be assumed to arise in these discrete axial locations, the discs, which are the same as used for placing of corrections. Suitable design of correction planes, one at each end and one at the centre, is increasingly common for turbine and electric armature assemblies, as careful selection of planes may avoid the necessity for more complex, time-consuming high-speed balancing.

Distributed mass rotors, such as cardan shafts, have to be treated rather differently. For a generalized rotor, whose flexural rigidity and mass per unit length are a function of their axial position, $EI(x)$ and $m(x)$ respectively, the deformation $y(x)$ at any rotation speed ω may be written:

$$[EI(x)y''']'' - \omega^2 m(x)y = \bar{F} \quad (12)$$

where \bar{F} is the sum of forces due to unbalance, and the prime designation' represents the differential operator d/dx . For unbalance having a "continuous" axial distribution $u(x)$, mass eccentricity about the axis of rotation, its form may be expressed as:

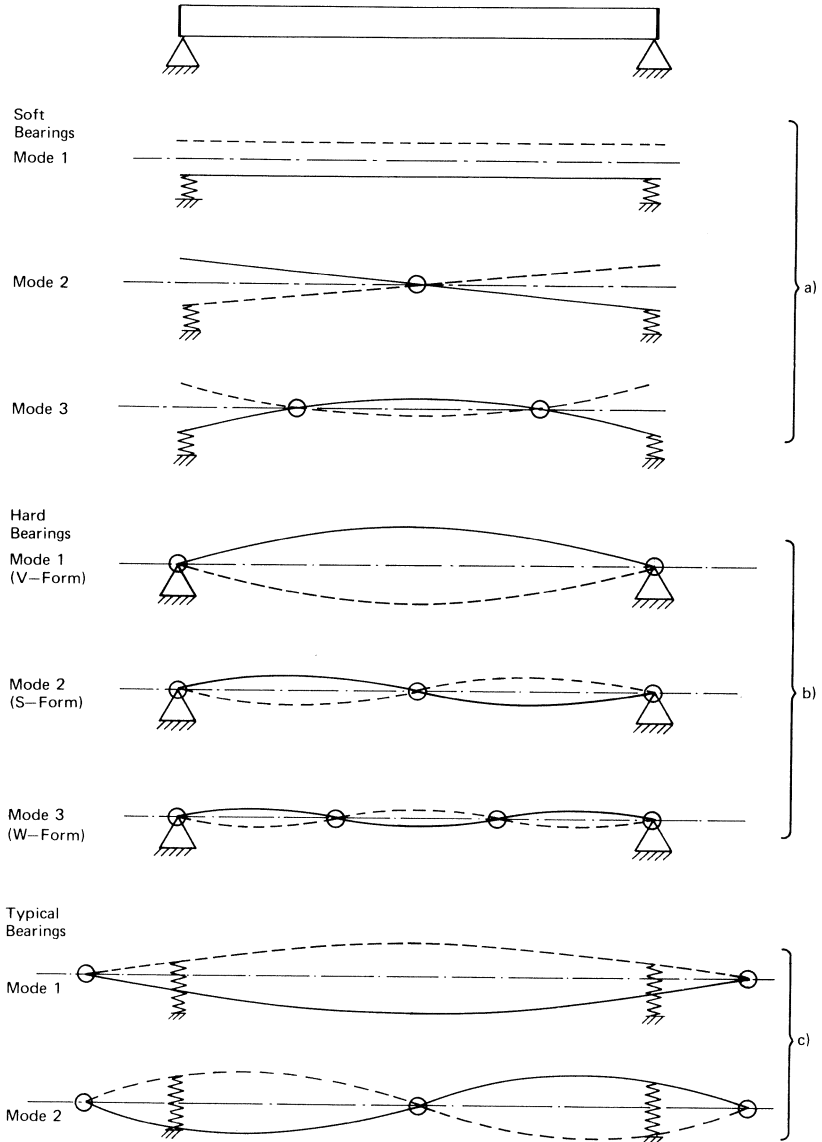
$$u(x) = \sum_n c_n m(x) \phi_n(x) \quad n = 1,2,3 \dots \quad (13)$$

where c_n is a constant and $\phi_n(x)$ is a solution, for the principal mode n , of differential equation (12) with $\bar{F} = 0$. By combination of (12) and (13) and using the principle of orthogonality, principal modes can be predicted to occur at critical frequencies ω_n , according to:

$$\omega_n = \int m(x) \phi_n^2(x) dx \quad (14)$$

Knowledge about the principal modes of the rotor, either calculated or estimated, enables optimal correction planes to be selected. So called "high-speed" balancing techniques may then be used to perform the "dynamic straightening" operation. Each mode may be balanced close to its critical speed, ω_n , balancing successive modes until the service speed lies at least 50% below the subsequent critical speed. At each stage correction weights are applied in sets so that correction of one mode does not excite lower modes already balanced.

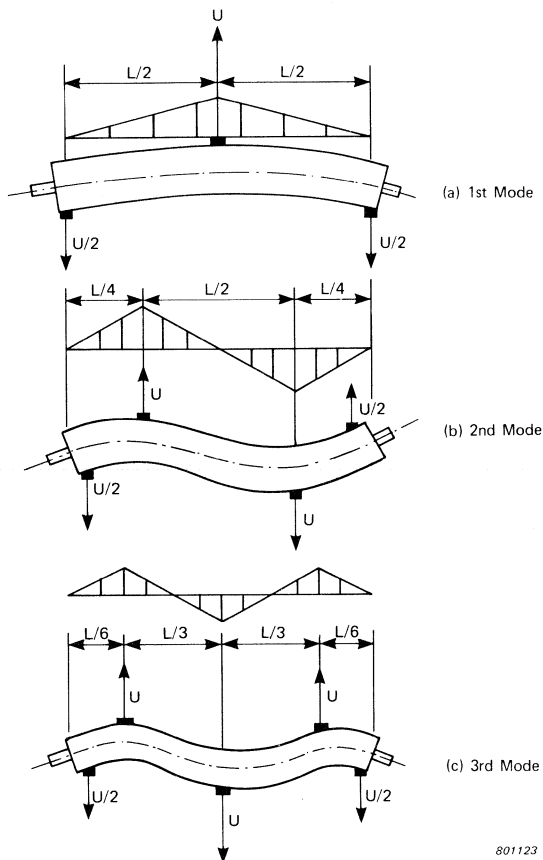
Rotor Mode Shapes



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Fig. 8. Rotor Mode Shapes

The actual critical speeds will depend on the support conditions. The first three principal modes for free-free, and rigid-rigid support systems are shown in Figs.8a and 8b respectively, whilst the effects of using supports of interim stiffness is shown in Fig.8c. It will be noted that since many soft-bearing balancing machines may be assumed rigid in the direction orthogonal to the "soft" direction, two sets of critical frequencies arise: it is sometimes convenient to achieve the rigid-rigid modes and to use a displacement transducer to measure the amplitude excursion at the associated anti-node(s). In the case of both hard- and



801123

Fig.9. Exciting first three flexible modes of uniform rotor. (Sets chosen not to influence static or couple unbalance). Bending moment diagram and shape

soft- support systems the signal occurring at the measuring supports is not a simple function of unbalance force, but also of the flexural rigidity and mass distribution of the rotor: thus both types require a test run using a known trial mass set to enable subsequent estimation of the corrections. This correction may be shown quite simply for the first three, V-, S-, W-modes of a uniform cylindrical shaft in Figs.9 (a), (b), (c): it may be noted that these mass sets do not affect the static- or couple-unbalance condition of the shaft. For each subsequent bending mode one extra correction plane is required. This technique, described as modal balancing, may not necessarily require rigid-rotor balancing at all: however, where the flexibility in service arises from characteristics of the foundation rigid-rotor balancing followed by high-speed balancing, or vice-versa, may be preferable. [Refs. 1, 2].

The modal balancing approach assumes that the individual modes are excited uniquely, whilst in practice, the unbalance is not distributed, but will comprise discrete local unbalance, U_k such that:

$$\bar{F} = \omega^2 \bar{u}(x) + \omega^2 \sum_{k=1}^{\infty} U_k \quad (15)$$

The physical consequence is that a single unbalance U_k can produce all principal modes $k = 1, 2 \dots \infty$, such that balancing at critical speeds up to rotational speed may not result in acceptable balance quality [Ref.3]. Much current investigation has centred on the "Influence Coefficient" method because these limitations are avoided. The rotor system is evaluated in terms of its sensitivity to unbalance corrections made in the correction planes. Influence coefficients can be calculated from measurements made by using a "trial unbalance" to systematically unbalance the rotor in each plane, and measuring the response in the measuring planes. By measuring on many planes (e. g. using displacement transducers) and at many speeds "least squares" techniques can be used to derive accurate coefficients for a rotor. The set of linear complex coefficients α_{jp} relating deflection j to unbalance at point p may be assembled in a matrix array $[\alpha]$ whose dimensions will be $m \times n$ where:

- m = no. of measuring stations
- n = no. of balancing planes

Thus for deflections and unbalance at discrete axial positions:

$$\begin{Bmatrix} y_1 \\ y_2 \\ \vdots \\ y_m \end{Bmatrix} = \begin{bmatrix} \alpha_{11} & \alpha_{12} & \cdots & \alpha_{1n} \\ \alpha_{21} & & & \vdots \\ \vdots & & & \vdots \\ \alpha_{m1} & & & \alpha_{mn} \end{bmatrix} \begin{Bmatrix} U_1 \\ U_2 \\ \vdots \\ U_n \end{Bmatrix} \quad (16)$$

Correction values are found by inversion of $[\alpha]$:

$$(U)_{\text{corr}} = - [\alpha]^{-1} (y) \quad (17)$$

To enable matrix inversion $[\alpha]$ must be square. This is achieved for $m = n$, or by balancing at different balancing speeds, [Ref.4]. The major feature of such an approach is that it is applicable to both balancing of multispan rotors in-situ and to production balancing, provided that satisfactory linearity of the bearings and foundation is observed. Influence coefficients generated in such a way may be considered to comprise separate matrices due to the rotor, $[\alpha_R]$, and the foundation, $[\alpha_F]$, such that:

$$[\alpha] = [\alpha_R] + [\alpha_F] \quad (18)$$

The use of this technique requires the use of modern computing facilities, but has been used to balance turbogenerators with three support bearings, and for production balancing applications.

Conclusions

Unbalance exists, to a lesser or a greater extent, in all rotating elements. The tendency towards higher service speeds and lighter overall physical construction has made the task of reducing the unbalance of such rotors to a residual unbalance level within a specified tolerance ever more important. Unbalance can arise at all stages of product life — component forming, assembly, operation, or repair.

An understanding of rigid rotor balancing is fundamental to the concept of the balancing machine. Specific unbalance, or centre of mass eccentricity ideas allow ready comparison of balance quality specifications, measuring sensitivities, and consequences of manufacturing or assembly tolerances. The measurement of unbalance allows correction by mass removal (drilling, grinding, etc.) or by mass addition (rivets, washers, etc.) as convenient for the rotor type.

The trend in high-volume balancing is following the pattern of current machine tool technology, with increasing use of fully automated balancing systems dedicated to a specific task. In contrast, the series- or one-off balancing installation has prime requirements for versatility and ease of use: work-rate in itself is not the overriding factor.

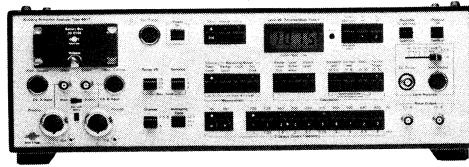
In recent years it has been found necessary to consider the interaction between the rotor and its bearings and foundation used in service. Where balancing on a balancing machine does not provide adequate unbalance quality following final assembly, portable balancing equipment is often used for the "trim"-balancing task. As an extension of this work computer techniques are currently under investigation for balancing of flexible rotors, and for flexible rotors with multiple support systems.

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News from the Factory

Building Acoustics Analyzer Type 4417



The Building Acoustics Analyzer Type 4417 is designed for the automatic measurement and subsequent calculation of the common quantities of interest in the sphere of building acoustics, and for precision sound power measurements. These quantities (e. g. reverberation time, impact and airborne insulation, sound power, normalised and standardised levels) are determined in 20 third octave bands covering the frequency range 100 Hz to 8 kHz (centre frequency). The results can be presented digitally on the Type 4417's display, graphically via a level recorder, printed via an alphanumeric printer or stored on a digital cassette recorder.

The Type 4417 is a portable (weight 7 kg), battery-powered instrument suitable both for on site measurements and for permanent laboratory arrangements. All the controls of the Type 4417, access to the battery box and all inputs and outputs (except the digital output) are on the front panel. This makes the Type 4417 eminently suitable for portable use and for installing in the practical Carrying Case KA 2000. The Type 4417 contains a random noise generator and is capable of producing third octave band limited random noise so that in conjunction with a microphone and a loudspeaker system (or other sound source or even a vibration exciter) the Type 4417 constitutes a complete measurement and computational system. List of the measurement and calculation capabilities of the Type 4417 is given in the Table. When both the two inputs (channel A and Channel B) and the two outputs (Channel A and Channel B) of the Type 4417 are employed, a completely automatic system for measuring insulation and reverberation times is obtained. The

Type 4417 will retain in its memory measurement data for 3 level-spectra and 1 reverberation-time spectrum. From these data and the entered values of the room's volume and the room's (or wall's) surface area, the Type 4417 will calculate any or all of 9 important spectra (see Table).

Calculation	Equation
Background Noise Level Correction	L_2 (and calculations using L_2) corrected for Background Noise Level L_{2B}
Level Difference	$D = L_1 - L_2$
–Standardised	$D_{nT} = D + 10 \log_{10} \left(\frac{T}{0,5} \right)$ (ISO 140)
–Normalised	$R = D + 10 \log_{10} \left(\frac{6,15 S T}{V} \right)$ (ISO 140)
–Index	$I_a =$ Insulation Index (ISO R 717)
Impact Level	$L_i = L_2$
–Standardised	$L_{iT} = L_i - 10 \log_{10} \left(\frac{T}{0,5} \right)$ (ISO 140)
–Normalised	$L_{in} = L_i - 10 \log_{10} \left(\frac{61,5 T}{V} \right)$ (ISO 140)
–Index	$I_i =$ Impact Index (ISO R 717)
Sound Power	$L_w = L_2 + 10 \log_{10} \left(\frac{V + \frac{43 S}{f}}{T} \right) - 14$ (ISO 3741 & 3742 Method 1)

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Miniature Accelerometers Types 4374 and 4375



These accelerometers are well suited for measurements on lightweight structures where relatively high level, high frequency vibrations are commonly encountered, and where the use of heavier transducers would alter the mode of vibration, invalidating measurements. Typical application areas are measurements on thin vibrating panels including

aircraft and automobile bodies, model testing, work in confined spaces and measurement of moderately high level shock.

Type 4374 is of planar shear construction, weighs approximately 0,65 grams and is suitable for measurement at frequencies up to 27 kHz, whilst Type 4375 is of delta shear construction, weighs approximately 2 grams and can be used for measurements of frequencies up to 18 kHz. Both accelerometers have an integral 40 cm long connection cable with miniature coaxial plug attached — Type 4374 having a plane base for wax or cement mounting and Type 4375 having an M3 screw thread for stud mounting.