

Rotor & Coupling Balancing



ISO 1940-1 methodology • coupling-specific engineering • workshop verification

- **40-slide technical course**

Rigid-rotor balancing principles, quality grades, tolerance allocation, coupling assemblies, and acceptance.

- **Engineering perspective**

Examples for elastomeric, gear, disc-pack, spacer, highly flexible, and fluid couplings.



Learning objectives

What participants will be able to calculate and specify

- Explain static, couple and dynamic unbalance in rigid rotors.
- Calculate permissible residual unbalance from grade G, mass and maximum service speed.
- Select appropriate balance planes, key convention, verification and documentation.
- Translate rotor criteria into coupling procurement and assembly practice.

Source hierarchy and scope

Interpretation must respect document applicability

- **ISO 1940-1:2003**

Rigid-state rotors: balance tolerances, plane allocation and verification. Not a flexible-rotor balancing procedure.

- **IRD Balancing technical paper**

Practical tutorial and allocation examples; secondary guidance, not a substitute for the standard.

- **FLENDER FLE 10 introduction**

Coupling families, balancing conventions, proprietary balancing options and application guidance.

The drive-train context

Motor → coupling → gearbox → driven machine

- Couplings transmit torque and may accommodate axial, radial and angular misalignment.
- Balancing concerns rotating mass distribution; alignment concerns shaft geometry.
- Misalignment, looseness and resonance may remain after perfect balance.



Source-informed training; illustrative calculations are not a substitute for OEM / project acceptance criteria.

Coupling families

Different constructions create different balance risks

- Torsionally rigid: gear, all-steel membrane and disc-pack families.
- Torsionally flexible: pin/bush and elastomeric jaw designs.
- Highly flexible: large-volume elastomeric ring, tire or disc elements.
- Hydrodynamic: fluid-filled assemblies requiring special manufacturer procedures.



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What is rotor unbalance?

A vector quantity, not merely a vibration reading

- **Definition**

Unbalance magnitude $U = m \times r$, commonly expressed in g·mm.

- **Phase**

Angular position of the equivalent unbalance vector relative to a marked reference.

- **Physical consequence**

Rotating centrifugal force scales approximately with $U \times \text{angular speed squared}$.

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Static unbalance

Resultant unbalance without a resultant couple

- The mass center is offset from the shaft rotation axis.
- For narrow rotor geometries, a single correction plane may be sufficient.
- A static correction does not necessarily eliminate a couple component in a long assembly.

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Couple unbalance

Equal and opposite vectors separated axially

- Two equal unbalance vectors 180° apart produce a rocking moment.
- Static balancing cannot detect a pure couple unbalance.
- Two-plane balancing is normally required to address both resultant and couple components.

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Dynamic unbalance

General rigid-rotor condition

- Combination of resultant and couple unbalance.
- Can be represented by two equivalent unbalance vectors in selected planes.
- The same physical state may have different numerical plane vectors depending on plane positions.

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Rigid versus flexible rotor state

Select the correct balancing framework

- ISO 1940-1:2003 applies to rotors behaving in a constant (rigid) state.
- Flexible-state rotor behavior requires a different engineering approach; ISO 11342 is referenced in the supplied standard.
- A long high-speed coupling spacer may require rotor-dynamic assessment before using rigid-state assumptions.

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Balance quality grade G

Meaning of the grade number

- **$G = e_{per} \times \omega$**

G is expressed in mm/s; e_{per} is permissible specific unbalance in mm; ω is angular speed in rad/s.

- **Interpretation**

At fixed G, permissible eccentricity decreases as speed increases.

- **Important**

A G grade alone does not define acceptable installed vibration.

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Primary calculation

From service speed to residual unbalance

- **$\omega = 2\pi n / 60$**
n in revolutions per minute; ω in radians per second.
- **$e_{per} [\mu m] \approx 9549 \times G / n$**
Numerical conversion for G in mm/s and n in rpm.
- **$U_{per} [g \cdot mm] \approx 9549 \times G \times M / n$**
M is rotor mass in kg. State the balanced assembly mass explicitly.

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Illustrative quality grades

Examples from the supplied IRD guidance

- G 16: some drive shafts and machinery components under representative conditions.
- G 6.3: common process-plant machinery, fans and pump impellers.
- G 2.5: turbines, turbo-compressors and selected high-quality rotors.
- These are illustrative groups, not automatic coupling acceptance requirements.

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Grade selection for couplings

Duty, speed and OEM rules come first

- FLENDER introduction notes G 16 may suffice for many couplings; vibration-sensitive duties may use G 6.3.
- More stringent grades require engineering justification and appropriate assembly control.
- Specify grade AND speed, mass boundary, key convention, and balancing planes.

FLENDER STANDARD COUPLINGS | INTRODUCTION

Balancing

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Balancing standard in accordance with DIN ISO 21940-32
Besides the required balance quality, it is necessary to set standards which define how the mass of the parallel key is to be taken into consideration when balancing. In the past, motor rotors have frequently been balanced in accordance with the full parallel key standard. The "appropriate" balance condition of the coupling hub was described as "balancing with open keyway" or "balancing after keyseating". Today it is usual for the motor rotor, as well as the gear unit and driven machine shaft, to be balanced in accordance with the half parallel key standard.

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$$e_{perm} = 9550 \cdot \frac{G}{n}$$
$$e_{coupl} \leq e_{perm}$$

Permitted eccentricity of center of gravity e_{perm} in μm
Eccentricity of center of gravity of coupling e_{coupl} in μm
Balancing quality level G in mm/s
Coupling speed n in rpm

Marking of shaft and hub with "F" (for "full").

The balanced parts must be marked with an "H". This marking can be dispensed with if it is absolutely clear which parallel key standard has been applied.

To prevent imbalance forces caused by projecting parallel key factors when balancing in accordance with the half parallel key standard in the case of applications with high balancing quality requirements, grooved spacer rings can be fitted or stepped parallel keys used.

Eccentricity of center of gravity of coupling e_{coupl}	Flender balancing quality	Order code
maximum 100 μm	standard balancing	without specification
maximum 40 μm	fine balancing	W02
maximum 16 μm	micro-balancing	W03
better than 16 μm	special balancing	on request

FLENDER E/11

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Speed sensitivity

Why maximum service speed matters

- At constant G , U_{per} is inversely proportional to speed.
- At constant residual unbalance, centrifugal force rises with speed squared.
- Verify continuous maximum speed, transient overspeed and the manufacturer's approved balance speed.

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Worked example 1

G 6.3 • 25 kg • 1,500 rpm

- **Specific residual unbalance**

$$e_{\text{per}} = 9549 \times 6.3 / 1500 = 40.1 \text{ } \mu\text{m}.$$

- **Total permissible unbalance**

$$U_{\text{per}} = 40.1 \times 25 = 1,003 \text{ g}\cdot\text{mm (approximately).}$$

- **Two-plane illustration**

For symmetric allocation: about 501 g·mm per tolerance plane, subject to geometry and allocation rules.

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Worked example 2

G 2.5 • 12 kg • 3,600 rpm

- **Specific residual unbalance**

$$e_{\text{per}} = 9549 \times 2.5 / 3600 = 6.63 \text{ } \mu\text{m}.$$

- **Total permissible unbalance**

$$U_{\text{per}} = 6.63 \times 12 = 79.6 \text{ g}\cdot\text{mm}.$$

- **Correction mass at 100 mm**

If all 79.6 g·mm is corrected at radius 100 mm: approximately 0.80 g (illustrative).

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Correction mass and radius

Workshop conversion and safety

- Correction mass m_{cor} [g] = correction unbalance U_{cor} [g·mm] / radius r [mm].
- At radius 80 mm, 160 g·mm corresponds to 2.0 g correction mass.
- Use approved drilling, milling or addition locations; preserve component strength and fatigue margin.

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One-plane versus two-plane

A geometrical and dynamic decision

- Single-plane: narrow rotor or proven negligible couple component.
- Two-plane: wider rotor, substantial axial separation or appreciable moment unbalance.
- Select tolerance planes and correction planes separately; they are not always coincident.

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Tolerance planes

A key concept in ISO 1940-1:2003

- Reference planes are used to state allowable residual unbalance.
- Correction planes are the physical locations where mass is removed or added.
- Transform tolerances to correction planes with rotor geometry and vector relationships.

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Symmetrical rotor allocation

Balanced around the center of mass

■ Typical case

For symmetric geometry and suitable correction planes, allocate $U_{per}/2$ to each plane.

■ Assumptions

The center of mass is near the midpoint and plane locations satisfy the relevant geometry.

■ Caution

Do not automatically use 50/50 for overhung or asymmetrical coupling assemblies.

BALANCE QUALITY GRADES

Table 1 shows the balance quality grades for a variety of rotor types. The "G" number 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.

$$G = e \times \omega = \text{constant}$$

This is based on the fact that geometrically similar rotors running at the same speed will have similar stresses in the rotor and its bearings.

Balance quality grades are separated by a factor of 2.5. However, G numbers of intermediate value may be used to satisfy special requirements. For example, a standard pump impeller has a suggested balance quality grade of G 6.3. Special conditions may require a better balance quality of G 4.0 to satisfy installation in an area with low structure-borne noise limits.

DETERMINING PERMISSIBLE RESIDUAL UNBALANCE - U_{per}

$$U_{per} = e_{per} \times m$$

(m = rotor mass)

Permissible residual unbalance is a function of G number, rotor weight and maximum service speed of rotation. Instead of using the graph to look up the "specific unbalance" value for a given G number and service RPM and then multiplying by rotor weight (taking care to use proper units), U_{per} can be calculated by using one of the following formulae:

$$U_{per} \text{ (oz-in)} = 6.015 \times G \times W/N \quad (W \text{ in lb})$$

$$U_{per} \text{ (g-in)} = 170.5 \times G \times W/N \quad (W \text{ in lb})$$

$$U_{per} \text{ (g-mm)} = 9549 \times G \times W/N \quad (W \text{ in kg})$$

G = Balance quality grade from Table 1

W = Rotor weight

N = Maximum service RPM

A slide rule that calculates U_{per} is also available from some balancing machine manufacturers.

ALLOCATION OF U_{per} TO CORRECTION PLANES

U_{per} is the total permissible residual unbalance and must be allocated to the balancing correction planes used based on rotor dimensions and configuration.

For rotors balanced in a **single correction plane**, all of the U_{per} applies to that correction plane.

For rotors balanced in **two correction planes**, U_{per} must be allocated to each correction plane based on rotor configuration and dimensions.

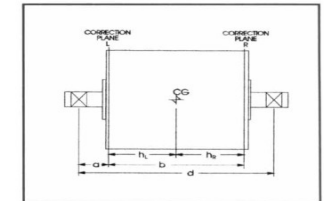


Figure 2 Symmetrical rotors

SYMMETRICAL ROTORS

Rules for symmetrical rotors. (See Figure 2.)

1. Correction planes are between bearings.
2. Distance "b" is greater than 1/3 "d."
3. Correction planes are equidistant from the center of gravity.

$$U_{per, \text{ left}} = U_{per, \text{ right}} = U_{per}/2$$

When correction planes are NOT equidistant from the center of gravity, then -

$$U_{per, \text{ left}} = U_{per} (b_r/b_l)$$
$$U_{per, \text{ right}} = U_{per} (b_l/b_r)$$

The $U_{per, \text{ left}}$ or $U_{per, \text{ right}}$ should not be less than 30% or more than 70% U_{per} . If they are, then use rules for narrow rotors.

Asymmetrical rotor allocation

Distance-weighted distribution

- For an idealized two-plane rotor with center of mass between planes, allocation may depend on distances to the center of mass.
- Example concept: $U_{\text{left}} = U_{\text{total}} \times h_{\text{right}} / b$; $U_{\text{right}} = U_{\text{total}} \times h_{\text{left}} / b$.
- Apply source geometry rules and verify limits for the actual rotor; this is not a universal shortcut.

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Overhung coupling arrangements

A higher sensitivity to couple unbalance

- Overhung mass produces larger bending moments at the bearings.
- Two-plane allocations and correction locations require extra attention.
- A short hub and long spacer cannot be assumed equivalent to a compact rotor.

ROTORS WITH OUTBOARD CORRECTION PLANES

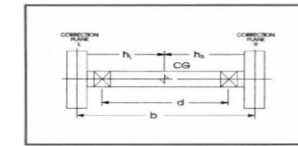


Figure 3 Rotor with outboard planes

Rules for rotors with correction planes outside the bearings. This is often referred to as a "dumb-bell" rotor configuration. (See Figure 3)

Both correction planes are outboard of the bearings.

$$b > d$$

Adjust U_{per} by ratio of d/b . (Reduces U_{per})

$$U_{per} = U_{per} (d/b) \quad U_{per} = \text{Adjusted value}$$

When correction planes are not equidistant from the center of gravity, calculate U_{per} left and right as follows:

$$U_{per, \text{left}} = U_{per} (h_1/b) \quad U_{per, \text{right}} = U_{per} (h_2/b)$$

OVERHUNG AND NARROW ROTORS

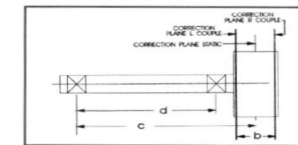


Figure 4 Overhung rotors

Rules for overhung and narrow rotors. (See Figures 4 and 5).

1. Distance between correction planes is less than 1/3 the distance between bearings. $b < 0.33 d$.
2. Assumes equal permissible dynamic bearing loads.

3. Couple corrections are made 180° apart in their respective planes.
4. The plane for static corrections may be a third plane or either of the planes used for couple corrections.
5. Allocate U_{per} as static and couple residual unbalance as follows:

$$U_{per, \text{static}} = U_{per}/2 \times d/2c$$

$$U_{per, \text{couple}} = U_{per}/2 \times 3d/4b$$

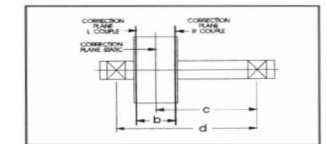


Figure 5 Narrow rotors

Permissible unbalance allocations for overhung and narrow rotors require that two plane unbalance corrections be divided into static and couple unbalance equivalents. This can be done graphically by plotting the two plane balance solution vectors U_1 and U_2 as shown in Figure 6. Connect vectors U_1 and U_2 as shown. The vector from the origin to the mid-point of vector C_1 - C_2 is one-half the rotor's static unbalance. Vectors C_1 and C_2 are the couple unbalance.

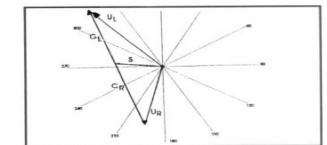


Figure 6 Static-couple graphical derivation

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Single-plane balancing workflow

Procedure for suitable compact hubs

- Identify datum, key convention and finished bore condition.
- Mount on a validated mandrel and measure initial magnitude/phase.
- Apply approved correction and repeat until residual is within the specified limit.
- Document mass, radius, phase, grade, speed and residual result.

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Two-plane balancing workflow

Procedure for rigid coupling subassemblies

- Identify plane A and plane B and record axial separation.
- Perform calibrated measurement or influence-coefficient trial corrections.
- Calculate vector corrections independently for both planes.
- Re-run and verify each plane against its allocated tolerance.

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Balancing-machine setup

Repeatability is essential

- Check machine calibration, drive method, tooling and mandrel concentricity.
- Minimize mounting errors, runout and contamination.
- Ensure speed and sensitivity suit the component and tolerance.
- Perform repeat runs and reversal/remount checks where appropriate.

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Residual unbalance verification

Acceptance must be measurable

- State permitted residual unbalance and plane basis.
- Measure residual vectors after the final correction.
- Consider uncertainty, tooling effects and machine error; ISO 1940-2 addresses balance errors.
- Record evidence, not only a pass/fail statement.

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Half-key balancing

Critical for keyed coupling hubs

- **H: half-key convention**

FLENDER identifies half-key balancing as the normally used practice for shaft and hub.

- **F: full-key convention**

The shaft is balanced with the full key; the hub treatment must be compatible.

- **N: no-key convention**

Do not mix conventions without a controlled mass correction and explicit agreement.

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Keyway mismatch case study

Why a balanced hub may create an unbalanced assembly

- Shaft balanced half-key; hub balanced full-key → inconsistent key-mass assumptions.
- Result can be an additional unbalance vector at the keyway radius.
- Specify key length, half-key mass, key orientation and marking for both mating parts.

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FLENDER balancing options

Manufacturer-specific categories, not generic ISO grades

- FLENDER excerpt: standard balancing up to 100 µm center-of-gravity eccentricity.
- Fine balancing: up to 40 µm; catalog code W02.
- Micro-balancing: up to 16 µm; catalog code W03.
- Special balancing: better than 16 µm by request; confirm series applicability.

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FLENDER example at 1,450 rpm

Cross-check grade against a proprietary option

- **Target G 6.3**

$$\text{eper} = 9549 \times 6.3 / 1450 = 41.5 \text{ } \mu\text{m}.$$

- **Fine-balancing option**

40 μm is below the calculated 41.5 μm limit; manufacturer cites W02 for this case.

- **Procurement note**

Always confirm component type, order code and latest OEM documentation.

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Elastomeric couplings

Jaw, pin/bush, tire and highly flexible designs

- Balance metallic hubs and any separately specified rotating components.
- Element mass variation, eccentricity and fit can alter assembled balance.
- Check whether elastic elements are installed during the balance test.
- Mark assembly orientation if match balancing is required.



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Gear couplings

ZAPEX-style gear coupling considerations

- Control gear hub, sleeve, flange and fastener mass distribution.
- Inspect lubricant distribution assumptions and seals where relevant.
- Verify assembly phasing, bolting and spacer configuration.
- Use OEM limits for speed, balance quality and assembly procedure.



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Disc-pack / membrane couplings

ARPEX and N-ARPEX style considerations

- Machined metallic components may offer good inherent balance, but that is not proof of final acceptance.
- Disc-pack hardware, spacer tubes and flange bolts affect rotating mass.
- Check whether OEM requires balance by subassembly or complete cartridge.
- Protect flexible elements from incorrect handling and imposed strain.



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Spacer couplings

Long DBSE and high-speed applications

- A long spacer raises sensitivity to eccentricity, assembly runout and potential flexible-rotor behavior.
- Evaluate critical speed and rotor dynamics when warranted.
- Record spacer orientation, fit, flange pilots and fastener grades.
- Specify complete-assembly balancing when required by the project or OEM.

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Fluid couplings

FLUDEX and liquid-filled assemblies

- FLENDER explicitly indicates special balancing standards for FLUDEX.
- Fluid fill, impeller/casing construction and sealing condition influence rotating behavior.
- Do not transfer dry rigid-coupling assumptions to a fluid coupling without OEM instructions.



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Assembly and installation effects

Factory balance is not the entire story

- Reassembly position, keys, bolts, pilots and spacer interchange may change residual balance.
- Shaft runout, bearing condition, alignment and soft foot can produce vibration independently.
- Use witness marks and controlled tightening sequence as specified by OEM.

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Technical RFQ and purchase specification

Minimum data required to avoid ambiguous balance orders

- Coupling type, manufacturer/series, configuration and mass boundary.
- Maximum service speed, balancing speed, grade G or permitted U per plane.
- Bore/key convention, tolerance/correction planes and assembly condition.
- Verification report, uncertainty, acceptance criteria and special overspeed requirements.

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Factory acceptance checklist

Practical inspection and documentation

- Confirm machine traceability and tooling identification.
- Check before/after residual vectors in g·mm and phase for each plane.
- Confirm key convention, assembly orientation, mass, speed and calculation.
- Review exceptions and any OEM-specific balancing class or code.

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Engineering summary & references

Specify • calculate • allocate • correct • verify

- **Core principle**

G alone is insufficient: document mass, speed, planes, key convention and assembly boundary.

- **Primary sources**

ISO 1940-1:2003 (uploaded); IRD Balancing, Balance Quality Requirements of Rigid Rotors; FLENDER FLE 10 Introduction.

- **Edition note**

ISO 1940-1:2003 is the supplied reference edition; check current contractual standard and applicable successor requirements before issuing a binding specification.

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THANK YOU FOR YOUR ATTENTION

We Welcome Your Questions, Comments & Feedback

MECHATRIC

Power Transmission & Motion Solutions