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Balancing of rotating tools and tool systems

Équilibrage pour outils rotatifs et systèmes d'outillage

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Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

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For an explanation on the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 29, *Small tools*.

Introduction

Increasing cutting speeds in combination with higher balancing requirements result in tighter balancing conditions for the tool spindle system (machine tool spindle, clamping device and tool system). Especially balancing tools and tool systems according to ISO 1940-1 are often being intensified by additionally choosing the next better balancing quality (e.g. G 2,5 instead of G 6,3). Not only that this is technically often not necessary and leading to high cost, it also cannot be achieved in many cases.

Unbalance acts as speed-harmonic excitation of the machine structure and the amount of the excited centrifugal force arises from the unbalance and the rotational speed. Another point of consideration in connection with this is the spindle load due to dynamic cutting forces (e.g. caused by the interrupted cut of a milling cutter) which are often remarkably higher than the centrifugal forces caused by demanded permissible residual unbalances.

The balancing quality requirements for rigid rotors stated in ISO 1940-1 (e.g. electromotor rotors, etc.) cannot be applied appropriately to these tool-spindle systems because machine tool spindles, clamping devices and tools show essentially different features:

— machine tool spindles, clamping devices and tools are varying systems (e.g. by tool changes in machining centres);

— due to radial and angular clamping inaccuracies, a repeated tool change within the spindle leads to varying balancing conditions for tool-spindle systems;

— fit tolerances of the individual components (spindle, clamping device and tool) set limits to the balancing process.

In particular, clamping inaccuracies between tool system and machine tool spindle set limits to the repeatability of the balancing conditions. This document, however, does not specify details for the balancing of tool-spindle systems that include the machine tool spindles.

In view of this, procedures have been developed to derive the balancing requirements of rotating tool systems taking all essential parameters into account. The main objective is the limitation of unbalance related machine vibrations and system loads, as well as process interferences.

The above circumstances were the reasons to develop a new approach to specify the requirements for the balancing of rotating tool systems. This document is based on research results gathered at the PTW "Institute of Production Management, Technology and Machine Tools of the Technical University Darmstadt", the GFE "Association for Manufacturing Technology and Development (Gesellschaft für Fertigungstechnologie und Entwicklung e. V.)" in Schmalkalden (Germany) and the discussions of the German standards working group "Requirements for Balancing of Rotating Tool Systems".

Research results and experiences in practice have shown that this document is suitable from both the technical and economical point of view.

Annex A shows several examples for static and dynamic balancing of differently shaped tools while modular tool systems are addressed by the examples of Annex B. Annex A also includes the derivations of the calculations of the dynamic permissible residual unbalances for the three different geometrical situations mentioned in this document.

An introduction to the subject "balancing" is also included in ISO 19499. This document includes useful information with regard to other standards dealing with the balancing of rotors.

EN 847 (all parts) contains additional specifications for the balancing tools for woodworking.

Balancing of rotating tools and tool systems

1 Scope

This document specifies requirements and provides calculations for the permissible static and dynamic residual unbalances of rotating single tools and tool systems. It is based on the guideline that unbalance related centrifugal forces induced by the rotational speed do not harm the spindle bearings, as well as prevent unbalance related impairments of machining processes, tool life and work piece quality.

NOTE 1 Tools and tool systems covered by this document are, for example, those with hollow taper interfaces (HSK) according to ISO 12164-1 and ISO 12164-2, modular taper interface with ball track system according to the ISO 26622 series polygonal taper interface according to the ISO 26623 series, taper 7/24 according to ISO 7388-1, ISO 7388-2, ISO 9270-1 and ISO 9270-2 related to their individual operating speed.

Modular tool systems are another important and complex issue of this document. Calculations and process descriptions for balancing these components and the assembled tool systems are included.

This document is putting an important focus on the possible clamping dislocations of tool shanks and their effects on the balancing procedure. These dislocations can occur between a tool or a tool system and the machine tool spindle (e.g. with every tool change), as well as within a modular tool system during its assembly.

NOTE 2 Unfavourable process or system conditions (e.g. partial resonances of the machine structure generated by particular rotational speeds) or design and machine-related technical conditions (e.g. the projecting length of the axes, narrow space conditions, vibration susceptible devices, clamping devices and tool design) may lead to increased vibration loads and balancing requirements. This is dependent on the individual interaction of the machine and the tool spindle system and cannot be covered by a standard. A deviation from the recommended limit values of this document can be required in individual cases.

NOTE 3 Wear of the shank interfaces may lead to possible variations of the clamping situation and thus to worse run-out and balancing conditions. These errors cannot be specifically addressed in a standard.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 1925, *Mechanical vibration — Balancing — Vocabulary*

3 Terms, definitions, symbols and abbreviated terms

3.1 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 1925 and the following apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

NOTE The specific field of balancing requires the introduction of terms and definitions which are not in accordance with ISO 13399.

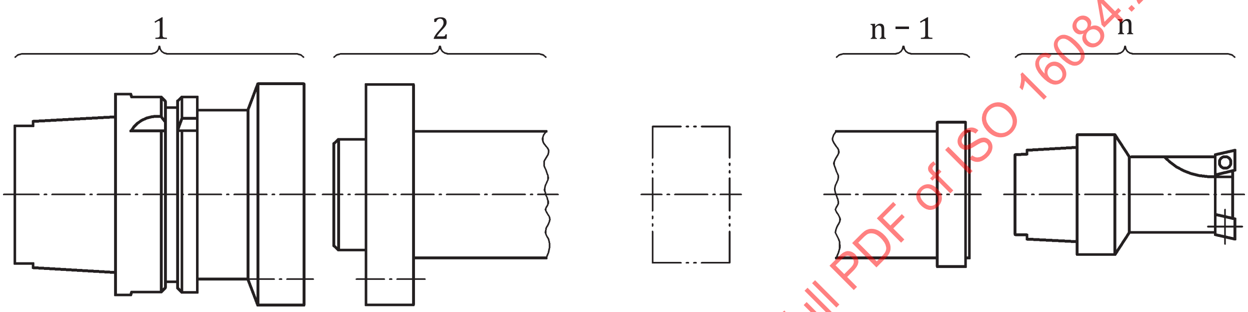
3.1.1
tool-spindle system
assembly of all components, i.e. machine tool spindle and *tool system* (3.1.2), which may cause unbalance due to design, shape, run-out, etc.

3.1.2
tool system
assembly of at least two components

EXAMPLE A shank adapter and a *single tool* (3.1.3).

Note 1 to entry: The term “modular tool system” is being synonymously used with “tool system” in this document.

Note 2 to entry: Component 1 (shank adapter) of Figure 1 could also be a tool that includes an interface to hold component 2.



Key

1 Component 1: Shank adapter

2 Component 2: Intermediate adapter

n − 1 Component n-1: Intermediate adapter

n Component n: (Single) cutting tool

Figure 1 — Example of possible components of a modular tool system

3.1.3
single tool
composition of the tool body, intermediate elements (e.g. cassettes, modular components) and the cutting edge(s) (e.g. cutting tip, bit) for removing material from a work piece through a shearing action at the defined cutting edge(s)

Note 1 to entry: The term “single tool” has the meaning “single cutting tool”.

3.1.4
basic adapter
adaptive item with different types and sizes of male or female connecting *interfaces* (3.1.7) on both the machine and workpiece side

3.1.5
intermediate adapter
adaptive item between a *basic adapter* (3.1.4) and a *single tool* (3.1.3) or another intermediate adapter

3.1.6
clamping device
device which constitutes the connection between machine tool spindle and *tool system* (3.1.2)

3.1.7
interface
contact point between the components of a *tool system* (3.1.2) and between a tool or a tool system and the machine tool spindle

3.1.8
unbalance moment
moment caused by an unbalance with an axial distance (i.e. a lever) to the front spindle bearing

3.1.9
couple unbalance
special kind of *unbalance moment* (3.1.8) caused by a pair of unbalance vectors of the same length, opposite direction and axial distance

Note 1 to entry: It mainly occurs due to quasi-static balancing (see Figure 5 and A.5.2).

3.2 Symbols and abbreviated terms

Symbols and abbreviated terms	Unit	Description
a	mm	Total lever arm — distance front bearing B1 to the tool centre of gravity CG
a_M	mm	Machine lever arm of generalized spindle model (i.e. distance from front bearing to spindle nose, e.g. HSK face)
B1	—	Spindle bearing 1
B2	—	Spindle bearing 2
b	mm	Distance between the balancing planes P1 and P2
b_{MIN}	mm	Minimal distance between the balancing planes P1 and P2
CG	—	Centre of gravity
CS	—	User (often also customer)
C_{DYN}	N	Dynamic load rating(s) of spindle bearing(s)
D	mm	Diameter
D_{REF}	mm	Reference diameter of a tool or a component for the G40 check
D_S	mm	Reference shank flange diameter (e.g. HSK-63 → D_S = 63 mm)
$e_{k,SYS,MAX}$	mm	Maximum radial dislocation of component k within a tool system
e_S	mm	Pure radial dislocation of a tool shank of a tool or a tool component
$e_{S,i}$	mm	Radial dislocation of the tool shank of component i
f_{BAL}	—	Weighting factor for the balancing quality
$f_{BAL,FINE}$	—	Weighting factor for fine balancing
$f_{BAL,STND}$	—	Weighting factor for standard balancing
$f_{P,MIN}$	—	Factor to prevent falling below a minimal permissible unbalance per plane
$f_{SYS,k}$	—	Factor to calculate the permissible component unbalances of special tool systems
$\vec{F}$	N	Force vector
F_B	N	Total force on a spindle bearing
F_{B1}	N	(Dynamic) Force on spindle bearing B1 due to an unbalance
F_{B2}	N	(Dynamic) Force on spindle bearing B2 due to an unbalance
$F_{B1,CPL}$	N	Force at bearing B1 due to a couple unbalance
$F_{B1,RES}$	N	Resultant force at bearing B1
$F_{B1,STAT}$	N	Force at bearing B1 due to a static unbalance
G (x)	mm/s	Balancing quality G (x) according ISO 1940-1, e.g. G 6,3
G40	mm/s	Safety limitation of the permissible unbalance according to ISO 15641
h_{P1}	mm	Distance from RP to plane P1
h_{P2}	mm	Distance from RP to plane P2
HSK-(x)	—	HSK of size (x) representing all different types (A, C, E, T, etc.), e.g. HSK-63

Symbols and abbreviated terms	Unit	Description
i	—	Index for serially numbered parameters or components (balancing planes, tool components, etc.)
k	—	Number of tool system components
k_{SYS}	—	Total number of tool system components
$k_{SYS,STND}$	—	Total number of components of a standard tool system $k_{SYS,STND} = 3$
L	mm	Length of a single tool or a tool system component
L_B	mm	Distance between the spindle bearings B1 and B2
L_{BL}	mm	Maximum length from RP to plane P2 that still enables mass compensation, i.e. $L_{BL} < L$
L_{CG}	mm	Lever arm from RP to the tool centre of gravity CG
$L_{CG,i}$	mm	Lever arm to the centre of gravity of component i
$L_{CG,i,SYS}$	mm	Lever arm to the centre of gravity of component i within a tool system
$L_{CG,SYS}$	mm	Lever arm to the centre of gravity of a tool system (distance from RP to CG)
$L_{CG,SYS,i}$	mm	Lever arm to the centre of gravity of a tool system of (i) components
$L_{CG,SYS,k}$	mm	Lever arm to the centre of gravity of a tool system of (k) components
$L_{CG,SYS,3}$	mm	Lever arm to the centre of gravity of a standard tool system of (3) components
L_{CPL}	mm	Distance between the planes of the initial unbalance and the compensating unbalance (in case of a couple unbalance due to quasi-static balancing)
L_{P1}	mm	Distance from the spindle reference point RP to plane P1
L_{P2}	mm	Distance from the spindle reference point RP to plane P2
L_{SYS}	mm	Length of a tool system
$L_{STAT,MAX}$	mm	Maximum length of a tool or a tool system for static balancing
m	g (kg)	Mass of a tool NOTE Input of masses in all formulae in gram (g).
m_{AVG}	g (kg)	Interface-relevant average reference mass of a tool or a tool system
m_i	g (kg)	Mass of tool system component i
m_k	g (kg)	Mass of tool system component k
m_{MAX}	g (kg)	Interface-relevant maximum reference mass of a tool or a tool system
m_{MIN}	g (kg)	Interface-relevant minimum reference mass of a tool or a tool system
m_{SYS}	g (kg)	Mass of a tool system
m_U	g	Unbalance mass
$m_{U,P1}$	g	Unbalance mass at plane P1
$m_{U,P2}$	g	Unbalance mass at plane P2
n	min^{-1}	Rotational speed
$n_{MAX,PER}$	min^{-1}	Maximum permissible rotational speed
n_{SYS}	min^{-1}	Rotational speed of a tool system
P1	—	Balancing plane 1
P2	—	Balancing plane 2
RP	—	Reference point at the spindle nose (e.g. the HSK face)
r	mm	Radius
R_{DYN}	—	Ratio of utilization of the dynamic load rating C_{DYN}
$R_{L/D}$	—	Ratio of tool length to diameter (to decide about static or dynamic balancing)
$R_{STAT,MAX}$	—	Limit ratio for static balancing ($R_{STAT,MAX} = 2,2$)
$R^*_{STAT,MAX}$	—	Limit ratio for static balancing of tools with guidance

Symbols and abbreviated terms	Unit	Description
TM	—	Tool or component manufacturer
U	gmm	Unbalance (NOTE The unit “gmm” is equal to “g·mm”.)
$\vec{U}$	gmm	Unbalance vector
$U_{\mathrm{BM,MIN}}$	gmm	Smallest measurable unbalance of a balancing machine
$U_{\mathrm{BM,ACC}}$	gmm	Measuring accuracy of a balancing machine
U_{CPL}	gmm^2	Couple unbalance
U_{ECC}	gmm	Unbalance due to radial dislocation (eccentricity) relative to the spindle rotary axis
$U_{\mathrm{ECC,i}}$	gmm	Unbalance of component *i* due to radial dislocation relative to the spindle rotary axis
$U_{\mathrm{ECC,MAX}}$	gmm	Unbalance due to a maximum radial dislocation (i.e. eccentricity)
$U_{\mathrm{ECC,i,SYS}}$	gmm	Unbalance due to radial dislocation of component *i* relative to component $i-1$ within a tool system
$U_{\mathrm{ECC,k,MAX}}$	gmm	Maximum unbalance of component *k* due to radial dislocation within a tool system
$U_{\mathrm{G\ (x),PER}}$	gmm	Permissible residual static unbalance according to ISO 1940-1
U_{G40}	gmm	G40 safety unbalance according to ISO 15641
U_{MIN}	gmm	Achievable minimum residual unbalance
$U_{\mathrm{MIN,SYS},k}$	gmm	Achievable minimum unbalance of a tool system of *k* components
$U_{\mathrm{MOM,STAT}}$	gmm	Moment of a static unbalance U_{STAT} (located in CG)
U_{P}	gmm	Unbalance per plane
$U_{\mathrm{P,MIN}}$	gmm	Minimum unbalance per plane
$U_{\mathrm{P,PER}}$	gmm	Permissible residual unbalance per plane
U_{P1}	gmm	Unbalance at balancing plane P1
U_{P2}	gmm	Unbalance at balancing plane P2
$U_{\mathrm{P1,PER}}$	gmm	Permissible residual unbalance at balancing plane P1
$U^{*}_{\mathrm{P1,PER}}$	gmm	$U_{\mathrm{P1,PER}}$ but with the same angular orientation as $U_{\mathrm{P2,PER}}$
$U_{\mathrm{P1,PER,LIM}}$	gmm	Limited permissible residual unbalance at balancing plane P1 (case F)
$U_{\mathrm{P2,PER}}$	gmm	Permissible residual unbalance at balancing plane P2
$U_{\mathrm{P2,PER,LIM}}$	gmm	Limited permissible residual unbalance at balancing plane P2 (case F)
U_{QS}	gmm	Quasi-static unbalance (see Figure 5)
U_{STAT}	gmm	Static unbalance
$U_{\mathrm{STAT,ACT}}$	gmm	Actual measured static unbalance
$U_{\mathrm{STAT,BAL}}$	gmm	Static unbalance weighted by f_{BAL}
$U_{\mathrm{STAT,MAX}}$	gmm	Maximum static unbalance
$U_{\mathrm{STAT,i,MAX}}$	gmm	Maximum static unbalance of component *i* in a tool system of k_{SYS} components
$U_{\mathrm{STAT,i,SYS,PER}}$	gmm	Permissible residual static unbalance of a universal component *i* that may be placed at any axial position within a tools system of k_{SYS} components
$U_{\mathrm{STAT,1\ \%}}$	gmm	Static unbalance to ensure $F_{\mathrm{B1}}/C_{\mathrm{DYN}} \leq 1\ \%$ at spindle bearing B1
$U_{\mathrm{STAT,MAX,A}}$	gmm	Maximum possible static unbalance of case A in Figure 7
$U_{\mathrm{STAT,MAX,B}}$	gmm	Maximum possible static unbalance of case B in Figure 7
$U_{\mathrm{STAT,MAX,C}}$	gmm	Maximum possible static unbalance of case C in Figure 7
$U_{\mathrm{STAT,PER}}$	gmm	Permissible residual static unbalance
$U_{\mathrm{STAT,PER,CS}}$	gmm	Permissible residual static unbalance for the user

Symbols and abbreviated terms	Unit	Description
$U_{STAT,PER,FINE}$	gmm	Permissible residual static unbalance for fine balancing
$U_{STAT,PER,FINE,RES}$	gmm	Resulting permissible residual static unbalance of a fine balanced tool or component taking U_{MIN} and G40 into account (see Figure 13)
$U_{STAT,PER,FINE,4}$	gmm	Permissible residual static unbalance of a component, fine balanced for a tool system of 4 components
$U_{STAT,PER,FINE,5}$	gmm	Permissible residual static unbalance of a component, fine balanced for a tool system of 5 components
$U_{STAT,PER,FINE,6}$	gmm	Permissible residual static unbalance of a component, fine balanced for a tool system of 6 components
$U_{STAT,PER,STND}$	gmm	Permissible residual static unbalance for standard balancing
$U_{STAT,PER,TM}$	gmm	Permissible residual static unbalance for the tool manufacturer
$U_{STAT,SYS,PER}$	gmm	Permissible residual static unbalance of a tool system
$U_{STAT,SYS,PER,FINE}$	gmm	Permissible residual static unbalance of an assembled (quasi monolithic) tool system for fine balancing
$U_{STAT,SYS,PER,STND}$	gmm	Permissible residual static unbalance of an assembled (quasi monolithic) tool system for standard balancing
$U_{STAT,P1,P2}$	gmm	Resulting static unbalance after a dynamic (two planes) balancing process
v_C	m/min	Peripheral speed at the cutting edge
v_{G40}	m/min	Peripheral speed limit of G40 according to ISO 15641→ v_{G40} = 1 000 m/min
v_{REF}	m/min	Peripheral speed of the reference tool diameter (i.e. biggest tool diameter)
x_{P1}	mm	Distance between plane P1 and tool centre of gravity CG
x_{P2}	mm	Distance between plane P2 and tool centre of gravity CG
α	°	Angle
α_{P1}	°	Angular orientation of the unbalance at plane P1
α_{P2}	°	Angular orientation of the unbalance at plane P2
α_U	°	Angle between static and couple unbalance
ρ_{ST}	mg/mm³	Density of steel (7,8 mg/mm³)
Ω	rad/s	Angular velocity of a component or a tool

4 Requirements

4.1 General

4.1.1 Clamping inaccuracies

Unbalances which are not related to the balancing quality of a tool may occur due to clamping inaccuracies caused by fit tolerances, e.g. when inserting a tool into the machine tool spindle. Even if a balancing result stands for a smaller eccentricity than the possible shank eccentricity, this balancing condition cannot be reproducibly achieved with every clamping action of this tool, either in the spindle of a machine tool or of a balancing machine. A radial joining inaccuracy of several micrometres may occur depending on shank type and size (see Table 2 for radial joining dislocations of different tool interfaces). Factors such as wear and run-out of the different interfaces may lead to a worse joining accuracy, thus generating a bigger residual unbalance of the tool-spindle system.

4.1.2 Influence of balancing machines

The achievable residual unbalance of a tool is limited by type and precision of the balancing machine (see 4.2.2 and 4.2.3). Table 2 shows the unbalance measuring limits of balancing machines built for different tool masses.

Systematic eccentricities like a run-out of the balancing spindle can be eliminated by index balancing. This procedure is described in ISO 21940-14.

NOTE ISO 21940-21 describes testing procedures for evaluating the limits and the performance of balancing machines.

4.1.3 Effects and frequent consequences of permissible residual unbalances according to ISO 1940-1

The following two examples show that common balancing qualities based on ISO 1940-1 are already exceeding possible limits.

The frequently required quality level G 2,5 at a rotational speed of 25 000 min^{-1} means a permissible residual unbalance of only 1 gmm/kg. The residual unbalance of 1 gmm for a tool mass of 1 000 g corresponds to a permissible eccentricity of just 1 µm for the tool centre of gravity. This value is lower than a new HSK can repeatedly provide (see Table 2).

In case of tools of even lower masses and higher rotational speeds, the requirements are increasing continuously. A HSK-40-tool of 350 g may only have a residual unbalance of 0,21 gmm (i.e. 0,6 gmm/kg) in order to comply with G 2,5 at a rotational speed of 40 000 min^{-1}. It also means a maximum eccentricity of the tool centre of gravity of only 0,6 µm.

Both examples show that neither the measurement of these residual unbalances nor their realization are reliably possible due to the clamping inaccuracies in a balancing machine itself and the measuring accuracy of commercially available balancing machines.

It also results from ISO 1940-1 that the same quality level permits different residual unbalances for different tool masses at the same rotational speed. Different unbalances as a consequence lead to varying centrifugal forces, i.e. spindle loads. The dynamic spindle load, however, is not dependent on the tool mass but on the unbalance of the tool system and its resulting forces.

4.1.4 Inherent properties of machine tools and components

The vibration amplitudes of a machine structure are related to the exciting force and frequency, as well as to the dynamic properties of the machine tool system. The same grade of excitation leads to higher vibration amplitudes if the frequency-dependent dynamic flexibility of a system is worse at certain frequencies of excitation.

Therefore, as far as the machine and in particular the tool-spindle system are concerned, the balancing requirements depend on the dynamic properties of the tool-spindle system. A universal description of the dynamic properties of machine tools is not possible. However, a limitation of unbalance-related machine vibrations may be achieved by balancing a tool to the limit of the reproducibly achievable residual unbalance, U_{MIN} (see 4.2.3).

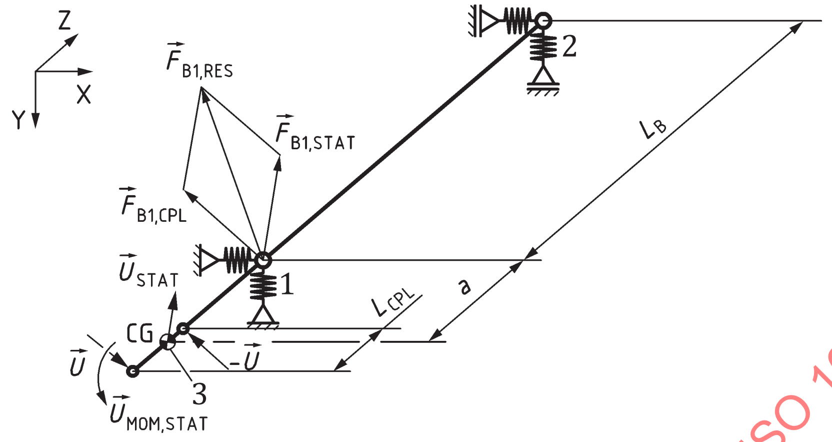
A reduction of machine vibrations can also be achieved by altering the cutting speed if the relevant machining process allows a modification of technological parameters within the relevant operating speed range. Thus, the excitation could take place in a more stable dynamic frequency range of the machine.

4.2 Balancing requirements based on the spindle load

4.2.1 General

In order to limit unbalance-related periodic loads on the spindle bearings, it is necessary to balance tool systems in dependence of the rotational speed and the properties of the spindle systems specified in this document (see Annex A for the theoretical approach). Figure 2 shows the structure and the geometric conditions of the general tool-spindle model with tool unbalances and their related forces for the bearing B1 taking the highest load.

NOTE This document indicates all permissible residual and other unbalances in “gmm”. There are no specific quality levels like “G 6,3” according to ISO 1940-1.



Key

1 bearing B1

2 bearing B2

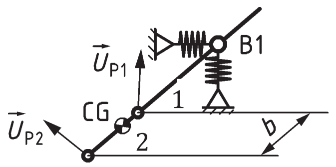
3 centre of gravity CG

Figure 2 — Tool-spindle model showing unbalance-related forces

The universal approach to determine the load $F_{B1,RES}$ at bearing B1 is calculating the vector sum of the forces $F_{B1,STAT}$ and $F_{B1,CPL}$. These forces are generated by a so-called “dynamic unbalance”, the combination of a static unbalance, U_{STAT}, located in the tool centre of gravity and a couple unbalance, U_{CPL}. When static balancing, material should be removed or added at or next to the place of the material imbalance in order to minimize residual dynamic unbalances.

The force $F_{B1,RES}$ on the front spindle bearing B1 shall not exceed 1 % of the dynamic load rating C_{DYN} within the relevant rotational speed range. It is important to note that this dynamic load limit ratio $F_{B1,RES}/C_{DYN}$ = 1 % is independent of the tool mass.

A dynamic unbalance situation of a rigid rotor can be alternatively described by two independent unbalance vectors $\vec{U}_{P1}$ and $\vec{U}_{P2}$ located in two axial planes P1 and P2 with the distance b (see Figure 3). This “two-plane-balancing” procedure has prevailed in the industrial balancing practice of rigid tools and tool systems.



Key

1 plane 1

2 plane 2

Figure 3 — Model of a spindle and a tool with a two-plane dynamic unbalance

The permissible residual static unbalance, $U_{STAT,1\,\%}$ [see Formula (3)], however, has been calculated for a static unbalance located in the tool centre of gravity CG (see Figure 4). The decision whether static or dynamic balancing is required depends on a certain L/D-ratio of the tool (see 4.2.4 for details).

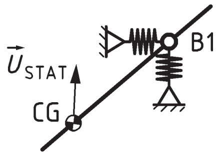
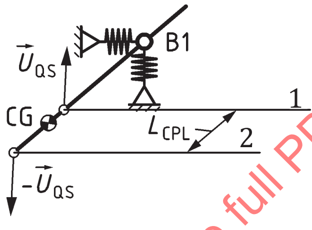


Figure 4 — Static unbalance in the tool centre of gravity CG

The permissible unbalances $U_{P1,PER}$ and $U_{P2,PER}$ for dynamic balancing have been derived from $U_{STAT,PER}$ located in the tool centre of gravity CG and based on the mandatory requirement that the load on spindle bearing B1 shall not exceed the load being generated by the permissible static unbalance, $U_{STAT,PER}$ (see 4.2.5).

A "short" tool according to 4.2.4 is being balanced statically. Due to functionally required tool designs, the centre of the related unbalance is usually not located in the tool centre of gravity CG. For single tools, this so-called quasi-static unbalance, U_{QS} (see Figure 5), is generated by cutting tips and chip flutes, thus often located near the tool front.



Key

1 correction plane

2 centre of quasi-static unbalance

Figure 5 — Couple unbalance due to quasi-static balancing

"At the balancing machine", it is difficult for an operator to define the centre of gravity of the unbalance position. Even if the unbalance position was obvious, removing material directly on the opposite side of the tool body is often not possible either.

Therefore, static mass compensations often happen near the shank due to the bigger tool diameters. This means a distance L_{CPL} between the initial unbalance, U_{QS}, and the correction plane of the compensating unbalance, $-U_{QS}$.

The result is a statically balanced tool with a remaining couple unbalance, U_{CPL} [see Formula (1)] — an exceptional type of dynamic unbalance with the unit (gmm^2).

$$U_{CPL} = U_{QS} \times L_{CPL} \tag{1}$$

The distance L_{CPL} varies unpredictably depending on the tool design and the position(s) of the balancing measures. Therefore, the forces on the spindle bearings induced by a couple unbalance, U_{CPL}, are unknown and cannot be taken into account for static balancing. Nevertheless, it is possible to calculate the bearing forces F_{B1} and F_{B2} which are equal for both bearings because of the couple unbalance (see Formula (2)).

$$F_{B1} = F_{B2} = \frac{U_{CPL}}{L_B} \times \Omega^2 = U_{QS} \times \frac{L_{CPL}}{L_B} \times \left(\frac{2\pi \times n}{60}\right)^2 \tag{2}$$

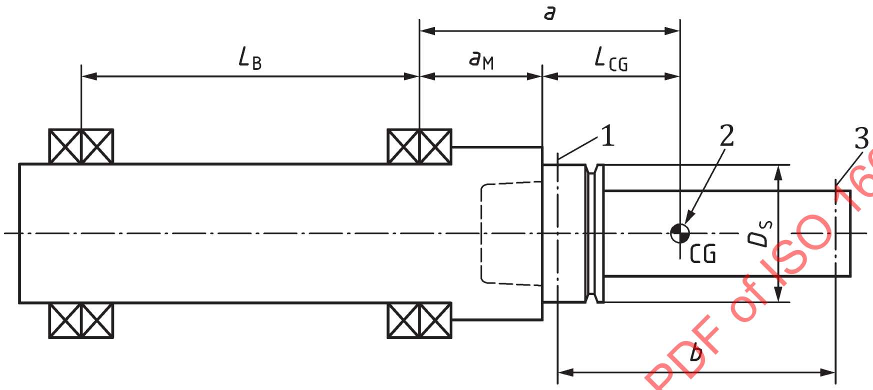
Formula (2) also shows that the ratio of L_{CPL} and L_B has a significant influence on the bearing forces.

In practice, this aspect has not been taken into account for static balancing. A.5.2 shows for a single tool with one cutting edge — statically "worst case" balanced with the biggest possible distance L_{CPL} — that this effect is far below the "1 % of C_{DYN} rule" and may therefore be neglected for static balancing.

When static balancing, material should be removed or added at or next to the place of the material imbalance in order to minimize residual dynamic unbalances.

The spindle parameters C_{DYN}, a_M and L_B are specific for each interface type and size (see Table 2).

The schematic sketch of a tool-spindle system in Figure 6 shows the main parameters that are also part of Figure 2.



Key

1 plane P1

2 centre of gravity CG

3 plane P2

Figure 6 — Model of a tool-spindle system showing the main parameters

The limitation of the loads on the bearings is the basis both for static and dynamic balancing in this document. The bearing planes represent the tolerance planes of the tool-spindle system in the sense of ISO 1940-1. As tools and tool systems are the only varying components of a tool-spindle system, it would not be sensible to add/remove material to/from a machine tool spindle.

So the planes P1 and P2 at the tool body are the correction planes of the tool-spindle system. At the same time, they are also tolerance and correction planes of the tool systems, thus enabling dynamic balancing independently from the machine tool spindle. The permissible dynamic unbalances (see 4.2.5) are calculated for the planes P1 and P2 which can be individually defined for over-mounted tool systems according to the rules of this document.

4.2.2 Determination of the balancing requirements

Formula (3) shows the residual unbalance at a maximum bearing load ratio of $F_{B1,RES}/C_{DYN}$ = 1 % at spindle front bearing B1. As already mentioned in NOTE 1 of the scope, this document has been initially developed for HSK interfaces (sizes HSK-25 to HSK-100) but is also applicable to shanks and spindles of similar quality and known possible clamping dislocations (see Table 2).

$$U_{STAT,1\,\%} = 9{,}12 \times 10^5 \times \frac{C_{DYN}}{n^2} \times \left(\frac{L_B}{L_B + a} \right) \quad (3)$$

with

$$a = a_M + L_{CG} \quad (4)$$

The residual unbalances calculated according to Formula (3) represent the range in which the spindle bearings remain undamaged.

In order to give an option to adapt the balancing quality to different machining processes, two weighting factors f_{BAL} for standard and fine balancing are proposed in Table 1. Both f_{BAL} factors can be applied to single tools and tool systems (see Clause 5 for components).

Table 1 — Weighting factors f_{BAL}

Standard balancing	$f_{BAL} = f_{BAL,STND} = 0{,}8$
Fine balancing	$f_{BAL} = f_{BAL,FINE} = 0{,}2$

Formula (4) and f_{BAL} modify Formula (3) to

$$U_{STAT,BAL} = f_{BAL} \times U_{STAT,1\,\%} = 9{,}12 \times 10^5 \times f_{BAL} \times \frac{C_{DYN}}{n^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG}} \right) \quad (5)$$

The permissible residual static unbalance, $U_{STAT,PER}$, also has to take the possible fault U_{MIN} [for details, see Formula (10) to Formula (12)] into account.

$$U_{MIN} = U_{BM,ACC} + U_{ECC} \quad (6)$$

A tool will not exceed $U_{STAT,1\,\%}$ in a machine tool spindle if U_{MIN} had been subtracted, thus defining the permissible static unbalance, $U_{STAT,PER}$, for its balancing process to

$$U_{STAT,PER} = U_{STAT,1\%} - U_{MIN} \quad (7)$$

So, the formula of the permissible static unbalance finally is

$$U_{STAT,PER} \le 9{,}12 \times 10^5 \times \frac{f_{BAL} \times C_{DYN}}{n^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG}} \right) - \left(U_{BM,ACC} + m \times e_S \right) \quad (8)$$

Tools and tool systems show certain radial and angular run-outs to the spindle axis (i.e. the axis of rotation) after insertion into the machine tool spindle. In the worst case, the full radial dislocation e_S might occur. In order to estimate the possible maximum actual unbalance in practice, U_{ECC} needs to be added again to $U_{STAT,PER}$ calculated with Formula (8).

$$U_{STAT,MAX} \le U_{STAT,PER} + U_{ECC} = U_{STAT,PER} + m \times e_S \quad (9)$$

NOTE Radial dislocations also occur between components of modular tool systems. B.1 shows a calculation example for a tool system consisting of three components.

The dynamic load rating, C_{DYN}, the machine lever arm, a_M, and the distance between the spindle bearings, L_B (see Figure 2 and Figure 6), are specific for any spindle interface and can be taken from Table 2. The total lever arm from the front bearing to the tool centre of gravity is composed of the machine lever arm, a_M (see Table 2), and the tool lever arm, L_{CG}. The tool related lever, L_{CG}, shall be specified by the user and is defined as the distance from the reference position RP at the spindle nose (e.g. the HSK face) to the tool centre of gravity, CG. The factor 9,12 × 10^5 arises from the converted units of the different parameters. The rotational speed, n, has to be entered in min^{-1}.

The smallest achievable unbalance is determined by the measurable unbalance minimum, $U_{BM,MIN}$, depending on the quality of the balancing machine. This is only possible without any run-out fault.

In practice the minimal unbalance measurement, U_{MIN}, might be bigger than $U_{BM,MIN}$. It depends on the measuring accuracy of balancing machines, $U_{BM,ACC}$, and on U_{ECC} caused by radial dislocation(s). U_{MIN} has to be subtracted from $U_{STAT,1\,\%}$ so that the permissible static unbalance, $U_{STAT,PER}$, of a tool shall not exceed $U_{STAT,1\,\%}$ even if both effects occurred during the balancing process or/and afterwards in the machine tool spindle.

In case of heavy tools and high rotational speeds, $U_{STAT,PER}$ may take on low values near U_{MIN}. This system status, however, may only be achievable by balancing the joined tool-spindle system composed of spindle and tool system.

If the unbalance $U_{ECC,MAX}$ caused by eccentricity happens to exceed the unbalance $U_{STAT,1\,\%}$, the permissible unbalance $U_{STAT,PER}$ takes on even negative values. U_{MIN}, however, remains the lowest possible value. Formula (10) to Formula (12) give an idea of these circumstances.

Table 2 — Interface specific parameters

			SZ								
	Name	ISO	1	2	3	4	5	6	7	8	9
Shank type	HSK	12164-1 12164-2	25	32	40	50	63	80	100	125	160
Shank type	HSK	other[a]	32	40	50	63	80	100	125	160	200
Shank type	PSC	26623-1 26623-2	—	32	40	50	63	80	100	—	—
Shank type	TS	26622-1 26622-2	—	32	40	50	63	80	100	—	—
Shank type	Taper 7/24	7388-1 7388-2 9270-1 9270-2	—	—	30	—	40	45	50	—	60
Interface-relevant component mass of a tool or tool system		m_{MIN} [kg][b]	0,1	0,2	0,4	0,6	1,0	2,0	3,0	5,0	10
Interface-relevant component mass of a tool or tool system		m_{MAX} [kg][b]	0,3	0,6	1,2	2,0	4,5	7,0	13	25	60
Interface-relevant component mass of a tool or tool system		m_{AVG} [kg]	0,2	0,4	0,8	1,3	2,8	4,5	8,0	15	35
Parameters		Units									
C_{DYN}		N	6 800	8 800	12 200	17 600	25 000	30 000	42 500	42 500	42 500
a_M		mm	20	25	35	45	50	60	90	110	130
L_B		mm	170	200	230	300	415	650	730	730	730
D_S[c]	HSK	mm	25	32	40	50	63	80	100	125	160
D_S[c]	PSC	mm	—	32	40	50	63	80	100	—	—
D_S[c]	TS	mm	—	32	40	50	63	80	100	—	—
D_S[c]	7/24	mm	—	—	50	—	63,55	82,55	97,50	—	155
e_S	HSK	mm	0,002	0,002	0,002	0,002	0,002	0,003	0,004	0,004	0,004
e_S	PSC	mm	—	0,002	0,002	0,002	0,002	0,003	0,004	—	—
e_S	TS	mm	—	0,002	0,002	0,002	0,002	0,003	0,004	—	—
e_S	7/24	mm	—	—	0,003	—	0,003	0,004	0,005	—	0,006
$U_{BM,MIN}$[d,e]		gmm	0,75	0,75	0,75	0,75	0,75	0,75	1,5	3,0	3,0
$U_{BM,ACC}$[f]		gmm	0,75	0,75	0,75	0,75	0,75	0,75	1,5	3,0	3,0
b_{MIN}[e]		mm	60	60	60	60	60	60	80	100	100

[a] “other” means one size smaller taper shank compared to the flange diameter (e. g. flange Ø 63mm combined with the taper of a HSK-50)

[b] m_{MIN} and m_{MAX} are reference masses to classify components with non-listed shank types for the spindle sizes 1 to 9.

[c] D_s is the diameter of the shank flange.

[d] Balancing for $U_{BM,MIN}$ requires optimal clamping conditions in the balancing machine.

[e] $U_{BM,MIN}$ and b_{MIN} refer to the average component mass m_{AVG}.

[f] The values of $U_{BM,ACC}$ can be set equal to $U_{BM,MIN}$.

4.2.3 Measuring accuracy of balancing machines, influence of run-out and repeatability of measuring results

The measuring accuracy of balancing machines is limited and unbalance measurements may also vary due to the influence of different faults occurring during the measuring sequence.

Spindle interfaces of balancing machines as well as tool/component shanks show certain minimal radial and angular run-outs. The corresponding unbalance, U_{ECC}, takes relevant effect as the measuring result depends on the joining accuracy in the balancing machine tool spindle. Table 2 shows reference values of e_S for the most common shank interfaces. The radial clamping dislocation in the balancing machine is supposed to be better than in a machine tool spindle.

The second factor is the smallest measurable unbalance, $U_{BM,MIN}$, which depends on the type and size of balancing machines. Low tool masses (<1,5 kg) make the measuring limits of balancing machines more apparent. $U_{BM,ACC}$ is the measuring accuracy of balancing machines. Values for $U_{BM,MIN}$ and $U_{BM,ACC}$ are listed in Table 2.

U_{ECC} and $U_{BM,ACC}$ sum up to the reproducibly measurable residual unbalance, U_{MIN}, of a tool-spindle system as shown in Formula (10). If better balancing results become necessary, e.g. due to application-related requirements, solely balancing of the complete tool-spindle system (machine tool spindle and tool system in their joint condition) may be an option.

As mentioned above, the accuracy of balancing depends on the measuring accuracy of balancing machines, $U_{BM,ACC}$, and the influence of limited run-out, e_S (see Table 2).

$$U_{MIN} \le U_{BM,ACC} + U_{ECC} = U_{BM,ACC} + m \times e_S \quad (10)$$

Adjusting and eliminating radial and angular run-out faults (i.e. $U_{ECC} \to 0$) may lead to the smallest possible value of U_{MIN}, as shown in Formula (11):

$$U_{MIN} \ge U_{BM,MIN} \quad (11)$$

what finally defines the range of U_{MIN} to Formula (12):

$$U_{BM,MIN} \le U_{MIN} \le U_{BM,ACC} + U_{ECC} \quad (12)$$

Measuring dynamic unbalances is more complex and difficult compared to static unbalances. As a consequence, the minimal permissible residual unbalance per plane, $U_{P,\,MIN}$, cannot be less than U_{MIN}, i.e. Formula (13):

$$U_{P,\,MIN} \ge U_{MIN} \quad (13)$$

This means in the worst case (i.e. the unbalances of both planes have the same angular orientation) that the resulting static unbalance, $U_{STAT,P1,P2}$, may take on twice the amount of U_{MIN} [see Formula (14)]:

$$U_{STAT,P1,P2} = \left|\vec{U}_{P1,PER} + \vec{U}_{P2,PER}\right| \ge \left|\vec{U}_{P1,PER}\right| + \left|\vec{U}_{P2,PER}\right| \ge 2 \times U_{MIN} \quad (14)$$

$U_{STAT,P1,P2}$ of a dynamically balanced tool may statically only reach the minimal value U_{MIN} with certainty if it is being finish balanced statically.

U_{MIN} is the horizontal bottom line in Figure 13 showing the graph of a typical HSK-63 tool with a mass of 1 000 g.

Values below U_{MIN}, even if displayed by a balancing machine, may not be reproducibly measurable after a subsequent tool changing action. So, balancing results below the horizontal lines of U_{MIN} may only be achievable by balancing the assembled tool-spindle system.

Different balancing conditions at tool manufacturers (TM) and users (CS) can lead to different balancing results for one and the same tool (e.g. caused by tool adapters of different clamping accuracies, brand, measuring quality, maintenance state of balancing machines, etc.). To prevent irritations due to this, it is recommended to implement a tolerance range of ±15 % for $U_{STAT,PER}$ on both the manufacturer's and the user's side.

This means that the permissible unbalance values for both static and dynamic balancing may be lowered for the initial balancing process and increased for a subsequent verification by 15 % each. In case of dynamic balancing, this tolerance range should be applied per plane.

NOTE ISO 1940-1 and ISO 21940-14 recommend a detailed consideration of machine-related faults.

EXAMPLE In case of a permissible static unbalance of 10 gmm, a tool has to be balanced to $U_{STAT,PER,TM}$ = 8,5 gmm (−15 %) by the manufacturer. For subsequent verification on a different balancing machine (e.g. at the user), a residual unbalance of $U_{STAT,PER,CS}$ = 11,5 gmm is a sufficient value.

4.2.4 Application criterion of static and dynamic balancing

The parameter $R_{L/D}$ to decide between either static (one-plane balancing) or dynamic balancing (two-plane balancing) is the ratio between L_{BL} and D_S, and defined as Formula (15):

$$R_{L/D} = \frac{L_{BL}}{D_S} \tag{15}$$

The limit ratio $R_{STAT,MAX}$ to indicate the beginning of dynamic balancing is shown in Formula (16):

$$R_{STAT,MAX} = \frac{L_{STAT,MAX}}{D_S} \leq 2,2 \tag{16}$$

L_{BL} and $L_{STAT,MAX}$ are lengths from the spindle reference point RP (e.g. the HSK face) to the front most possible balancing plane (where the tool body still enables a mass compensation) and, therefore, shorter than the tool length, L, i.e. $L_{BL} \leq L$. The size of the interface represents the reference diameter, D_S, (see Figure 6), e.g. HSK-A63 → D_S = 63 mm.

Tools and tool systems that exceed $L_{STAT,MAX}$ (i.e. $R_{L/D} > R_{STAT,MAX}$) should be balanced dynamically.

Dynamic balancing, however, is only sensible if L_{BL} is bigger than the minimal distance, b_{MIN}, between the balancing planes, P1 and P2. This dimension can be determined for different balancing machines from Table 2.

4.2.5 Permissible residual dynamic unbalances

4.2.5.1 General

The derivation of the permissible residual dynamic unbalances also follows the guideline of the static balancing that the dynamic load rating on the front spindle bearing(s) shall not exceed 1 % of C_{DYN}. This is the case if the load of bearing, B1, caused by a dynamic unbalance of 2-planes corresponds with the load of a static unbalance at the same conditions.

Unbalances are vectors of varying length and angular orientation, perpendicular to the rotating axis of the tool. In order to keep dynamic balancing jobs as simple as possible and to retain the prevailed tooling balancing practice, the permissible dynamic unbalances shall not depend on their angular orientation. This is the case if the sum of the absolute values of the unbalance vectors of planes, P1 and P2, cannot exceed the permissible static unbalance, $U_{STAT,PER}$.

In case of a dynamic unbalance, the static unbalance can be calculated by adding the two unbalance vectors of the balancing planes, P1 and P2. Figure 7 shows three unbalance situations A, B and C of different angles between the unbalance vectors $\vec{U}_{P1}$ and $\vec{U}_{P2}$.

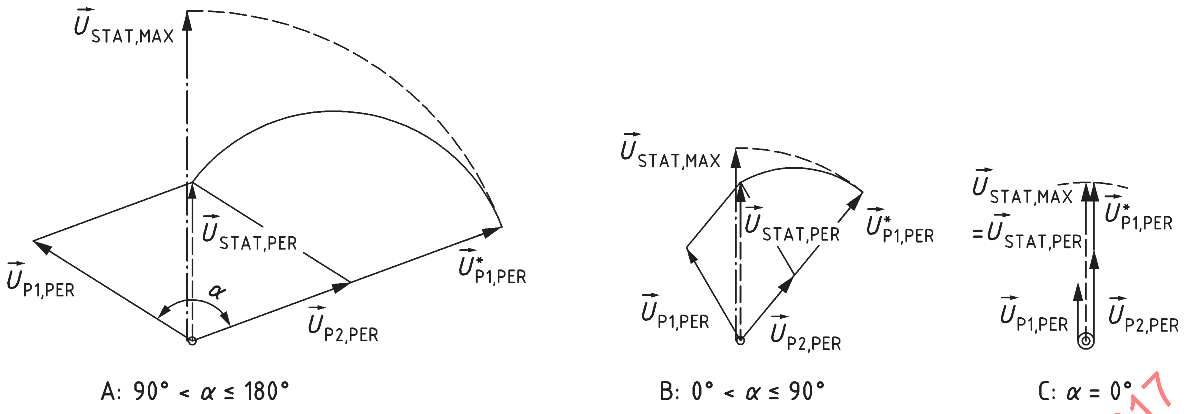


Figure 7 — $\vec{U}_{STAT,MAX}$ and $\vec{U}_{STAT,PER}$ depend on the angle α between $\vec{U}_{P1,PER}$ and $\vec{U}_{P2,PER}$

$\vec{U}_{P1,PER}$ is the vector of the permissible residual static unbalance with the same orientation like $\vec{U}_{P2,PER}$. All three dynamic unbalance situations of Figure 7 result in the same absolute value (i.e. length) of $\vec{U}_{STAT,PER}$. Different angles between the unbalances of the planes, P1 and P2 (see cases A and B), however, lead to different possible maximum values of $\vec{U}_{STAT,MAX}$ which are bigger than $\vec{U}_{STAT,PER}$. Only case C with both unbalances having the same orientation, i.e. α = 0° between $\vec{U}_{P1,PER}$ and $\vec{U}_{P2,PER}$, takes care that $\vec{U}_{STAT,MAX}$ and $\vec{U}_{STAT,PER}$ are mandatorily equal.

NOTE 1 The bodies of tools, tool components and modular tool systems are assumed to be rigid in the range of their operating speeds.

NOTE 2 The process of dynamic balancing by using two balancing planes is well-established in the industrial practice and also applied in this document. The cases D, E and F of dynamic balancing result from three different possible configurations of the two balancing planes and the centre of gravity. 4.2.5.2 and 4.2.5.3 provide the relevant calculations.

NOTE 3 Detailed mathematical derivations of the three dynamic balancing cases are shown in the A.4.1 and A.4.2.

NOTE 4 The calculations of the permissible unbalances of the cases D, E and F are based on the same angular orientation of both unbalances of plane P1 and plane P2 (i.e. α = 0°; see case C in Figure 7).

Formula (17) represents this as follows:

$$U_{STAT,PER} = U_{STAT,MAX,C} < U_{STAT,MAX,B} < U_{STAT,MAX,A} \tag{17}$$

For α = 0°, the determination of the maximum possible unbalance $U_{STAT,MAX}$ created by the unbalances of the planes P1 and P2 is shown in Formula (18):

$$U_{STAT,MAX} = \left|\vec{U}_{STAT,MAX}\right| = \left|\vec{U}_{P2,PER} + \vec{U}^*_{P1,PER}\right| = \left|\vec{U}_{P2,PER}\right| + \left|\vec{U}_{P1,PER}\right| \tag{18}$$

4.2.5.2 Case D: Tool centre of gravity CG between the planes P1 and P2 (P1-CG-P2)

NOTE Case D "P1-CG-P2" represents the standard case and should be preferred to the special cases "CG-P1-P2" (case E, 4.2.5.3.2) and "P1-P2-CG" (case F, 4.2.5.3.3).

The derivation of Formula (19) and Formula (20) for the permissible residual unbalances $U_{P1,PER}$ and $U_{P2,PER}$ is based on Figure 8 which also shows the equivalent permissible static unbalance $U_{STAT,PER}$ (dashed line) for a better overview. All unbalances are supposed to have the same angular orientation as described along case C of Figure 7.

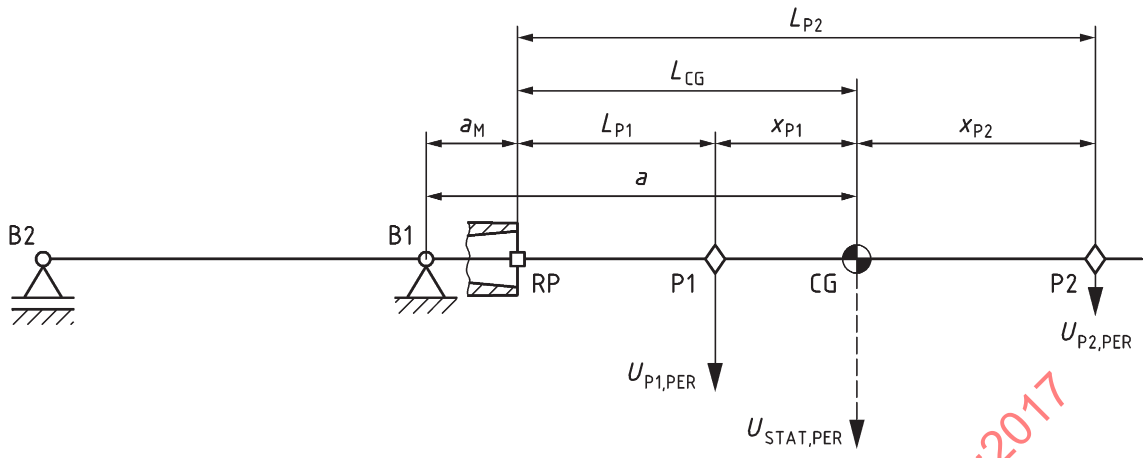


Figure 8 — Case D: Tool centre of gravity CG between planes P1 and P2 (P1-CG-P2)

The load on the spindle bearings due to the dynamic unbalances of planes P1 and P2 shall not be greater than the load due to the permissible static unbalance located in the centre of gravity.

The equations of equilibrium for the unbalances $U_{STAT,PER}$, $U_{P1,PER}$ and $U_{P2,PER}$ and their unbalance moments according to Figure 8 finally lead to the following permissible unbalances for the planes P1 and P2. The length parameters of Formula (19) and Formula (20) refer to the centre of gravity CG.

$$U_{P1,PER} = U_{STAT,PER} \times \frac{x_{P2}}{x_{P1} + x_{P2}} \quad (19)$$

$$U_{P2,PER} = U_{STAT,PER} \times \frac{x_{P1}}{x_{P1} + x_{P2}} \quad (20)$$

At the balancing machine, it may be more suitable if the length parameters of the equations for $U_{P1,PER}$ and $U_{P2,PER}$ refer to the reference point (RP). This leads to the alternative Formula (21) and Formula (22).

$$U_{P1,PER} = U_{STAT,PER} \times \frac{L_{P2} - L_{CG}}{L_{P2} - L_{P1}} \quad (21)$$

$$U_{P2,PER} = U_{STAT,PER} \times \frac{L_{CG} - L_{P1}}{L_{P2} - L_{P1}} \quad (22)$$

If there is an asymmetric position of the balancing planes to the centre of gravity (i.e. one of the planes is too close to CG), it can come to an unfavourable split of the permissible static unbalance $U_{STAT,PER}$. A sensible split of the permissible static unbalance is achieved if the minimal share per plane does not get less than 20 % of $U_{STAT,PER}$ ($f_{P,MIN} = 0{,}2$). A similar situation is being described in ISO 1940-1:2003, 7.2.2 and 7.2.3 for the tolerance planes at the bearings.

It applies Formula (23):

$$U_{P,PER} \geq U_{P,MIN} = f_{P,MIN} \times U_{STAT,PER} = 0{,}2 \times U_{STAT,PER} \geq U_{MIN} \quad (23)$$

The dynamic unbalance values $U_{P,PER}$ of each plane should not fall below U_{MIN}. This may easily happen with low values of $U_{STAT,PER}$ which are slightly bigger than U_{MIN}.

4.2.5.3 Tool centre of gravity CG not between planes P1 and P2

4.2.5.3.1 General

Instead, between the two planes P1 and P2, the centre of gravity CG can either be located between reference point RP and plane P1 (case E, Figure 9) or beyond plane P2 (case F, Figure 10). In both cases, Formula (A.16) and Formula (A.18) lead to a negative unbalance value at one of the two planes. In terms of technical mechanics, this would be correct because the permissible residual unbalances of the two planes were opposite to each other.

This, however, would mean that unbalances were to be treated as vectors. Only taking their angular orientations into account could safely prevent the residual static unbalance from exceeding $U_{STAT,PER}$. In the tooling industry, this has not been required yet and would not be practicable as well.

Therefore, two different ratios of x_{P1} and x_{P2} are being introduced in Formula (24) and Formula (29) to take care of this issue, as well as to address the smaller permissible unbalance value to plane P2 having the longer lever arm.

4.2.5.3.2 Case E: Tool centre of gravity CG between spindle reference, RP, and plane P1 (CG-P1-P2)

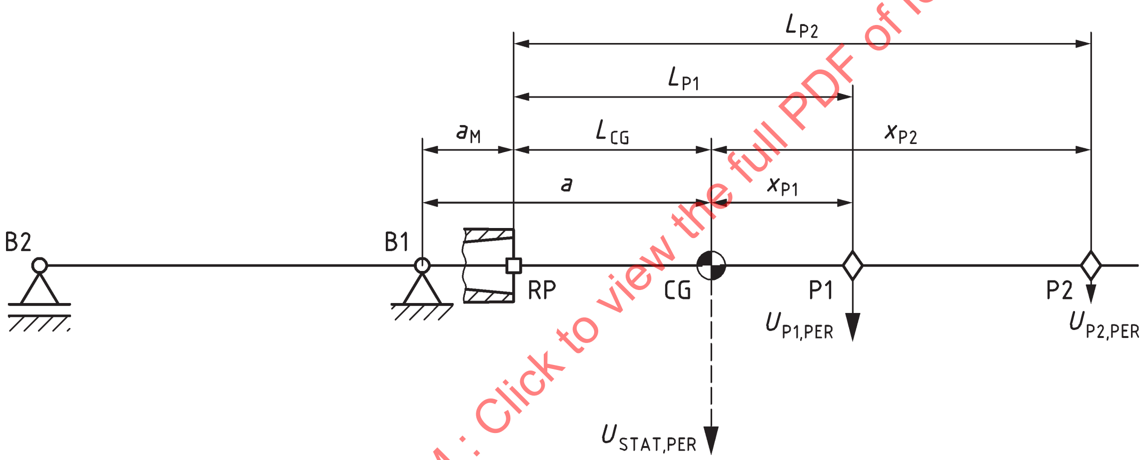


Figure 9 — Case E: Tool centre of gravity, CG, between spindle reference point, RP, and plane P1 (CG-P1-P2)

Formula (24) ensures that plane P2 with the longer lever arm to the front bearing gets the smaller share of the permissible unbalance in case E of Figure 9.

$$\frac{U_{P1,PER}}{U_{P2,PER}} = \frac{x_{P2}}{x_{P1}} \tag{24}$$

Together with Formula (A.34) and Formula (A.35), the results referring to the tool centre of gravity CG are:

$$U_{P1,PER} = U_{STAT,PER} \times \frac{(a_M + L_{CG}) \times x_{P2}}{(a_M + L_{CG}) \times (x_{P1} + x_{P2}) + 2 \times x_{P1} \times x_{P2}} \tag{25}$$

$$U_{P2,PER} = U_{STAT,PER} \times \frac{(a_M + L_{CG}) \times x_{P1}}{(a_M + L_{CG}) \times (x_{P1} + x_{P2}) + 2 \times x_{P1} \times x_{P2}} \tag{26}$$

Referring to the reference point, RP, Formula (25) and Formula (26) can be transformed into Formula (27) and Formula (28):

$$U_{P1,PER} = U_{STAT,PER} \times \frac{(a_M + L_{CG}) \times (L_{P2} - L_{CG})}{(a_M + L_{CG}) \times (L_{P1} + L_{P2} - 2 \times L_{CG}) + 2 \times (L_{P1} - L_{CG}) \times (L_{P2} - L_{CG})} \quad (27)$$

$$U_{P2,\ PER} = U_{STAT,PER} \times \frac{(a_M + L_{CG}) \times (L_{P1} - L_{CG})}{(a_M + L_{CG}) \times (L_{P1} + L_{P2} - 2 \times L_{CG}) + 2 \times (L_{P1} - L_{CG}) \times (L_{P2} - L_{CG})} \quad (28)$$

Both lever arms of planes P1 and P2 are longer compared to the lever of CG with the consequence of relatively small permissible residual unbalances, $U_{P1,PER}$ and $U_{P2,PER}$. Their sum is less than $U_{STAT,PER}$ and may easily lead to values below U_{MIN}.

Case E usually does not occur in practice or should be avoided if possible.

4.2.5.3.3 Case F: Tool centre of gravity CG beyond plane P2 (P1-P2-CG)

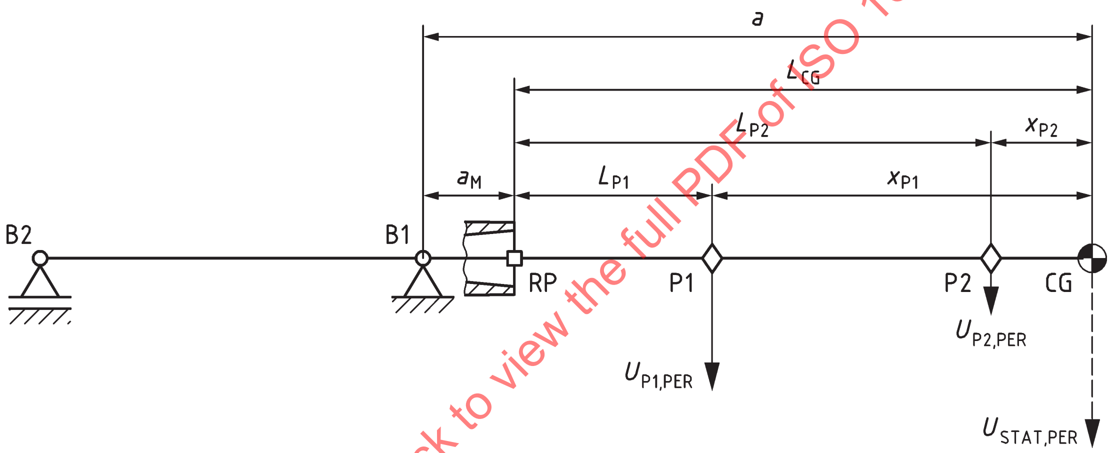


Figure 10 — Case F: Tool centre of gravity, CG, beyond plane P2 (P1-P2-CG)

Case F of Figure 10 is more likely in practice than case E. Providing the smaller share of the unbalance again to plane P2 (the plane with the longer lever arm) leads to the split-up ratio of Formula (29):

$$\frac{U_{P1,PER}}{U_{P2,PER}} = \frac{x_{P1}}{x_{P2}} \quad (29)$$

Case F with the smaller lever arms of P1 and P2 compared to L_{CG}, however, leads to relatively big values for $U_{P1,PER}$ and $U_{P2,PER}$ according to the equilibrium of unbalance moments of Formula (A.29). Their sum would be bigger than $U_{STAT,PER}$ while the spindle load was still correctly limited. Depending on the angular orientation, this may lead to a higher static unbalance measurement than $U_{STAT,PER}$ and cause irritations in practice. Therefore, the two unbalances $U_{P1,PER}$ and $U_{P2,PER}$ shall be limited so that their sum cannot exceed $U_{STAT,PER}$ (see A.4.2.3 for details).

Together with the split-up ratio of Formula (29), the corrected permissible residual unbalances, $U_{P1,PER,LIM}$ and $U_{P2,PER,LIM}$, referring to CG are given in Formula (30) and Formula (31):

$$U_{P1,PER,LIM} = U_{STAT,PER} \times \frac{x_{P1}}{x_{P1} + x_{P2}} \quad (30)$$

$$U_{P2,PER,LIM} = U_{STAT,PER} \times \frac{x_{P2}}{x_{P1} + x_{P2}} \tag{31}$$

The corresponding Formula (32) and Formula (33) refer to the reference point, RP:

$$U_{P1,PER,LIM} = U_{STAT,PER} \times \frac{L_{CG} - L_{P1}}{2 \times L_{CG} - (L_{P1} + L_{P2})} \tag{32}$$

$$U_{P2,PER,LIM} = U_{STAT,PER} \times \frac{L_{CG} - L_{P2}}{2 \times L_{CG} - (L_{P1} + L_{P2})} \tag{33}$$

A too small distance from plane P2 to the gravity centre CG would here as well lead to an unfavourable split of the permissible static unbalance. The minimum unbalance value per plane, $U_{P,\ MIN}$, according to Formula (23) has to be kept here as well.

Finally, the application of the general rule of Formula (10) prevents that any calculated plane unbalance falls below U_{MIN} for case F.

4.2.6 Balancing requirements for tool systems with guidance

Cutting tools with guidance in and/or outside the bore (e.g. slim tools for the machining of valve guide bores or long boring bars for camshaft bores) transfer their cutting forces and the load of their residual unbalances mainly to the supporting points along their tool body. The specification of the permissible unbalance values for each support bearing location is up to the manufacturers' choice.

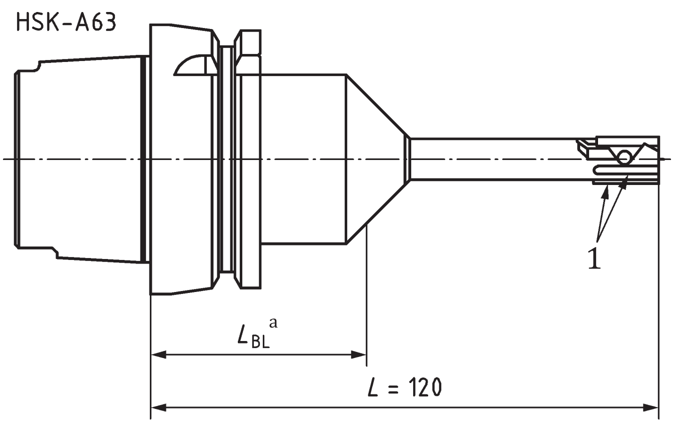
Standard tools with guiding properties like twist drills, core drills and reamers are usually designed more or less symmetrical. Gun drills, as well as special reamers and fine boring tools, however, often have to be designed asymmetrically due to their required function.

These tools are supported in the bore by their own guiding elements (e.g. guide pads). Cutting forces and unbalance-related centrifugal forces are transferred by the guide pads to the work piece. Thus, the spindle bearings are not or only partly strained by centrifugal forces due to unbalance at plane P2. This kind of tools can be balanced according to the following description.

NOTE Due to the design of tools with guidance (L_{BL} is too short or even not available; see Figures 11 and 12), the calculation of the ratio $R^{*}_{STAT,MAX}$ refers to L instead of L_{BL} [see Formula (15)], as shown in Formula (34).

$$R^{*}_{STAT,MAX} = \frac{L}{D_S} = 2{,}2 \tag{34}$$

Short tools ($L < 2{,}2 \times D_S$; see 4.2.4) with guide pads shall be balanced statically at the marked balancing area in Figure 11.



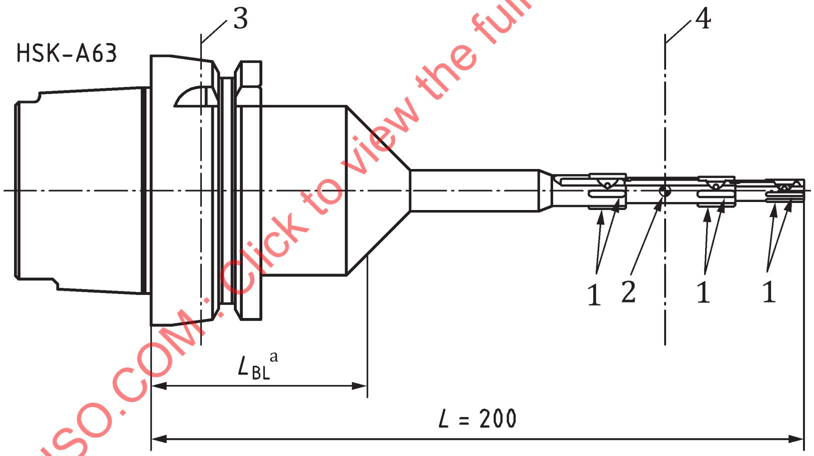
Key

1 guide pads

a Balancing area.

Figure 11 — Short fine boring tool with guide pads, $L < 2,2 \times D_S$ and $L_{BL} < b_{MIN}$

Longer tools ($L > 2,2 \times D_S$) with guide pads as shown in the example of Figure 12 often show considerable unbalances in the cutting area at balancing plane P2 due to chip grooves, etc.



Key

1 guide pads

2 centre of unbalance

3 plane P1

4 plane P2

a Balancing area.

Figure 12 — Longer fine boring tool with guide pads, $L > 2,2 \times D_S$ and $L_{BL} < b_{MIN}$

The angular position of the unbalance at plane P2 is located opposite to the chip groove near the guide pads which mostly consist of material of higher density than steel (e.g. carbide).

At small diameter tools, these unbalances cannot be removed for stability reasons. A symmetrical design of such tools is not possible either due to the required function. An appropriate dynamic balancing procedure of tools according to Figure 12 is described below.

Positioning the plane P2 in the centre of an unbalance simulates the machining situation because the unbalance load of P2 will be transferred by the guide pads to the bore of the work piece and its fixture on the machine tool table. Thus, there is no significant load on the spindle caused by the unbalance of plane P2, making this (impossible) balancing procedure obsolete. The unbalance of plane P1 outside the bore, however, would stress the spindle bearings when the tool is in operation and should, therefore, be removed.

Pure static balancing of such tools at the balancing area near the HSK would lead to a considerably worse dynamic balancing condition in operation because a higher unbalance than before would remain at plane P1.

Tools as shown in Figure 12 often have no guidance outside the work piece. A recommended procedure is to enter the work piece that often has a pilot bore at a reduced rotational speed and switch to normal parameters when the guidance is active. Detailed advice should be given by the manufacturers of this kind of tools.

In order to minimize residual unbalances, a rotationally symmetrical design of the tool systems should be generally aimed at.

4.2.7 Influence of the HSK (hollow taper shank) on the dynamic unbalance

Static balancing of HSK adapters with the predominantly used HSK types A and C results in remaining dynamic unbalances. In this case, it is a couple unbalance (see Figure 5) due to the asymmetrical HSK design.

This shows particular effect on simple modular tool systems consisting of a short HSK adapter and a single tool. Here rotation symmetrical adapters like shrink fit chucks, hydraulic expansion chucks or collet chucks are used in combination with symmetrical standard tools like drills and milling cutters as well as statically or dynamically balanced special tools.

Most HSK adapters are statically balanced at the HSK flanges according to the provisions of ISO 12164-1. The axial distance between the two drive keys of different depths (the actual unbalance) and the balancing plane at the flange collar creates a couple unbalance. Balancing of modularly designed HSK tools thus determines certain residual dynamic unbalances.

Although static balancing of HSK adapters has prevailed in practice due to a low influence on the tool-spindle system (see A.5.2), measurements of residual dynamic unbalance often lead to irritations and complaints of the users. The reason for those unbalances, however, is exclusively caused by the above described mandatory balancing of HSK adapters type A and C.

Table 3 shows the relevant two-plane unbalances for different HSK sizes. Balancing plane P2 has an exemplary distance of 150 mm to plane P1 which is located at the HSK face.

The balancing check of a statically balanced HSK tool or tool system according to the positions of the balancing planes of Table 3 enables to determine whether a residual dynamic unbalance has been mainly caused by the HSK shank.

Table 3 — Dynamic unbalances of statically balanced HSK-A and HSK-C shanks

HSK size	Static unbalance	Dynamic unbalance per plane[b]	Couple unbalance[a]	Angle[c]		Unbalance mass
D_S[d]	U_{STAT}	$U_P = U_{P1} = U_{P2}$ (h_{P1} = 0 mm) (h_{P2} = 150 mm)	U_{CPL} (= U_P × 150 mm)	a_{P1}	a_{P2}	$m_{U,P1} = m_{U,P2}$
mm	gmm	gmm	gmm²	°	°	g
25[e]	0	0,2	30	0	180	0,02
32	0	0,5	75	25	205	0,03
40	0	1,5	225	15	195	0,08
50	0	5,2	780	12	192	0,21
63	0	15,9	2 385	5	185	0,51
80	0	29,8	4 470	7	187	0,75
100	0	97,9	14 685	6	186	1,96

NOTE Particular applications like grinding do not need wheel adapters with the angular orientation of the two different HSK drive keys. Either a HSK with two deep keys or the HSK type E provide rotationally symmetrical shanks without design related residual dynamic unbalances.

[a] See 3.1.9, Figure 5 and A.5.2 for explanation and details regarding couple unbalances.

[b] Dynamic unbalances of HSK-A (with or without data chip hole), in balanced condition according to the provisions of the HSK standard ISO 12164-1 (h_1 = 0 mm) represents the face of the HSK flange.

[c] Angular reference: 0° cutting edge position according to ISO 12164-1 for single-edge tools on the low drive key (driver hub); angle indication of the unbalance position is clockwise with the line of vision from the shaft towards the tool tip. HSK-C: Angles a_{P1} and a_{P2} of all sizes are 0° and 180° due to their rotationally symmetrical collar (see HSK-C25).

[d] Related to the HSK nominal diameters D_S, e.g. HSK-A63 → $D_{P1} = D_{P2} = D_S$ = 63 mm.

[e] HSK-C25.

4.3 Safety-related unbalance limitations (G40) according to ISO 15641

ISO 15641 provides the “G40” safety requirement to be taken into account for rotating tool systems. For safety reasons, the balancing quality of tools with a peripheral speed of more than 1 000 m/min shall not be worse than G40 according to ISO 15641 when applying this document. This means that the static unbalance U_{G40} shall not be exceeded if the peripheral speed, v_{REF}, at the biggest tool diameter, D_{REF} (cutting edge or tool body diameter), is beyond v_{G40} = 1 000 m/min. See Formula (35):

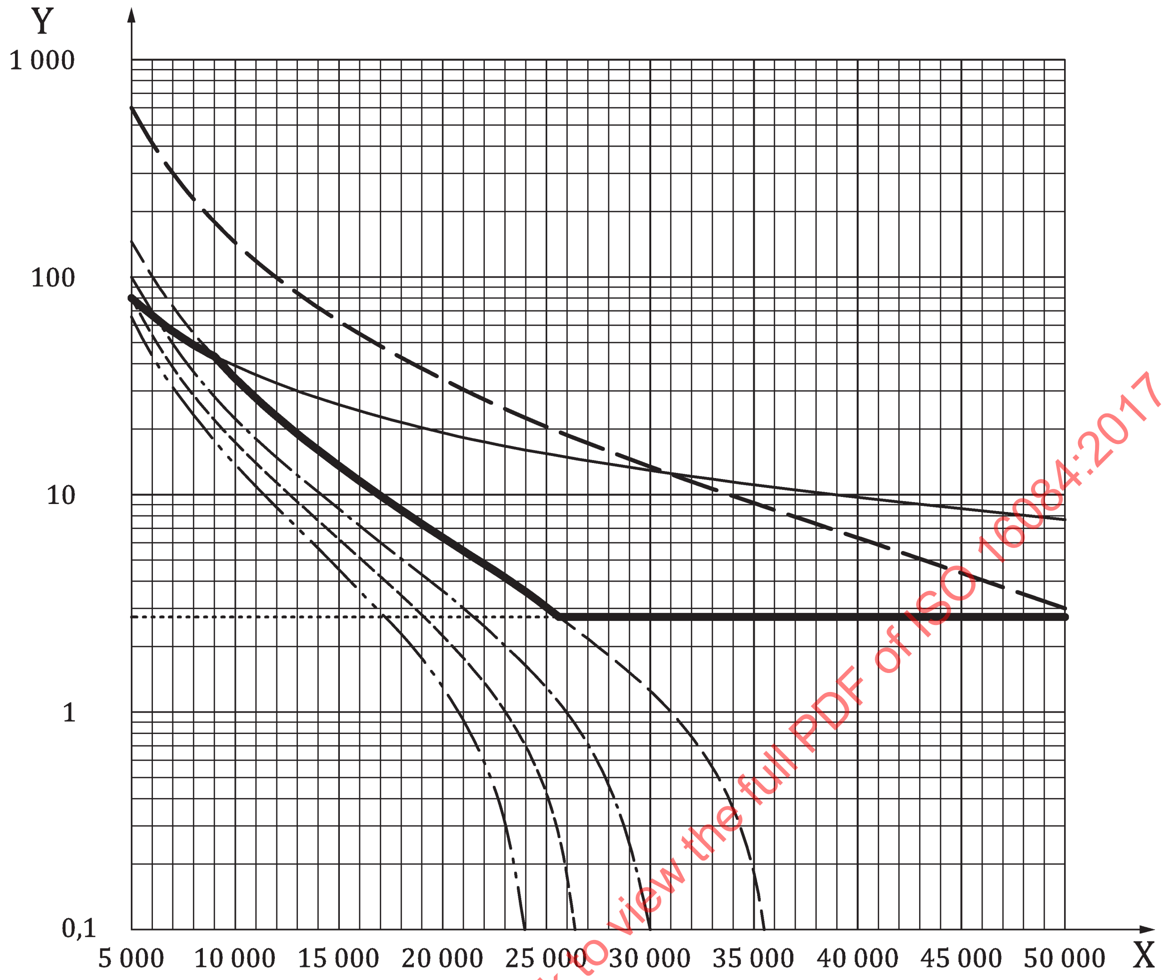
$$U_{G\,40} = \frac{m \times \mathrm{G}\ 40}{\Omega} = m \times \mathrm{G}\ 40 \times \frac{60}{2\pi \times n} \quad (35)$$

U_{G40} shall be compared with the permissible unbalances of standard and/or fine balancing. The smaller value shall be applied.

4.4 Graphic visualization of the balancing requirements

As this document takes a lot of tool, spindle and machining-related parameters into account, the balancing requirements cannot be derived from diagrams (compared to ISO 1940-1). It is easier and less susceptible to faults, as well as more comfortable to calculate the permissible residual unbalances with a computer program using the formulae of this document.

The single logarithmic graphs of the permissible static residual unbalances in Figure 13 visualize the results of the permissible static unbalance and the restrictions for a HSK-A63 tool (m = 1 000 g, L_{CG} = 60 mm), as a single tool and as a component of different tool systems (see Clause 5).



Key

X rotational speed (1/min)

Y permissible static unbalance (gmm)

— —	$U_{STAT,PER,STND}$	▬▬	$U_{STAT,PER,FINE,RES}$
— — —	$U_{STAT,PER,FINE}$	—·—	$U_{STAT,PER,FINE,4}$
——	U_{G40}	– – – – –	$U_{STAT,PER,FINE,5}$
············	U_{MIN}	—··—	$U_{STAT,PER,FINE,6}$

Figure 13 — Permissible static residual unbalances of a HSK-63 tool or component

The curves of standard and fine balancing are all cut at the bottom by the horizontal line of U_{MIN} and may be limited at the top by the U_{G40} graph (see 4.3). The limitation by U_{MIN} (see 4.2.3) results from the accuracy of the balancing machine, $U_{BM,ACC}$, and the unbalance, U_{ECC}, caused by a radial clamping dislocation, e_S. The unbalance, U_{ECC}, has to be calculated according to Formula (10) for each tool mass and its possible radial dislocation (see Table 2).

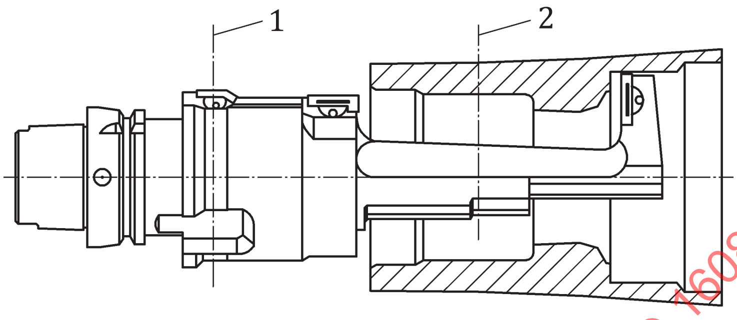
NOTE See A.5.4 for a G40 calculation example.

Residual unbalances below the U_{MIN} lines [see Formula (6)] cannot be achieved process-safely by balancing the tool system only. If this becomes necessary, the complete tool-spindle system would have to be balanced to be able to eliminate the unbalance, U_{ECC}, caused by radial dislocation. In this case, machine tool spindle and tool system are considered monolithic and the unbalance may be reduced to $U_{MIN} \geq U_{BM,MIN}$.

4.5 Special tools with asymmetric body shapes

Manufacturing operations that require asymmetric tool bodies result in major unbalances which sometimes cannot be eliminated.

Figure 14 shows a special tool with an eccentric single point cutting edge combining a two-step forward bore operation with a reverse face cutting operation for machining an enlarged diameter.



Key

1 plane P1

2 plane P2

Figure 14 — Special single tool for standard forward and eccentric reverse machining

The shape of the tool body is based on the radial offset of the tool to enter the bore and does, therefore, not allow a mass compensation opposite to the eccentric cam. Removing sufficient material on the side of the cutting edge is not possible either due to stability reasons. The consequence is that this kind of tools often cannot be balanced to fulfil the requirements of this document.

In order to provide the best conditions for the machining process, as well as for the spindle bearings, it is recommended to

— minimize residual unbalance(s) as good as possible,

— choose the machine with the most rigid spindle bearings,

— double check the bearing load resulting from residual unbalance,

— adapt the machining parameters if the process makes this possible, and

— use damping measures to minimize the unbalance related vibrations.

NOTE In these exceptional cases of special tools with residual unbalance, the main objective is to produce the work piece. Finding the best machining parameters primarily aims at optimizing the machining process but usually also improves the dynamic load rating on the spindle bearings.

5 Balancing of tool systems

5.1 General

Tool systems consist of two or more components such as adapters, prolongations, reductions and cutting tools. The majority of tool systems consist of two or three components that are either all standard components (see Figure 15, G and I) or have at least one standard shank adapter and up to two special components like an intermediate adapter and a single tool (see Figure 15, H and K). Standard single tools like drills and milling cutters, usually clamped in shrink fit holders or collet chucks, are not counting for the total number of components k_{SYS} (see note 1).Therefore, “standard tool systems” are defined as tool systems of three components and constitute the rule for components balancing (e.g. so-called “catalogue components”). See Formula (36):

$$k_{SYS,STND} = 3 \qquad (36)$$

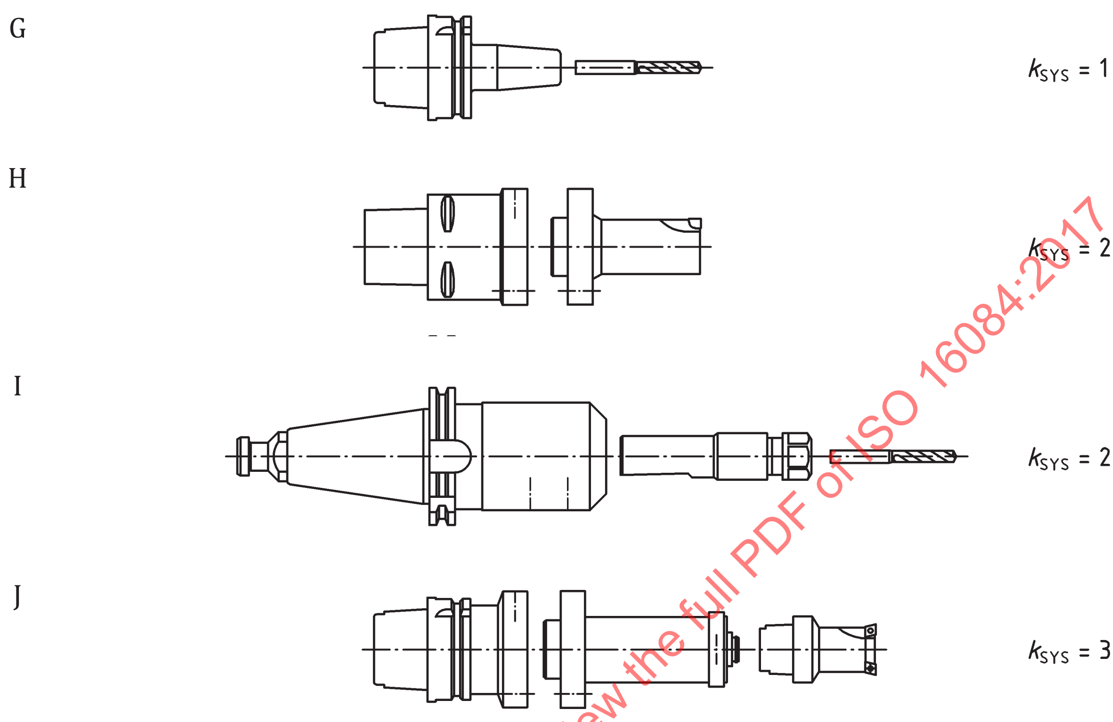


Figure 15 — Examples of common standard tool systems (k_{SYS} ≤ 3)

Tool system "G" shall be seen as a single tool (k_{SYS} = 3). The involved "catalogue" HSK shrink fit adapter, however, is a tool system component and shall be balanced according to Formula (37). This enables tool system “G” to be also used as the “modular single tool” of tool system "J".

Special tool systems are considered assemblies of four or more components. Independent of the number of components, all modular tool systems may be addressed according to the general guidelines of 5.2.

NOTE 1 Predominantly symmetrical cutting tools (e.g. monolithic tools like drills, milling cutters, etc.) with less than 20 % of a tool system mass do not require balancing due to a low influence on the overall balancing condition of the tool spindle system. These tools are therefore not taken into account when counting the number of components (see Figure 15).

The general principle of this document is limiting the unbalance-related dynamic load on the bearings, which depends only on centrifugal forces, to maximally 1 % of the dynamic load rating, C_{DYN} (see 4.2). The tool mass has no influence on the basic unbalance value $U_{STAT,1\ \%}$. $U_{STAT,PER}$, however, is dependent on the tool mass due to the subtraction of U_{MIN} [see Formula (10)]. The possible unbalance, U_{ECC}, is part of U_{MIN} and caused by radial dislocation. Thus, heavier tools create higher centrifugal forces due to a possible radial dislocation and, therefore, have to be better balanced than light tools running at the same rotary speed.

NOTE 2 ISO 1940-1:2003, 6.2.3 allows proportionally more unbalance for increasing tool masses.

It is also essential to note that modular tool systems have the same balancing requirements as monolithic tools of similar design with the same mass and rotational speed. Their tool components, therefore, require a sufficiently better balancing quality to fulfil the balancing requirements of relevant tool systems. So, balancing one and the same tool as a tool system component instead of a single tool reduces $U_{STAT,1\ \%}$ by taking the number of components k_{SYS} into account (see A.4.3 for details).

The graphs of Figure 13 describe this situation for a HSK-63 single tool and for components of a tool system. A fine balanced single tool can run at 24 500 min^{-1} to fulfil the criteria of fine balancing before falling below U_{MIN}. The criteria for a tool system component are the same. A single tool, balanced for standard quality, theoretically enlarges its speed range up to almost 50 000 min^{-1}.

Interface tolerances between modular components may cause additional unbalances of the tool system due to run-outs caused by radial dislocations. Especially radially and angularly adjustable components should therefore be carefully aligned when assembling a tool system.

NOTE 3 The rules of Clause 5 are valid for static and dynamic balancing. Dynamic balancing of tool systems, however, may be complex and requires individual measures because tool systems, composed of statically well-balanced components, have a certain unpredictable dynamic residual unbalance (see 5.10). So, all following rules for components of modular systems are for static balancing, but they can be transferred to dynamic balancing according to 4.2.5 and B.3.

Adaptive components are supposed to be parts of standard tool systems of three components and shall be balanced accordingly. Single tools with spindle suitable shanks (e.g. 7/24 taper or HSK) may have most likely been balanced according to the balancing conditions of single tools and should be double checked before implementing them into a tool system.

NOTE 4 All approaches of Clause 5 and the calculation examples of Annex B take the worst-case scenario into account, i.e. the unbalance vectors of all components have the same orientation.

Varying axial positions of components within a modular tool system also mean variable lever arms of their centres of gravity. Only the first component that is directly clamped into the machine tool spindle keeps its distance, $L_{CG,i}$, from the spindle reference point to its centre of gravity. The lever arms of all other components are being prolonged towards the front end of a tool system (see Figure 16).

Varying lever arms, however, mean different requirements for the permissible unbalance of one and the same component. This would not be practicable as components have to flexibly suit the design of tool systems. Formula (37) solves this issue (see A.4.3 for the detailed derivation).

5.2 Balancing of tool system components

The following aspects have been taken into account.

— As the basic permissible static unbalance $U_{STAT,1\ \%}$ is mass independent [see Formula (3)], the permissible static unbalances of each single component and of the assembled tool system would be the same at the same rotational speed. Because all static component unbalances might, in the “worst case”, sum up to the static unbalance of an assembled tool system (if all unbalances showed into the same direction, compared to 5.4), the number of tool system components is an important criterion.

— The final axial positions of “catalogue components”, especially adapters, in tool systems are unknown. This, however, changes the “active” lever arm of the centre of gravity. The lever is getting bigger if a component is mounted at the second or third position towards the front end of a tool system (see Figure 16). Thus, the effect of a component unbalance within a tool system varies accordingly. Its individual setup, however, should not have an influence on the unbalance of a tool system.

Components that may be placed at any position of a tool system, therefore, require better balancing qualities compared to the location at the first position, i.e. directly clamped in the machine tool spindle, and calculated like a single tool according to Formula (8).

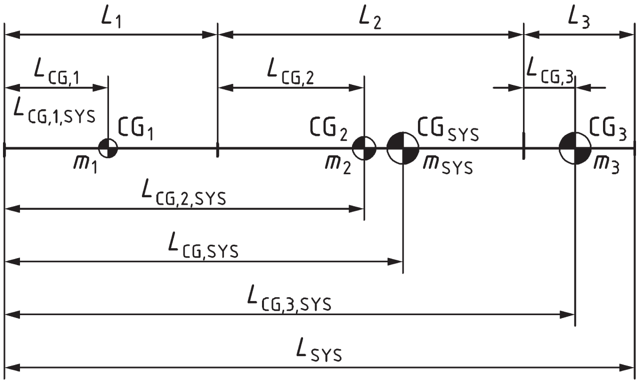


Figure 16 — Centres of gravity, $L_{CG,i,SYS}$, in a standard tool system

The derivation of Formula (37) includes both above criteria (see A.4.3) and leads to the result that all three components of a standard tool system should be balanced to the quality "fine", i.e. $f_{BAL,FINE}$ = 0,2.

The factor $f_{SYS,k}$ enables calculating the permissible unbalance of components for special tool systems of four and more components.

$$U_{STAT,i,SYS,PER} = 9{,}12 \times 10^5 \times f_{BAL,FINE} \times f_{SYS,k} \times \frac{C_{DYN}}{n^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG,i}} \right) - \left(U_{BM,ACC} + m_i \times e_{S,i} \right)$$

(37)

The following points are important for balancing components and tool systems.

— The rotational speed of a tool system, n_{SYS}, should not be bigger than the lowest permissible speed, n_i, of the involved components.

— If components are balanced especially for a certain specific tool system, the rotational speeds, n_i, should not be lower than the speed n_{SYS} of the tool system.

— Well-aligned tool systems composed of fine balanced components will then fulfil the standard quality of static balancing $f_{BAL,STND}$ = 0,8.

Special tool systems are set up by four or more components. Table 4 shows values of $f_{SYS,k}$ for up to six components.

Table 4 — Parameter k_{SYS} for special tool systems of up to 6 components

Tool system components, k_{SYS}	up to 3	4	5	6
$f_{SYS,k}$	1,00	0,70	0,55	0,45

It is always recommended and often necessary to double check the unbalance of tool systems after assembly. Subsequent balancing of an assembled tool system means treating it like a monolithic tool. Any disassembly and re-assembly of such a rebalanced tool system, even in the same configuration, requires a new balancing operation.

NOTE 1 Radial dislocations of components and their angular orientations within tool systems depend on the shank type and the way of assembly. They may sum up towards the front end of a tool system and lead to an increase of the "real" static and/or dynamic unbalance. Non-adjustable shanks as listed in Table 2 show radial dislocations of random angular orientation. It is therefore supposed that radial dislocations of shank interfaces like HSK, taper 7/24, etc. do not sum up in a tool system. Careful assembly minimizes radial dislocation. The possible single radial dislocation of each component is therefore taken into account only once for calculating the permissible static unbalance. Radially and angularly adjustable interfaces can be treated like a HSK.

NOTE 2 See 5.10 for more information about setup and balancing of tool systems.

5.3 Influence of the angular orientation of component unbalances

Residual component unbalances have random angular orientations within modular tool systems, thus leading to compensation to a certain extent. This statistical effect allows a minor increase of the permissible component unbalances, $U_{STAT,PER}$, of Formula (37) without exceeding the permissible unbalance of the tool system, $U_{STAT,SYS,PER}$, calculated with the lever of the system centre of gravity, $L_{CG,SYS}$. The theoretical sum of the permissible component unbalances may be up to 15 % bigger than the permissible unbalance calculated for the assembled tool system itself.

5.4 Influence of clamping dislocations

The assembly of tool system components may incorporate both radial and angular clamping dislocations. Figure 17 shows the different modes and possible combinations.

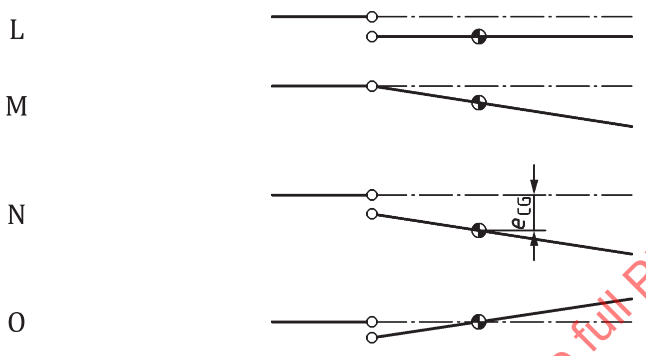


Figure 17 — Modes and combinations of radial and angular dislocation

The extent of these dislocations depends on the shank types and varies randomly within this range.

The influence of angular faults on the unbalance not only depends on the component mass but also on its length, i.e. the axial position of the centre of gravity. Measuring the run-outs of a component at the shank and at the centre of gravity enables to evaluate the static unbalance related to an axial angular misalignment.

Radial clamping inaccuracies of non-adjustable shanks occur randomly regarding their radial angular orientation (see Figure 18, case P) within the range of the possible run-out, $e_{S,i}$, and lead to the unbalances $U_{ECC,i,SYS}$. Cases M, N and O represent scenarios of radial and/or angular clamping dislocations in a tool system.

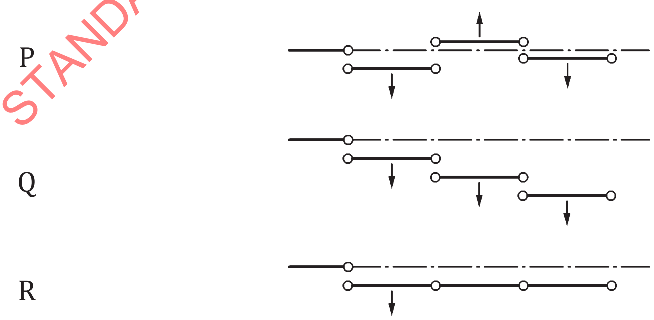


Figure 18 — Radial dislocations: Random (P) or oriented to the same direction (Q, R)

Case Q represents the worst case of radial dislocations because all component unbalances act into the same direction and sum up towards the front end. This can unavoidably happen if cylindrical shanks are radially clamped by screws (e.g. Weldon or Whistle Notch). This case may occur if all clamping screws of several cylindrical shanks were aligned inline.

Case R shows a well-aligned tool system with a radial dislocation of the basic adapter relative to the machine tool spindle. This means that all components are radially dislocated by the same amount, $e_{S,i}$. As this may happen quite commonly in practice, the permissible residual unbalance of tool system components is calculated like single tools by subtracting $U_{ECC,i} = m_i \times e_{S,i}$ in Formula (37). $U_{ECC,i}$ is a part of U_{MIN} [see Formula (10)].

The possible maximum component unbalance, $U_{ECC,\,i,\,MAX}$, within the assembled system depends on its actual radial dislocation, as shown in Formula (38):

$$e_{k,\,SYS,\,MAX} = \sum_{i=1}^{k} e_{S,\,i} \tag{38}$$

and can be calculated to

$$U_{ECC,k,\,MAX} = e_{\,k,\,SYS,\,MAX} \times m_k \tag{39}$$

Formula (39) gives a hint that component unbalances within a tool system due to clamping inaccuracies may be even bigger than the permissible unbalance of Formula (8). It is therefore evident that these radial offsets cannot be pre-compensated by "better balancing". As a consequence, all assembled tool systems should be finally double checked regarding run-out and unbalance.

Whereas it is possible to specify maximum values of pure radial dislocations (see Figure 17, case L) for the most common shanks like HSK and 7/24 taper (see Table 2), real radial and angular faults cannot be specified because their orientations occur randomly (see the exemplary cases M to O of Figure 17).

Adjustable adapters regarding radial and axial angular dislocations enable minimizing both types of run-out by careful assembly.

NOTE Non-adjustable run-outs may also occur between the two sides of one and the same intermediate adapter, e.g. due to manufacturing tolerances between the male and female side of a HSK prolongation.

5.5 Integration of tool system components balanced according to ISO 1940-1

Tool system components that have been balanced according to ISO 1940-1 may also be evaluated according to the above rules in order to be used in tool systems of this document.

The balancing parameters of ISO 1940-1 are the balancing quality G°(x), e.g. G 6,3, the rotational speed *n* and the component mass *m*.

$$U_{G\,(x),\,PER} = \frac{G\,(x) \times m \times 60}{2\pi \times n} \tag{40}$$

This permissible unbalance, $U_{G\,(x),\,PER}$, shall then be compared with the permissible unbalance of this component at its determined location within the assembled tool system according to Formula (37).

5.6 Calculation of the permissible rotational speed depending on actual unbalance

An actual (e.g. measured) static unbalance value, $U_{STAT,\,ACT}$, can also be used for calculating the maximum permissible rotational speed, $n_{\,MAX,\,PER}$, using Formula (41):

$$n_{\,MAX,\,PER} \leq \sqrt{\frac{f_{BAL} \times 9{,}12 \times 10^5 \times C_{DYN}}{U_{STAT,\,ACT} \times \left(\dfrac{L_B + a_M + L_{CG}}{L_B}\right)}} \tag{41}$$

NOTE The unbalance of a radial dislocation is part of the measured unbalance, $U_{STAT,ACT}$. If $U_{STAT,ACT}$ is bigger than $U_{STAT,PER}$, a reduction of the permissible rotational speed according to Formula (41) is the consequence. Leaving the rotational speed at $n_{MAX,PER}$ would lead to a corresponding increase of the spindle load (e.g. a doubled unbalance value $U_{STAT,ACT} = 2 \times U_{STAT,PER}$ leads to a spindle bearing load of 2 % of C_{DYN}). It is up to the user to decide whether a higher spindle load can be exceptionally acceptable for a certain machining period.

5.7 Determination and calculation of the position of the centre of gravity

5.7.1 Experimental determination of the centre of gravity

The position of the centre of gravity, CG, is generally required to calculate the permissible residual unbalances of tools, components and tool systems according to this document. The most convenient way is to indicate the centre of gravity in the drawing as nowadays three-dimensional design systems are quite common.

However, if this information is not available, it can be easily determined before balancing by finding the equilibrium of a component experimentally (see Figure 19).

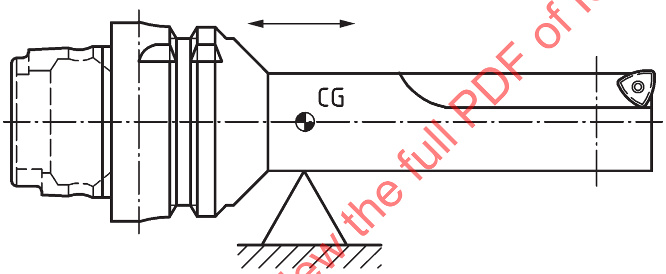


Figure 19 — The equilibrium determines the centre of gravity of a component or a tool system

This empiric method also works for assembled tool systems.

5.7.2 shows the alternative to calculate the position of the tool system centre of gravity from the individual centres of gravity of its components.

5.7.2 Calculation of the centre of gravity of a modular tool system

The permissible static unbalance of an assembled tool system can be calculated with the position of its centre of gravity, $L_{CG,SYS,k}$. Figure 16 shows the relevant parameters of a standard system of three components.

The two formulae of the equilibrium of the CG moments [see Formula (42)]:

$$m_1 \times L_{CG,1} + m_2 \times \left(L_1 + L_{CG,2}\right) + m_3 \times \left(L_1 + L_2 + L_{CG,3}\right) = m_{SYS} \times L_{CG,SYS,3} \qquad (42)$$

and of the mass of the tool system [see Formula (43)]:

$$m_{SYS} = m_1 + m_2 + m_3 \qquad (43)$$

lead to the position of the system centre of gravity, $L_{CG,SYS,3}$, as given by Formula (44):

$$L_{CG,SYS,3} = \frac{m_1 \times L_{CG,1} + m_2 \times \left(L_1 + L_{CG,2}\right) + m_3 \times \left(L_1 + L_2 + L_{CG,3}\right)}{m_1 + m_2 + m_3} \qquad (44)$$

Formula (45) tells the centre of gravity of a tool system of k components.

$$L_{CG,SYS,k} = \frac{m_1 \times L_{CG,1} + m_2 \times \left(L_1 + L_{CG,2}\right) + \cdots + m_{k\text{-}1} \times \left(L_1 + L_2 + \cdots + L_{k-2} + L_{CG,k-1}\right) + m_k \times \left(L_1 + L_2 + \cdots + L_{k\text{-}1} + L_{CG,k}\right)}{m_1 + m_2 + \cdots + m_{k-1} + m_k} \qquad (45)$$

5.8 Balancing of tools and components with alternative interfaces

Intermediate components of tool systems sometimes have other interfaces than listed in Table 2. Examples are shown in Figure 1 and Figure 15. Quite common interfaces are radial or radial and angular adjustable flange connections because they allow minimizing both types of run-outs which is especially important for long components.

These non-listed shanks cannot be individually addressed in a standard. For these cases, Table 2 includes the two parameters, m_{MIN} and m_{MAX}, which are meant to be reference masses helping to classify these components into suitable spindle sizes. The mass figures are overlapping the spindle sizes and in case of doubt, it is recommended to choose the parameters of the smaller spindle size as a precaution.

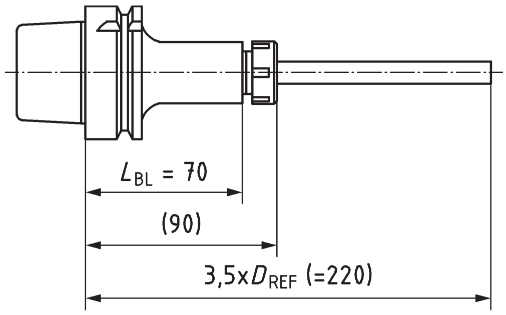
It is further possible to individually balance a tool component for a distinguished spindle size which does not necessarily correspond to the listed reference masses.

5.9 HSK adapters with rotationally symmetrical tools

Figure 20 shows two simple modular HSK tool systems of the same overall length which provide different situations for the possible balancing modes. The two HSK-A63 adapters of different length with collet chucks (or shrink fit holders, hydraulic chucks, etc.) carry the same type of rotationally symmetrical tools (e. g. carbide drills, milling cutter). The overall length of both tool systems is the same.

The shorter 90 mm adapter “S” has a L_{BL} / D_S-ratio of 70/63 = 1,11 < 2,2 with the consequence that static balancing is sufficient. The 180 mm long version “T” shows $R_{L/D} = L_{BL} / D_S = 160/63 = 2{,}5 > 2{,}2$ and, therefore, theoretically fulfils the requirements of dynamic balancing. In practice, both examples, however, are balanced statically at the HSK collar according to the provisions of ISO 12164-1. Their dynamic unbalances are absolutely the same, caused by the statically balanced HSK only (see also Table 3). So, although a longer adapter may enable dynamic balancing, it is not mandatory and/or sensible in any case.

S



T

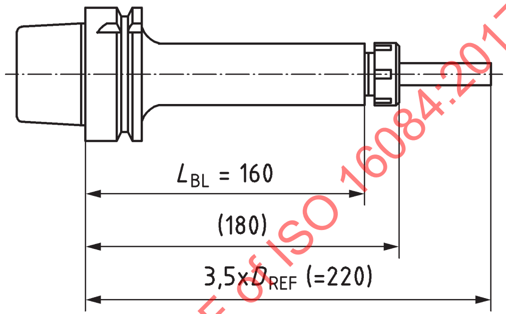


Figure 20 — HSK-A63 collet chuck adapters of different balancing lengths, L_{BL}

5.10 Remarks for setup and balancing of tool systems

a) Design and/or size of many single components often only allow static balancing. The assembly of modular tool systems leads to a vector addition of the static and dynamic component unbalances which either compensates or increases both the static and dynamic unbalance situation. As a consequence, a certain residual dynamic unbalance will remain.

b) Radial and/or angular dislocations due to fit tolerances of the component interfaces are leading to additional static and dynamic unbalances. This can only be prevented with adjustable shanks, e. g. a flange connection. In all other cases (HSK, 7/24 taper, etc.), both radial offset and angular orientation occur randomly.

c) The actually resulting assembly dislocations of components with non-adjustable interfaces cannot be foreseen. A compensation of the maximum possible clamping dislocation (see Table 2) is taken into account in the calculation of the permissible static residual unbalance by subtracting U_{ECC} [see Formula (8) and Formula (10)].

d) Especially if bigger dislocations and/or higher component masses are involved, the unbalance $U_{ECC,MAX}$ may easily exceed the unbalance $U_{STAT,1\%}$ of the dynamic load rating. This even leads to negative permissible unbalances, $U_{STAT,PER}$, especially according to Formula (37). U_{MIN}, however, remains the lowest possible value.

e) Radial clamping dislocations, $e_{S,i}$, may sum up towards the tool front (see Figure 18, case Q), thus increasing the unbalance of both the mounted components and the tool system. As a rule, the unbalance of a tool system with run-outs does not fulfil the requirements of fine balancing but mostly those of standard balancing.

f) Unbalances due to assembly related radial dislocations, however, should be avoided not only from the spindle life point of view but also due to possible negative influences on the work piece quality.

g) Calculating the unbalance of a modular tool system based on permissible residual component unbalances has to take into account that certain run-out unbalances (e.g. due to non-adjustable interfaces) occur despite careful assembly. The real static and dynamic system unbalances finally have to be measured on a balancing machine. Optimizing the run-out situation (if possible) and rebalancing of the assembled tool system are the ultimate possibilities for improvement.

h) The scheduled operating parameters of a tool system describe the minimum demand for the selection of its components. The documentations (catalogue, manual, etc.) of manufacturers and suppliers of modular components have to list all parameters that are required to identify suitable components according to the provisions of this document (see Clause 6).

i) Modular tool systems that are supposed to fulfil the requirements of fine balancing without rebalancing shall be only used up to the rotary speed of their “worst” component.

j) Any sum of component unbalances represents the ideal setup of a tool system without run-out.

k) In case of advanced operating parameters (e.g. high rotational speeds for finishing operations), already, the permissible unbalances of components and as a consequence also of the related tool system tend to fall below U_{MIN} (see 4.2.3). As angular orientations of U_{MIN} depend on randomly varying clamping dislocations making the related unbalance situation of a tool system unpredictable, it is recommended to either use monolithic tools or high precision tool systems with adjustable interfaces.

l) The assembly of balanced single components to a tool system may also require dynamic balancing of the assembled tool system due to its overall length (see 4.2.4). It is then often necessary that previous static balancing measures have to be partially compensated or even reversed. It can be therefore sensible to use unbalanced single components and to balance the assembled modular tool system as a “quasi monolithic” tool.

m) Dynamically balanced joint modular tool systems have to be considered monolithic tools. Components which have been modified by dynamic balancing of a joint tool system have to be rebalanced if applied in a new configuration.

n) Simple tool systems consisting of statically balanced tool adapters (e. g. “catalogue products” such as shrink fit or collet chuck adapters) and unbalanced mainly rotationally symmetrical single tools (e. g. drills or milling cutters) do not require rebalancing. The same applies for the combination of adapters with statically or dynamically balanced special tools.

6 Data representation and exchange

ISO 1940-1 has been well-known in balancing practice for a long time without indication on the tool body. To prevent misunderstandings, balancing according to this document shall be recognizable on the body of tool systems, tools or tool components.

The permissible residual unbalances according to this document depend on too many parameters (shank type and size, rotational speed, dynamic load rating, lever arm of the centre of gravity, single tool or modular tool system, etc.) as if this information could be marked sufficiently on tool bodies.

Consistently, the marking of the tool body shall be

ISO 16084

All required parameters for the calculation of the resulting permissible residual unbalance(s) shall be represented and exchanged via XML data exchange.

In order to create a uniform XML file, Table 5 lists the required elements for the XML structure.

Table 5 — XML-data definition entries

No.	XML symbol	XML symbol description	Equivalent or formula symbols
1	TCM	tool or component mass	m
2	RPM	operating speed	n
3	SZ	spindle size #1 to #9 (see Table 2)	SZ
4	CDYN	dynamic load rating	C_{DYN}
5	ES	radial clamping accuracy	e_S
6	FBAL	balancing quality ($f_{BAL,STND}$ = 0,8 or $f_{BAL,STND}$ = 0,2)	f_{BAL}
7	CCNT	number of components (e.g. single tool k = 1, three parts modular tool k = 3)	k
8	LCG	position of the centre of gravity	L_{CG}
9	LP1	position of the balancing plane	L_{P1}
10	LP2	position of the balancing plane	L_{P2}
11	USTAT	calculated permissible static residual unbalance	$U_{STAT,PER}$
12	UP1	calculated permissible residual unbalance at plane 1	$U_{P1,PER}$
13	UP2	calculated permissible residual unbalance at plane 2	$U_{P2,PER}$

Annex C shows the basic structure of a data exchange file based on the XML technology.

Annex A
(informative)

Permissible residual unbalances — Theoretical approach and calculation examples

A.1 Additional symbols and abbreviated terms for this annex, Annex B and Annex D

Symbols and abbreviated terms	Unit	Description
CG_i	—	Centre of gravity of component *i*
D_C	mm	Maximum cutting edge diameter
$D_{C,SYS}$	mm	Maximum cutting edge diameter of a tool system
D_{P1}	mm	Balancing diameter at plane P1
D_{P2}	mm	Balancing diameter at plane P2
$D_{REF,i}$	mm	Reference diameter of component *i* for the G40 check
$D_{REF,SYS}$	mm	Reference diameter of a tool system for the G40 check[a]
$e_{CG,PER}$	mm	Permissible radial dislocation of the centre of gravity, CG
e_{HSK}	mm	Radial joining dislocation of a HSK interface
$e_{i,SYS}$	mm	Maximum radial dislocation of component *i* within a tool system relative to the tool-spindle rotary axis
$e_{k,SYS}$	mm	Maximum radial dislocation of component *k* within a tool system relative to the tool-spindle rotary axis
$e_{PER,1940\text{-}1}$	mm	Permissible eccentricity of the centre of gravity CG according ISO 1940-1
$e_{S,SYS}$	mm	Radial dislocation of a tool system
f_{CG}	—	General enlarging factor for the rotational speed of tool system components
$f_{CG,i}$	—	Individual enlarging factor for the rotational speed of tool system component *i*
$f_{LIM,C}$	—	Factor for limiting the dynamic unbalances of case C
$f_{RED,i}$	—	Combined factor for deriving the permissible component unbalance
$f_{RED,i,FINE}$	—	Combined factor, based on $f_{BAL,FINE}$
$f_{RED,i,STND}$	—	Combined factor, based on $f_{BAL,STND}$
$f_{RED,AVG}$	—	Average combined factor for deriving the permissible component unbalance
F_U	N	Centrifugal force generated by a rotating unbalance
$F_{U,STAT}$	N	Centrifugal force due to a static unbalance located in the centre of gravity, CG
H_G	mm	Depth of a chip groove
$L_{CG,k}$	mm	Lever arm to the centre of gravity of component *k*
L_G	mm	Length of a chip groove
L_i	mm	Length of component *i*
n_i	min^{-1}	Rotational speed of a tool or a component *i*
$n_{CG,i}$	min^{-1}	Corrected rotational speed for balancing of component *i*, located at a definite position within a tool system
$n_{i,MAX}$	min^{-1}	Maximum rotational speed of component *i*

[a] $D_{REF,SYS}$ can also be a body diameter, not only a cutting diameter.

Symbols and abbreviated terms	Unit	Description
$n_{SYS,MAX}$	min^{-1}	Maximum rotational speed of a tool system
r_{COMP}	mm	Radius to the centre of gravity of a compensation mass
r_U	mm	Radius to the centre of gravity of an unbalance
$R_{DYN,CPL}$	—	Ratio of utilization of C_{DYN} due to a couple unbalance
s	mm	Distance from the front spindle bearing B1 to plane P1
U_{B1}	gmm	Auxiliary unbalance at spindle bearing B1 due to the unbalance of a tool
U_{B2}	gmm	Auxiliary unbalance at spindle bearing B2 due to the unbalance of a tool
$U_{G40,i}$	gmm	G40 safety unbalance of component *i* according to ISO 15641
$U_{G40,SYS,k}$	gmm	G40 safety unbalance of a tool system of *k* components according to ISO 15641
$U_{MIN,i}$	gmm	Achievable minimum unbalance of component *i*
$U_{MIN,i,SYS}$	gmm	Achievable minimum unbalance of component *i* within a tool system
$U_{MOM,B1}$	gmm^2	Moment of unbalance U_{B1} at bearing B1
$U_{MOM,P1,PER}$	gmm^2	Permissible moment of unbalance U_{P1}
$U_{MOM,P2,PER}$	gmm^2	Permissible moment of unbalance U_{P2}
$U_{MOM,STAT,PER}$	gmm^2	Permissible moment of the static unbalance U_{STAT} (located at CG)
$U_{P1,PER,LIM}$	gmm	Limited permissible residual unbalance at balancing plane P1 (case C)
$U_{P1,PER,FINE}$	gmm	$U_{P1,PER}$ for fine balancing
$U_{P1,PER,STND}$	gmm	$U_{P1,PER}$ for standard balancing
$U_{P1,PER,LIM,FINE}$	gmm	$U_{P1,PER,LIM}$ for fine balancing
$U_{P1,PER,LIM,STND}$	gmm	$U_{P1,PER,LIM}$ for standard balancing
$U_{P2,PER,FINE}$	gmm	$U_{P2,PER}$ for fine balancing
$U_{P2,PER,STND}$	gmm	$U_{P2,PER}$ for standard balancing
$U_{P2,PER,LIM}$	gmm	Limited permissible residual unbalance at balancing plane P2 (case C)
$U_{P2,PER,LIM,FINE}$	gmm	$U_{P2,PER,LIM}$ for fine balancing
$U_{P2,PER,LIM,STND}$	gmm	$U_{P2,PER,LIM}$ for standard balancing
$U_{STAT,i}$	gmm	Static unbalance of component *i*, taking the total number of tool system components k_{SYS} into account
$U_{STAT,i,SYS,CG}$	gmm	Permissible residual static unbalance of component *i* referring to the axial position of its centre of gravity CG within a tool system
$U_{STAT,SYS,MAX}$	gmm	Maximum static unbalance of an assembled tool system
$U_{STAT,SYS,PER,k}$	gmm	Permissible residual static unbalance of an assembled (quasi monolithic) tool system of *k* fine balanced components
$U_{STAT,SYS,PER,SUM}$	gmm	Sum of the permissible residual static component unbalances (fine balancing) of a tool system
$U_{STAT,SYS,PER,3}$	gmm	Permissible residual static unbalance of an assembled (quasi monolithic) standard tool system of 3 fine balanced components
W_G	mm	Width of a chip groove
$v_{REF,i}$	m/min	Reference peripheral speed of the biggest diameter of component *i*
[a] $D_{REF,SYS}$ can also be a body diameter, not only a cutting diameter.		

A.2 Principles

Centrifugal loads that increase square with the rotational speed primarily act on the machine tool spindle bearings. Loads generated by unbalances shall therefore be limited, dependent on the operational speed so that a reduction of the spindle bearing life shall be excluded.

The effects of tool system unbalances on machine tool spindles of different types and sizes could, if at all, be only determined by long-term tests and experimental investigations. Therefore, it was necessary to introduce a mathematical model which enabled a qualitative and quantitative determination of the unbalance effects based on general mechanical principles.

This approach is based on two considerations.

a) All relevant rolling bearing spindles of machine tools show comparable mechanical properties.

b) A generalized mechanical model of the mechanical structure reflects the dynamic properties of spindle systems due to mechanical similarities.

This mechanical model represents spindles with the interfaces HSK-25 to HSK-100 and shanks of similar size and mechanical quality.

By means of this generalized mechanical structure (see Figure 2), the speed and unbalance dependent centrifugal forces on the front spindle bearing are being calculated. The total dynamic load, $F_{B1,RES}$, of the spindle bearing B1 is compared for evaluation with the dynamic load ratings, C_{DYN}, of bearings typical for the different spindle series.

A ratio of F_B / C_{DYN} (ratio of total bearing load to dynamic load rating) over 10 % due to tool mass, machining forces and centrifugal forces due to unbalance is considered a high load to bearings.

This document allows a dynamic load on the front bearing(s) caused by unbalance forces of not more than one percent of the dynamic load rating, i.e. $F_{B1,RES} / C_{DYN} \leq 0{,}01$. For the derivation of this approach, as well as for the generalized spindle parameters, see A.3 and ISO 1940-1:2003, 6.4.1.

A.3 Approach

The balancing requirements for the tool-spindle system have to limit the unbalance-related spindle load. Similar mechanical structures of spindle systems with common roller bearings allow the mathematical derivation according to generalized mechanical models.

Particularly, the front spindle bearing(s) is (are) exposed most to the tool unbalance-related forces. Figure A.1 shows the parameters for the calculation of the spindle bearing loads due to unbalance forces.

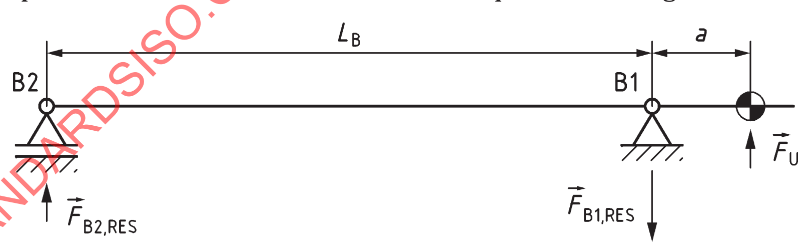


Figure A.1 — Two-span-beam for the load calculation of the front bearing

The variable parameters are the lever arm of the unbalance a [see Formula (4)], the distance L_B between the centres of the front and the rear spindle bearings, the rotational speed, n, and the unbalance, U. Relevant values of interface specific parameters are given in Table 2.

For each size of the four listed interfaces the typical dynamic load ratings of relevant bearings (C_{DYN}), minimum and maximum reference component masses (m_{MIN} and m_{MAX}), joining accuracies of the interfaces (e_S) and reproducibly measurable residual unbalances (U_{BM}) are additionally defined.

The unbalance force, $F_{U,STAT}$ (i.e. a centrifugal force), generated by the rotating unbalance U

$$F_{U,STAT} = U_{STAT} \times \Omega^2 = U_{STAT} \times \left(\frac{2\pi \times n}{60} \right)^2 \tag{A.1}$$

is converted by means of the leverage ratios (see Figure A.1) into the bearing force, $F_{B1,RES}$, as given by Formula (A.2):

$$\left| F_{B1,RES} \right| = F_{U,STAT} \times \left(1 + \frac{a}{L_B} \right) \tag{A.2}$$

and $F_{B2,RES}$, as given by Formula (A.3):

$$\left| F_{B2,RES} \right| = F_{U,STAT} \times \frac{a}{L_B} \tag{A.3}$$

The balancing relevant bearing force, $F_{B1,RES}$, is related to the dynamic load rating, C_{DYN}, of bearing B1, as shown by Formula (A.4):

$$F_{B1} \leq F_{B1,PER} = 0{,}01 \times C_{DYN} \tag{A.4}$$

By additionally taking the measuring limits of the balancing machine (U_{BM}) and the influence of the clamping inaccuracy (U_{ECC}) into account, Formula (A.1) can be completed to Formula (A.5):

$$U_{STAT,PER} \leq \left[\frac{9{,}12 \times 10^5 \times C_{DYN}}{n^2 \times \left(1 + \frac{a}{L_B} \right)} \right] - U_{BM,ACC} - U_{ECC} \tag{A.5}$$

with Formula (4):

$$a = a_M + L_{CG}$$

A.4 Conversion of the permissible static residual unbalance into permissible dynamic unbalances of two planes

A.4.1 Case D: Tool centre of gravity CG between planes P1 and P2 (P1-CG-P2)

All unbalance vectors $U_{P1,PER}$, $U_{P2,PER}$ and $U_{STAT,PER}$ have the same angular orientation.

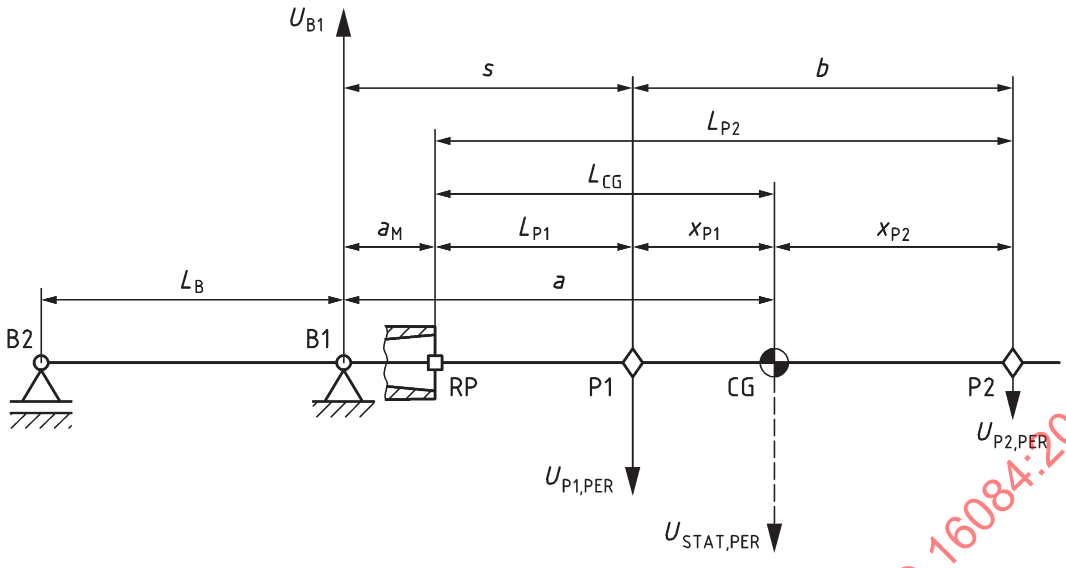


Figure A.2 — Tool centre of gravity CG between planes P1 and P2 (case D ≡ P1-CG-P2)

The formulae of equilibrium for the unbalance moments U_{MOM} around B2 are:

for the static unbalance

$$U_{\mathrm{MOM,STAT,PER}} + U_{\mathrm{MOM,B1}} = 0 \tag{A.6}$$

and for the dynamic unbalance:

$$U_{\mathrm{MOM,P1,PER}} + U_{\mathrm{MOM,P2,PER}} + U_{\mathrm{MOM,B1}} = 0 \tag{A.7}$$

Together with the parameters of Figure A.2, Formula (A.6) and Formula (A.7) lead to Formula (A.8) and Formula (A.9):

$$U_{\mathrm{STAT,PER}} \times (a + L_{\mathrm{B}}) + U_{\mathrm{B1}} \times L_{\mathrm{B}} = 0 \tag{A.8}$$

$$U_{\mathrm{P1,PER}} \times (L_{\mathrm{B}} + s) + U_{\mathrm{P2,PER}} \times (L_{\mathrm{B}} + s + b) + U_{\mathrm{B1}} \times L_{\mathrm{B}} = 0 \tag{A.9}$$

Subtraction of Formula (A.8) from Formula (A.9) eliminates $U_{\mathrm{B1}} \times L_{\mathrm{B}}$, as shown in Formula (A.10):

$$U_{\mathrm{P1,PER}} \times (L_{\mathrm{B}} + s) + U_{\mathrm{P2,PER}} \times (L_{\mathrm{B}} + s + b) - U_{\mathrm{STAT,PER}} \times (a + L_{\mathrm{B}}) = 0 \tag{A.10}$$

The formulae of equilibrium regarding the unbalances are:

for the static unbalance

$$U_{\mathrm{STAT,PER}} + U_{\mathrm{B1}} + U_{\mathrm{B2}} = 0 \tag{A.11}$$

for the dynamic unbalance

$$U_{\mathrm{P1,PER}} + U_{\mathrm{P2,PER}} + U_{\mathrm{B1}} + U_{\mathrm{B2}} = 0 \tag{A.12}$$

Subtraction of Formula (A.11) from Formula (A.12) eliminates U_{B1} and U_{B2}:

$$U_{\mathrm{P2,PER}} = U_{\mathrm{STAT,PER}} - U_{\mathrm{P1,PER}} \tag{A.13}$$

Entering Formula (A.13) into Formula (A.10) leads to Formula (A.14):

$$U_{\mathrm{P1,PER}} \times (L_{\mathrm{B}} + s) + (U_{\mathrm{STAT,PER}} - U_{\mathrm{P1,PER}}) \times (L_{\mathrm{B}} + s + b) - U_{\mathrm{STAT,PER}} \times (a + \mathrm{L}_{\mathrm{B}}) = 0 \tag{A.14}$$

$$U_{\mathrm{P1,PER}} \times (L_{\mathrm{B}} + s) - U_{\mathrm{P1,PER}} \times (L_{\mathrm{B}} + s + b) + U_{\mathrm{STAT,PER}} \times (L_{\mathrm{B}} + s + b) - U_{\mathrm{STAT,PER}} \times (a + L_{\mathrm{B}}) = 0 \tag{A.15}$$

$$U_{\mathrm{P1,PER}} = U_{\mathrm{STAT,PER}} \times \frac{s + b - a}{b} \tag{A.16}$$

Entering Formula (A.16) into Formula (A.13) results in Formula (A.17) and Formula (A.18):

$$U_{\mathrm{P2,PER}} = U_{\mathrm{STAT,PER}} \times \left(1 - \frac{s + b - a}{b}\right) \tag{A.17}$$

$$U_{\mathrm{P2,PER}} = U_{\mathrm{STAT,PER}} \times \frac{a - s}{b} \tag{A.18}$$

with Formulae (4) and (A.19):

$$a = a_{\mathrm{M}} + L_{\mathrm{CG}}$$

$$s = a_{\mathrm{M}} + L_{\mathrm{P1}} \tag{A.19}$$

and the correlations for the centre of gravity, as shown in Formulae (A.20) to (A.22):

$$x_{\mathrm{P1}} = a - s \tag{A.20}$$

$$x_{\mathrm{P2}} = s + b - a \tag{A.21}$$

$$b = x_{\mathrm{P1}} + x_{\mathrm{P2}} \tag{A.22}$$

Formula (A.16) and Formula (A.18) change into

$$U_{\mathrm{P1,PER}} = U_{\mathrm{STAT,PER}} \times \frac{x_{\mathrm{P2}}}{x_{\mathrm{P1}} + x_{\mathrm{P2}}} \tag{A.23}$$

$$U_{P2,PER} = U_{STAT,PER} \times \frac{x_{P1}}{x_{P1} + x_{P2}} \quad (A.24)$$

Referring Formula (A.23) and Formula (A.24) to the reference point, RP (instead to the gravity centre), with Formula (A.25) and Formula (A.26):

$$x_{P1} = L_{CG} - L_{P1} \quad (A.25)$$

$$x_{P2} = L_{P2} - L_{CG} \quad (A.26)$$

turns out to

$$U_{P1,PER} = U_{STAT,PER} \times \frac{L_{P2} - L_{CG}}{L_{P2} - L_{P1}} \quad (A.27)$$

$$U_{P2,PER} = U_{STAT,PER} \times \frac{L_{CG} - L_{P1}}{L_{P2} - L_{P1}} \quad (A.28)$$

NOTE 1 Formula (A.23) and Formula (A.24) or Formula (A.27) and Formula (A.28) can also be derived from the equilibrium of unbalances [see Formula (A.13)] and the equilibrium of the static and the dynamic unbalances [see Formula (A.29)]. Formula (A.29) sets the moments of the permissible dynamic unbalances equal to the moment of the permissible static unbalance.

$$U_{P1,PER} \times s + U_{P2,PER} \times (s + b) = U_{STAT,PER} \times a \quad (A.29)$$

The derivations of the following special cases E and F (see A.4.2.2 and A.4.2.3) are based on the easier approach according to Formula (A.13) and Formula (A.29).

NOTE 2 Plausibility check for static balancing: The position of the centre of gravity of static balancing is defined to be in plane P1 which means $x_{P1} = 0$. This means for Formula (A.24) that $U_{P2,PER} = 0$. As x_{P2} is in the denominator of the quotient, the plausibility check of Formula (A.23) needs to determine the limit value for $x_{P2} \rightarrow 0$. This limit value of the quotient x_{P2}/x_{P2} is "1" and so the result of Formula (A.23) $U_{P1,PER} = U_{STAT,PER}$ is correct.

A.4.2 Tool centre of gravity CG not between planes P1 and P2

A.4.2.1 General

The following two special cases E and F require an appropriate ratio of x_{P1} and x_{P2} in addition to the derivations of the standard case D (see A.4.1).

A.4.2.2 Case E: Tool centre of gravity CG between spindle reference point RP and plane P1 (CG-P1-P2)

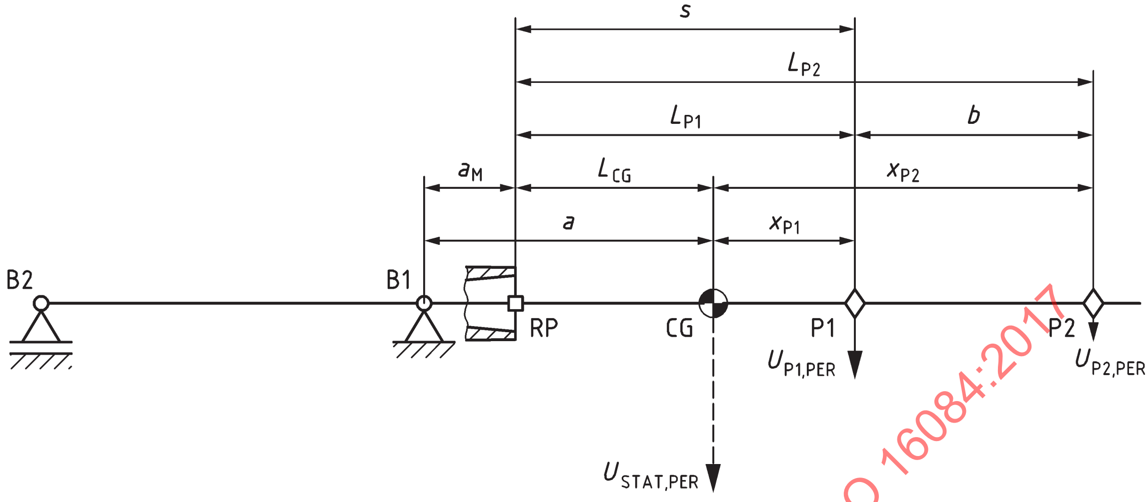


Figure A.3 — Tool centre of gravity CG between reference point RP and plane P1 (case E ≡ CG-P1-P2)

The rule of case E to split up the permissible unbalances:

$$\frac{U_{P1,PER}}{U_{P2,PER}} = \frac{x_{P2}}{x_{P1}} \tag{A.30}$$

is leading to

$$U_{P1,PER} = \frac{x_{P2}}{x_{P1}} \times U_{P2,PER} \tag{A.31}$$

Formula (A.31) into Formula (A.29), the formula of the required equilibrium of static and dynamic unbalance moments [given by Formula (A.32)]:

$$\left(\frac{x_{P2}}{x_{P1}} \times U_{P2,PER}\right) \times s + U_{P2,PER} \times (s+b) = U_{STAT,PER} \times a \tag{A.32}$$

results in

$$U_{P2,PER} = U_{STAT,PER} \times \frac{a}{s \times \left(\frac{x_{P2}}{x_{P1}} + 1\right) + b} \tag{A.33}$$

Formula (A.33) into Formula (A.31) is

$$U_{P1,PER} = U_{STAT,PER} \times \frac{\frac{x_{P2}}{x_{P1}} \times a}{s \times \left(\frac{x_{P2}}{x_{P1}} + 1\right) + b} \tag{A.34}$$

Referring to CG, the parameters b and s in Formula (A.33) and Formula (A.34) can be substituted by Formula (A.35) and Formula (A.36):

$$b = x_{P2} - x_{P1} \tag{A.35}$$

$$s = a + x_{P1} \tag{A.36}$$

which finally results in

$$U_{P1,PER} = U_{STAT,PER} \times \frac{a \times x_{P2}}{a \times (x_{P1} + x_{P2}) + 2 \times x_{P1} \times x_{P2}} \tag{A.37}$$

$$U_{P2,PER} = U_{STAT,PER} \times \frac{a \times x_{P1}}{a \times (x_{P1} + x_{P2}) + 2 \times x_{P1} \times x_{P2}} \tag{A.38}$$

Focusing on the reference point, RP, Formula (A.37) and Formula (A.38) can be reformulated with Formula (A.39) and Formula (A.40):

$$x_{P1} = L_{P1} - L_{CG} \tag{A.39}$$

$$x_{P2} = L_{P2} - L_{CG} \tag{A.40}$$

and Formula (4):

$$a = a_M + L_{CG})$$

to Formulae (A.41) and (A.42):

$$U_{P1,PER} = U_{STAT,PER} \times \frac{(a_M + L_{CG}) \times (L_{P2} - L_{CG})}{(a_M + L_{CG}) \times (L_{P1} + L_{P2} - 2 \times L_{CG}) + 2 \times (L_{P1} - L_{CG}) \times (L_{P2} - L_{CG})} \tag{A.41}$$

$$U_{P2,PER} = U_{STAT,PER} \times \frac{(a_M + L_{CG}) \times (L_{P1} - L_{CG})}{(a_M + L_{CG}) \times (L_{P1} + L_{P2} - 2 \times L_{CG}) + 2 \times (L_{P1} - L_{CG}) \times (L_{P2} - L_{CG})} \tag{A.42}$$

A.4.2.3 Case F: Tool centre of gravity, CG, between plane P2 and the tool front end (P1-P2-CG)

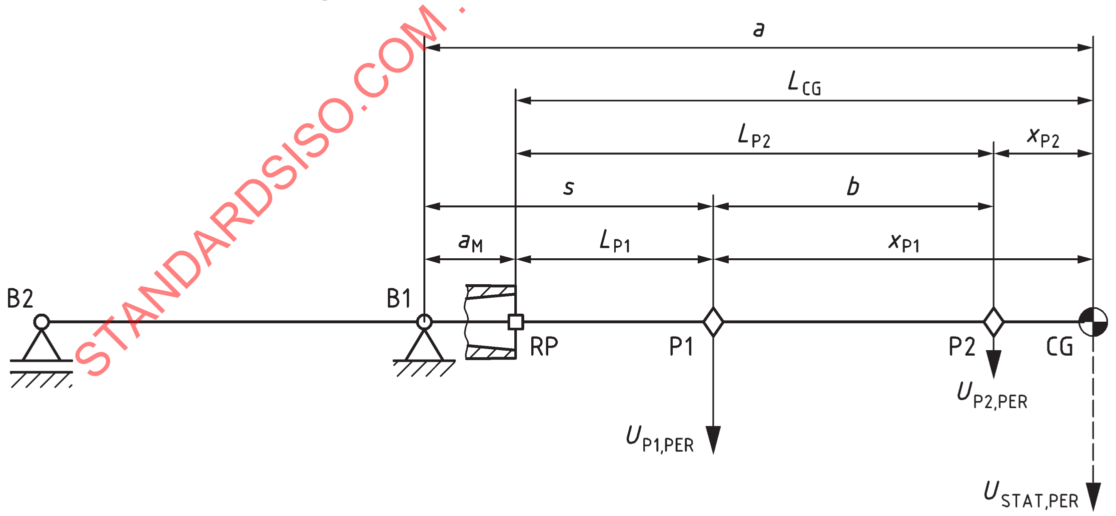


Figure A.4 — Tool centre of gravity, CG, between plane P2 and tool front end (case F ≡ P1-P2-CG)

The split of the permissible unbalances of planes P1 and P2 of case F follows the rule as given by Formula (A.43):

$$\frac{U_{\text{P1,PER}}}{U_{\text{P2,PER}}}=\frac{x_{\text{P1}}}{x_{\text{P2}}} \tag{A.43}$$

i.e.

$$U_{\text{P1,PER}}=U_{\text{P2,PER}}\times\frac{x_{\text{P1}}}{x_{\text{P2}}} \tag{A.44}$$

Formula (A.44) in Formula (A.29) gives Formulae (A.45) and (A.46):

$$U_{\text{P2,PER}}\times\frac{x_{\text{P1}}}{x_{\text{P2}}}\times s+U_{\text{P2,PER}}\times(s+b)=U_{\text{STAT,PER}}\times a \tag{A.45}$$

$$U_{\text{P2,PER}}\times\left(\frac{x_{\text{P1}}}{x_{\text{P2}}}\times s+(s+b)\right)=U_{\text{STAT,PER}}\times a \tag{A.46}$$

Replacing the parameters *b* and *s* of Formula (A.46) by Formula (A.47) and Formula (A.48):

$$b=x_{\text{P1}}-x_{\text{P2}} \tag{A.47}$$

$$s=a-x_{\text{P1}} \tag{A.48}$$

results in

$$U_{\text{P2,PER}}=U_{\text{STAT,PER}}\times\frac{a\times x_{\text{P2}}}{a\times\left(x_{\text{P1}}+x_{\text{P2}}\right)-x_{\text{P1}}^{2}-x_{\text{P2}}^{2}} \tag{A.49}$$

Formula (A.49) in Formula (A.44) gives

$$U_{\text{P1,PER}}=U_{\text{STAT,PER}}\times\frac{a\times x_{\text{P1}}}{a\times\left(x_{\text{P1}}+x_{\text{P2}}\right)-x_{\text{P1}}^{2}-x_{\text{P2}}^{2}} \tag{A.50}$$

The sum of the unbalances $U_{\text{P1,PER}}$ and $U_{\text{P2,PER}}$ calculated by Formula (A.49) and Formula (A.50), however, is now bigger than the permissible static unbalance $U_{\text{STAT,PER}}$. The reason is that the lever arms of the dynamic unbalances are both smaller than the lever of the static unbalance located in the centre of gravity. Formula (A.51) applies:

$$U_{\text{P1,PER}}+U_{\text{P2,PER}}>U_{\text{STAT,PER}} \tag{A.51}$$

Therefore, a limitation of $U_{\text{P1,PER}}$ and $U_{\text{P2,PER}}$ is getting necessary to lead again to Formula (A.52):

$$U_{\text{P1,PER,LIM}}+U_{\text{P2,PER,LIM}}=U_{\text{STAT,PER}} \tag{A.52}$$

$U_{\text{P1,PER,LIM}}$ and $U_{\text{P2,PER, LIM}}$ are calculated with the factor $f_{\text{LIM, C}}$ to

$$U_{\text{P1,PER,LIM}}=f_{\text{LIM,C}}\times U_{\text{P1,PER}} \tag{A.53}$$

$$U_{P2,PER,LIM} = f_{LIM,C} \times U_{P2,PER} \tag{A.54}$$

Formula (A.53) and Formula (A.54) in Formula (A.52) gives

$$f_{LIM,C} \times \left(U_{P1,PER} + U_{P2,PER} \right) = U_{STAT,PER} \tag{A.55}$$

Formula (A.49) and Formula (A.50) into Formula (A.55) results to Formula (A.56) and Formula (A.57):

$$f_{LIM,C} \times U_{STAT,PER} \times \left(\frac{a \times x_{P1} + a \times x_{P2}}{a \times \left(x_{P1} + x_{P2} \right) - x_{P1}^2 - x_{P2}^2} \right) = U_{STAT,PER} \tag{A.56}$$

$$f_{LIM,C} = \frac{a \times \left(x_{P1} + x_{P2} \right) - x_{P1}^2 - x_{P2}^2}{a \times \left(x_{P1} + x_{P2} \right)} \tag{A.57}$$

Formula (A.49), Formula (A.50) and Formula (A.57) in Formula (A.53) and Formula (A.54) finally results in the modified formulae referring to CG.

$$U_{P1,PER,LIM} = U_{STAT,PER} \times \frac{x_{P1}}{x_{P1} + x_{P2}} \tag{A.58}$$

$$U_{P2,PER,LIM} = U_{STAT,PER} \times \frac{x_{P2}}{x_{P1} + x_{P2}} \tag{A.59}$$

Now, the sum of the two permissible dynamic unbalances $U_{P1,PER,LIM}$ and $U_{P2,PER,\ LIM}$ is equal to the permissible static unbalance $U_{STAT,PER}$.

Referring Formula (A.58) and Formula (A.59) to the reference point, RP, can be done by substituting x_{P1} and x_{P2} by using Formula (A.60) and Formula (A.61):

$$x_{P1} = L_{CG} - L_{P1} \tag{A.60}$$

$$x_{P2} = L_{CG} - L_{P2} \tag{A.61}$$

So, the formulae referring to RP are Formula (A.62) and Formula (A.63):

$$U_{P1,PER,LIM} = U_{STAT,PER} \times \frac{L_{CG} - L_{P1}}{2 \times L_{CG} - \left(L_{P1} + L_{P2} \right)} \tag{A.62}$$

$$U_{P2,PER,LIM} = U_{STAT,PER} \times \frac{L_{CG} - L_{P2}}{2 \times L_{CG} - \left(L_{P1} + L_{P2} \right)} \tag{A.63}$$

A.4.3 Derivation of the “balancing of tool system components”

NOTE 1 This subclause explains the balancing of tool system components generally to the balancing quality “fine” f_{BAL} = 0,2. Some parameters of A.1 are introduced only for the derivation and not required for the application of this document according to 5.2.

The static unbalances of all assembled components may sum up to the total static unbalance of a tool system.

A consistent balancing quality, even for the worst case when all component unbalances have the same angular orientation, can be achieved if the sum of all component unbalances, $U_{STAT,i,PER}$, is not higher than the permissible residual unbalance $U_{STAT,SYS,PER}$ of the assembled tool system at its rotational operating speed.

This is only possible if the number of tool system components k_{SYS} is taken into account when calculating the permissible unbalances $U_{STAT,i}$ with Formula (8). The number of components, k_{SYS}, modifies Formula (8) for a single component *i* to Formula (A.64):

$$U_{STAT,i} = 9{,}12 \times 10^5 \times \frac{f_{BAL}}{k_{SYS}} \times \frac{C_{DYN}}{n_i^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG,i}} \right) - \left(U_{BM,ACC} + m_i \times e_{S,i} \right) \quad (A.64)$$

Different axial component positions within a tool system, however, change the "active" lever arm of the centre of gravity (see Figure 16) – the further away from the spindle, the bigger the lever effect of the unbalance-related centrifugal forces on the bearings. Reduced permissible residual unbalances of components can take this into account.

Virtually increased rotational speeds lead to this reduction of the permissible unbalances. The factor $f_{CG,\,i}$ (for derivation, see below) is limiting the component unbalances and subsequently their load on the spindle bearing(s).

The virtually increased speed, $n_{CG,\,i}$, and the real axial position of the component centre of gravity within the tool system, $L_{CG,i,SYS}$, modify Formula (A.64) of the permissible unbalance of this component *i* to Formula (A.65):

$$U_{STAT,i,SYS,CG} = 9{,}12 \times 10^5 \times \frac{f_{BAL}}{k_{SYS}} \times \frac{C_{DYN}}{n_{CG,i}^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG,i,SYS}} \right) - \left(U_{BM,ACC} + m_i \times e_{S,i} \right) \quad (A.65)$$

The virtual rotational speed, $n_{CG,\,i}$, of a component is calculated to Formula (A.66):

$$n_{CG,i} = f_{CG,i} \times n_i \quad (A.66)$$

So, Formula (A.65) changes to Formula (A.67):

$$U_{STAT,i,SYS,CG} = 9{,}12 \times 10^5 \times \frac{f_{BAL}}{k_{SYS}} \times \frac{C_{DYN}}{\left(f_{CG,i} \times n_i \right)^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG,i,SYS}} \right) -$$
$$\left(U_{BM,ACC} + m_i \times e_{i,SYS} \right) = 9{,}12 \times 10^5 \times \frac{f_{BAL}}{k_{SYS} \times f_{CG,i}^2} \times \frac{C_{DYN}}{n_i^2} \times \quad (A.67)$$
$$\left(\frac{L_B}{L_B + a_M + L_{CG,i,SYS}} \right) - \left(U_{BM,ACC} + m_i \times e_{i,SYS} \right)$$

An equal load on the spindle bearing(s) will be achieved if the unbalances of Formula (A.64) and Formula (A.67) are equal, as in Formula (A.68).

$$U_{STAT,i} = U_{STAT,i,SYS,CG} \quad (A.68)$$

i.e. (Formula A.69)

$$\left(\frac{1}{L_B + a_M + L_{CG,i}} \right) = \frac{1}{f_{CG,i}^2} \times \left(\frac{1}{L_B + a_M + L_{CG,i,SYS}} \right) \quad (A.69)$$

Finally, the rotary speed enlarging factor, $f_{CG,\,I}$, results in Formula (A.70):

$$f_{CG,i} = \sqrt{\frac{L_B + a_M + L_{CG,i,SYS}}{L_B + a_M + L_{CG,i}}} \qquad (A.70)$$

The position of the centre of gravity of a certain component k within a tool system (see Figure 16) is calculated to Formula (A.71):

$$L_{CG,k,SYS} = \sum_{i=1}^{k} L_{i-1} + L_{CG,k} \qquad (A.71)$$

An exemplary representative standard tool system of three-components is assumed for the following calculations. It is composed of a shank adapter, an intermediate adapter and a front end component (cutting tool). Two different realistic tool system setups are calculated below in order to find a simple general procedure.

All components of the below representative HSK-A63 tool system example have the same mass m_i, length dimensions L_i, individual positions of the centres of gravity $L_{CG,i}$ and possible radial dislocations $e_{S,i}$ (compared with 4.4).

$$L_1 = L_2 = L_3 = 120\ \text{mm}$$

$$L_{CG,1} = L_{CG,2} = L_{CG,3} = 60\ \text{mm}$$

$$m_1 = m_2 = m_3 = 1\,000\ \text{g}$$

$$e_{S,1} = e_{S,2} = e_{S,3} = e_{S,SYS} = 0{,}002\ \text{mm}$$

$$n_1 = n_2 = n_3 = n_{SYS} = 12\,000\ \text{min}^{-1}$$

The centres of gravity of the three components in the tool system according to Formula (A.71) are

$$L_{CG,1,SYS} = \sum_{i=1}^{k=1} L_{i-1} + L_{CG,k} = (L_0 +) L_{CG,1} = (0+) 60\ \text{mm} = 60\ \text{mm} \qquad (A.72)$$

$$L_{CG,2,SYS} = \sum_{i=1}^{k=2} L_{i-1} + L_{CG,k} = (L_0 +) L_1 + L_{CG,2} = (0+) 120 + 60\ \text{mm} = 180\ \text{mm} \qquad (A.73)$$

$$L_{CG,3,SYS} = \sum_{i=1}^{k=3} L_{i-1} + L_{CG,k} = (L_0 +) L_1 + L_2 + L_{CG,3} = (0+) 120 + 120 + 60\ \text{mm} = 300\ \text{mm} \qquad (A.74)$$

The centre of gravity of the assembled tool system is calculated according to Formula (44):

$$L_{CG,SYS,3} = \frac{m_1 \times L_{CG,1} + m_2 \times (L_1 + L_{CG,2}) + m_3 \times (L_1 + L_2 + L_{CG,3})}{m_1 + m_2 + m_3} = 180\ \text{mm}$$

The centres of gravity, $L_{CG,i,SYS}$, of Formula (A.72) to Formula (A.74), together with the HSK-63 spindle parameters L_B and a_M (see Table 2), lead to the following speed enlarging factors, $f_{CG,i}$, [see Formulae (A.75) to (A.77)], according to Formula (A.70):

$$f_{CG,1} = 1{,}0 \qquad (A.75)$$

$$f_{CG,2} = 1{,}11 \tag{A.76}$$

$$f_{CG,3} = 1{,}21 \tag{A.77}$$

NOTE 1 Even a long intermediate (second) component does not lead to a raise of $f_{CG,\,3}$ over 1,3 of the front end (third) component.

For an easier further derivation, the three factors, f_{BAL}, k_{SYS} and $f_{CG,i}$, of Formula (A.67) are combined to the unbalance reduction factor, $f_{RED,i}$, that limits the basic permissible static unbalance, $U_{STAT,1\,\%}$ [see Formula (3)] of tool system components [see Formula (A.78)].

$$f_{RED,i} = \frac{f_{BAL}}{k_{SYS} \times f_{CG,i}^2} \tag{A.78}$$

These unbalance reducing values, $f_{RED,\,i}$, for the three components of the above standard tool system example are Formula (A.79) to Formula (A.81) for fine balancing ($f_{BAL} = f_{BAL,FINE} = 0{,}2$):

$$f_{RED,1,FINE} = \frac{0{,}2}{3 \times 1{,}0^2} = 0{,}067 \tag{A.79}$$

$$f_{RED,2,FINE} = \frac{0{,}2}{3 \times 1{,}11^2} = 0{,}054 \tag{A.80}$$

$$f_{RED,3,FINE} = \frac{0{,}2}{3 \times 1{,}21^2} = 0{,}046 \tag{A.81}$$

and Formula (A.82) to Formula (A.84) for standard balancing ($f_{BAL} = f_{BAL,STND} = 0{,}8$):

$$f_{RED,1,STND} = 0{,}268 \tag{A.82}$$

$$f_{RED,2,STND} = 0{,}216 \tag{A.83}$$

$$f_{RED,3,STND} = 0{,}184 \tag{A.84}$$

The values $f_{RED,i,FINE}$ of Formula (A.79) to Formula (A.81) are too small and would mean only 25 % of the permissible unbalance of “fine” balanced single tools. It is therefore obvious that it is not possible to realize component unbalances by applying $f_{BAL,FINE} = 0{,}2$, together with $f_{CG,\,i}$ and k_{SYS}.

The weighting factor for standard balancing, $f_{BAL,STND} = 0{,}8$, however, is leading to sensible values of $f_{RED,i,STND}$ between 0,184 and 0,268. The average value, $f_{RED,AVG}$, is calculated using Formula (A.85):

$$f_{RED,AVG} = \frac{f_{RED,1,STND} + f_{RED,2,STND} + f_{RED,3,STND}}{3} = \frac{0{,}268 + 0{,}216 + 0{,}184}{3} = 0{,}22 \qquad \text{(A.85)}$$

and can be set to Formula (A.86):

$$f_{RED,AVG} = f_{BAL,FINE} = 0{,}2 \qquad \text{(A.86)}$$

The conclusion is that components should be balanced to the balancing quality "fine" like single tools.

$$U_{STAT,i,SYS,PER} = 9{,}12 \times 10^5 \times f_{BAL,FINE} \times \frac{C_{DYN}}{n^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG,i}} \right) - \left(U_{BM,ACC} + m_i \times e_{S,i} \right) \qquad \text{(A.87)}$$

Special tool systems may have more than three components. Applying the factor $f_{SYS,k}$ to Formula (A.87) addresses the actual number of components k_{SYS}. Table 4 shows values of $f_{SYS,k}$ for up to six components.

So, the permissible unbalance of a tool system component finally is calculated using Formula (37):

$$U_{STAT,i,SYS,PER} = 9{,}12 \times 10^5 \times f_{BAL,FINE} \times f_{SYS,k} \times \frac{C_{DYN}}{n^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG,i}} \right)$$

$$- \left(U_{BM,ACC} + m_i \times e_{S,i} \right)$$

The rotational speed, n, can be either the general speed of a "catalogue component", n_i, or the individual speed, n_{SYS}, of a certain tool system.

In the worst case, the component unbalances $U_{STAT,i,SYS,PER}$ [see Formula (37)] sum up the unbalance $U_{STAT,SYS,PER,SUM}$ of a tool system [see Formula (A.88)].

$$U_{STAT,SYS,PER,SUM} = \sum_{i=1}^{3} U_{STAT,i,SYS,PER} = U_{STAT,1,SYS,PER} + U_{STAT,2,SYS,PER} + U_{STAT,3,SYS,PER} \tag{A.88}$$

The permissible unbalances of the components ($f_{BAL,FINE} = 0,2$) of the above HSK-A63 tool system example are shown in Formula (A.89):

$$U_{STAT,1,SYS,PER} = U_{STAT,2,SYS,PER} = U_{STAT,3,SYS,PER} = 22,3 \text{ gmm} \tag{A.89}$$

The worst case result of the permissible unbalances of this tool system is shown in Formula (A.90):

$$U_{STAT,SYS,PER,SUM} = U_{STAT,1,SYS,PER} + U_{STAT,2,SYS,PER} + U_{STAT,3,SYS,PER}$$
$$= 3 \times 22,3 \text{ gmm} = 66,9 \text{ gmm} \tag{A.90}$$

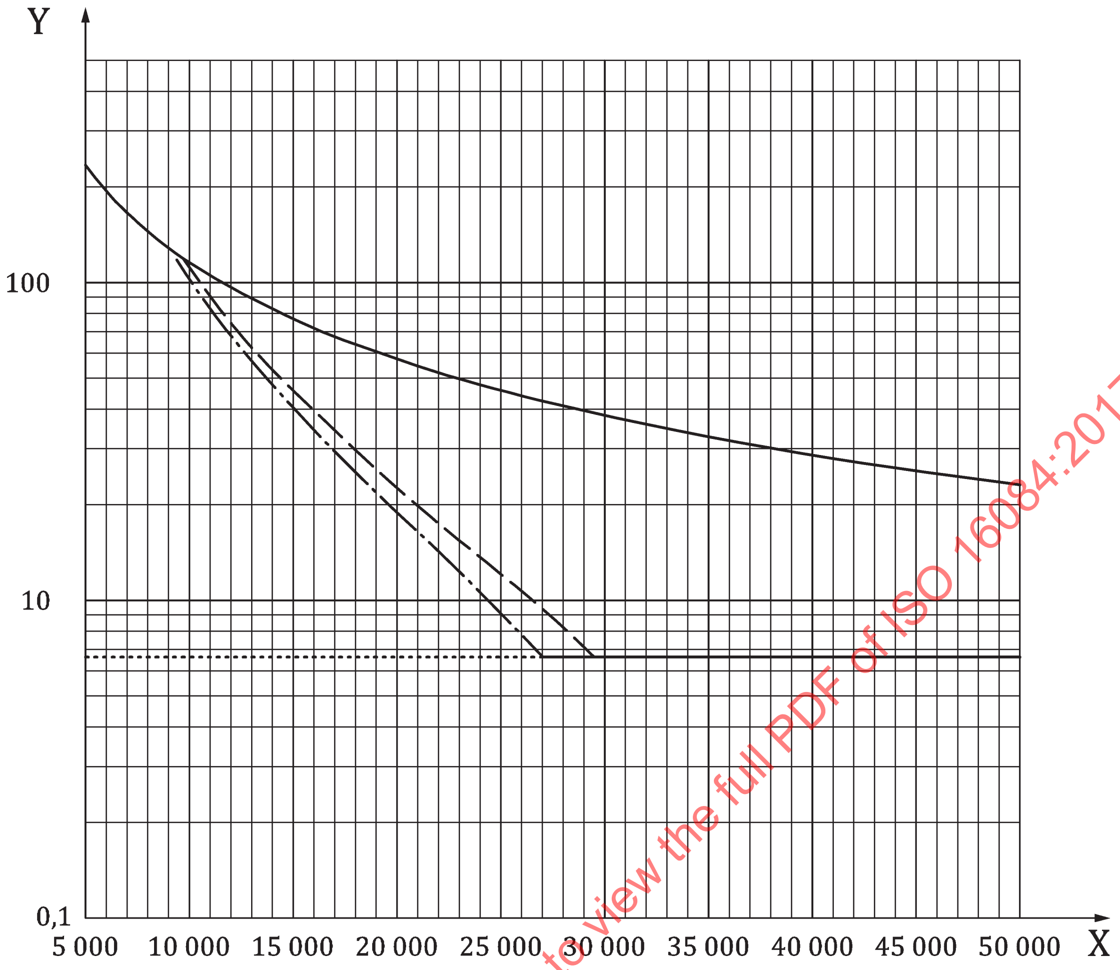
The calculation for the assembled tool system ($f_{BAL,STND} = 0,8$ and $f_{SYS,3} = 1,0$) results in Formula (A.91):

$$U_{STAT,SYS,PER,3} = 74,8 \text{ gmm} \tag{A.91}$$

The sum of the permissible component unbalances is smaller than the permissible unbalance of the assembled tool system. So, the relevant general requirement for a free setup of tool systems is fulfilled.

$$U_{STAT,SYS,PER,SUM} < U_{STAT,SYS,PER,3} \tag{A.92}$$

The diagram of Figure A.5 shows the unbalance graphs of both the sum of the component unbalances, as well as of the calculated unbalance of the assembled tool system. The line of the component sum is beneath the line of the tool system over the whole range of the rotational speed (between the lines of $U_{G\,40}$ and U_{MIN}). The graph of a fine balanced single component of this tool system is shown in the diagram of Figure 13.



Key

X rotational speed (1/min)

Y permissible static unbalance (gmm)

— — — $U_{STAT,SYS,PER,3}$

—·— $U_{STAT,SYS,PER,SUM}$

——— $U_{G\ 40,SYS,3}$

------------- $U_{MIN,SYS,3}$

Figure A.5 — Permissible static unbalances of a HSK-63 component and the standard tool system

A well-aligned standard tool systems, composed of three fine balanced universal components, fulfils the standard quality of static balancing at the same rotational speed. Tool systems, however, are usually in favour of lower operating speeds than the rotational speed to which the participating smaller and lighter components have been balanced for.

NOTE 1 Radial component dislocations may sum up towards the front end of an assembled tool system, thus leading to an increase of the "real" static and/or dynamic unbalance. This depends on the shank types and interfaces, as well as the angular orientation of their radial dislocations. These effects are random (comparable to the orientation of residual component unbalances) and a careful assembly can minimize cumulative radial dislocation. Therefore, the possible single radial dislocation of an individual component is taken into account once.

B.2 also calculates the worst case unbalance of this tool system which might occur if all components were dislocated to their full extent with the same angular orientation during assembly.

It is always recommended to minimize radial dislocation by a careful assembly and finally measure the actual tool system unbalance.

NOTE 2 If balancing of a certain individual component to the required residual unbalance according to Formula (37) would not be possible; the relevant tool system, as a consequence, might exceed its permissible static value as well. If this is seen critical or leading to problems in operation, the residual unbalance of any other component(s) can be minimized as compensation.

NOTE 3 Refer to Figure 15 and note 2 in 5.1 for the rule of counting the number of components k_{SYS} of a tool system.

A.4.4 Maximum radial dislocation of components within a tool system

The maximum radial dislocation of a component $e_{i,SYS,MAX}$ within an assembled tool system could occur if all single dislocations show into the same direction. Although this is very unlikely, it could lead to the maximum unbalance of an assembled tool system, $U_{STAT,SYS,MAX}$.

$$e_{k,SYS} = \sum_{i=1}^{k} e_{S,i} \quad \text{(A.93)}$$

The maximum eccentricities of the three components of a standard tool system according to Formula (39) are given by Formula (A.94) to Formula (A.96).

$$e_{1,SYS} = e_{S,1} \quad \text{(A.94)}$$

$$e_{2,SYS} = e_{S,1} + e_{S,2} \quad \text{(A.95)}$$

$$e_{3,SYS} = e_{S,1} + e_{S,2} + e_{S,3} \quad \text{(A.96)}$$

An example calculation is included in B.2.

A.5 Calculation examples

A.5.1 Static balancing of a single tool with $L_{BL} \leq L_{STAT,MAX}$

The example is based on:

a) HSK-A63 tool with L = 80 mm and L_{BL} = 70 mm (measured from the HSK face/spindle nose to the foremost possible balancing plane); the distance to the tool centre of gravity is L_{CG} = 22 mm;

b) this type of tool is being used for standard machining ($f_{BAL,STND} = 0,8$);

c) interface HSK-A63 → a_M = 50 mm, D_S = 63 mm;

d) double-check whether static or dynamic balancing has to be applied:

— $L_{STAT,MAX} = 2,2 \times D_S = 2,2 \times 63\ \text{mm} = 138,6\ \text{mm}$;

— $L_{BL} = 70\ \text{mm} < L_{STAT,MAX}$;

Result: static balancing is sufficient;

e) tool mass: $m = 600\ \text{g}$;

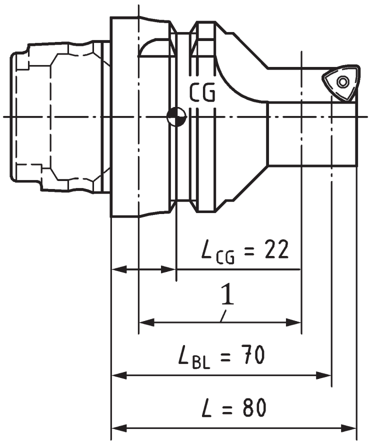
f) cutting edge diameter: $D_C = 30\ \text{mm}$;

g) reference diameter: $D_{REF} = D_S = 63$ mm;

h) rotational speed: $n = 4\,000$ min^{-1};

i) peripheral speed: $v_{REF} = 791$ m/min $< v_{G\,40} = 1\,000$ m/min;

j) radial clamping accuracy HSK-A63 → $e_S = e_{HSK\text{-}63} = 0{,}002$ mm.



Key

1 balancing area

Figure A.6 — HSK-A63 single tool for static balancing

The permissible static residual unbalance is calculated according to the universal Formula (8):

$$U_{STAT,PER} = 9{,}12 \times 10^5 \times \frac{f_{BAL} \times C_{DYN}}{n^2} \times \left(\frac{L_B}{L_B + a_M + L_{CG}} \right) - \left(U_{BM,ACC} + m \times e_S \right)$$

to Formula (A.97):

$$U_{STAT,PER} \leq 9{,}12 \times 10^5 \times \frac{0{,}8 \times 25\,000}{4\,000^2} \times \left(\frac{415}{415 + 50 + 22} \right) - \left(0{,}75 + 600 \times 0{,}002 \right) = 970 \text{ gmm} \quad \text{(A.97)}$$

The minimal achievable residual unbalance of this tool is determined according to Formula (10) by the accuracy of the balancing machine (see Table 2) and the joining repeatability of the tool (mass) as shown in Formula (A.98):

$$U_{MIN} \geq U_{BM,ACC} + m \times e_{HSK} = 0{,}75 \text{ gmm} + 600 \text{ g} \times 0{,}002 \text{ mm} = 1{,}95 \text{ gmm} \quad \text{(A.98)}$$

For the initial balancing ($U_{STAT,PER,TM}$) and a subsequent verification ($U_{STAT,PER,CS}$), the tolerance range of ±15 % can be applied either statically to $U_{STAT,PER}$. or dynamically to $U_{P1,PER}$ and $U_{P2,PER}$ (as described in 4.2.3). So, the following unbalance limits should be set for the two relating balancing procedures of the tool manufacturer and the user [see Formulae (A.99) and (A.100)].

Tool manufacturer:

$$U_{STAT,PER,TM} < 0{,}85 \times U_{STAT,PER,STND} = 0{,}85 \times 970 \text{ gmm} = 825 \text{ gmm} \quad \text{(A.99)}$$

User:

$$U_{STAT,PER,CS} < 1{,}15 \times U_{STAT,PER,STND} = 1{,}15 \times 970 \text{ gmm} = 1\,116 \text{ gmm} \quad \text{(A.100)}$$

A.5.2 Couple unbalance due to quasi-static balancing with $L_{BL} \leq L_{STAT,MAX}$

The example of Figure A.7 is based on:

a) flange tool with $L = 155\text{ mm}$ and $L_{BL} = 140\text{ mm}$;

b) flange interface: $D_S = D_{REF} = 80\text{ mm}$;

c) double-check whether static or dynamic balancing has to be applied:

— $L_{STAT,MAX} = 2{,}2 \times D_S = 2{,}2 \times 80\text{ mm} = 176\text{ mm}$;

— $L_{BL} = 140\text{ mm} < L_{STAT,MAX}$;

Result: static balancing is sufficient;

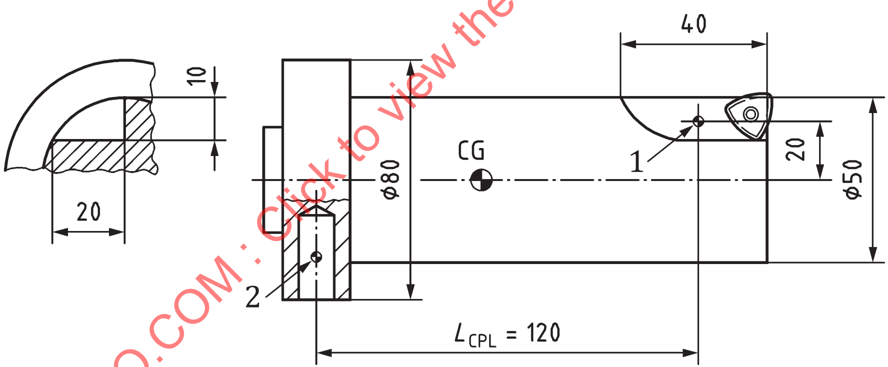
d) cutting edge diameter: $D_C = 50\text{ mm} \rightarrow v_C = 314\text{ m/min}$;

e) radius of the chip groove unbalance: $r_U = 20\text{ mm}$;

f) rotational speed: $n = 1\,000\text{ min}^{-1}$;

g) peripheral speed: $v_{REF} = 503\text{ m/min} < v_{G\,40} = 1\,000\text{ m/min}$;

h) HSK-63-spindle → spindle size #5 → $L_B = 415\text{ mm}$, $C_{DYN} = 25\,000\text{ N}$.



Key

1 centre of unbalance

2 centre of compensation mass

Figure A.7 — Couple unbalance due to quasi-static balancing

Figure A.7 shows a worst-case example of static balancing due to the maximum distance between the initiating unbalance and the material removing bore at the flange. It is only to estimate the influence of a couple unbalance after static balancing.

The missing mass of the chip groove (cross-section similar to a triangle) is estimated to Formula (A.101):

$$m_U \approx 0{,}5 \times W_G \times H_G \times L_G \times \rho_{ST} = 0{,}5 \times 10\ \text{mm} \times 20\ \text{mm} \times 40\ \text{mm} \times 7{,}8\ \frac{\text{mg}}{\text{mm}^3} = 31\,200\ \text{mg} \approx 31\ \text{g} \qquad \text{(A.101)}$$

This means a quasi-static unbalance according to Formula (D.1) given by Formula (A.102):

$$U_{QS} = m_U \times r_U = 31\ \text{g} \times 20\ \text{mm} = 620\ \text{gmm} \qquad \text{(A.102)}$$

The resulting couple unbalance according to Formula (1) is given by Formula (A.103):

$$U_{CPL} = U_{QS} \times L_{CPL} = 620\ \text{gmm} \times 120\ \text{mm} = 74\,400\ \text{gmm}^2 \qquad \text{(A.103)}$$

The load F_B on the spindle bearings according to Formula (2) is calculated using Formula (A.104):

$$F_B = F_{B1,CPL} = F_{B2,CPL} = \frac{U_{CPL}}{L_B} \times \left(\frac{2\pi \times n}{60}\right)^2 = \frac{74\,400 \times 10^{-9}\ \text{kgm}^2}{0{,}415\ \text{m}} \times \left(2\pi \times \frac{2\,000}{60s}\right)^2 = 7{,}86\ \text{N} \qquad \text{(A.104)}$$

Compared to the dynamic load rating C_{DYN} = 25 000 N (see Table 2, spindle size #5), the relevant rate of utilization, $R_{DYN,CPL}$, by this couple unbalance is given by Formula (A.105):

$$R_{DYN,CPL} = \frac{F_B}{C_{DYN}} = \frac{7{,}86\ \text{N}}{25\,000\ \text{N}} \Rightarrow 0{,}031\,\% << 1\,\% \qquad \text{(A.105)}$$

So the total rate of utilization R_{DYN} would only raise from 1 % to 1,031 % if the couple unbalance had the same orientation like the residual static unbalance. Even doubling the rotational speed to 4 000 min^{-1} would just lead to $R_{DYN,CPL}$ = 0,124 %.

Result: Couple unbalances of usual tools due to sensible static balancing do not need to be taken into account regarding the spindle load even at high working speeds. This is indirectly also confirmed by the common way of static balancing in the tool balancing practice. Special applications, however, may need dynamically well-balanced tools.

In case of special tool designs (eccentric tools, non-symmetrical tools of big diameters, etc.), it is recommended to calculate the bearing load due to quasi-static balancing according to Formula (1) and Formula (2).

A.5.3 Dynamic balancing of a single tool with $L_{BL} > L_{STAT,MAX}$

a) HSK-A63 tool with $L = 195\ \text{mm}$ and $L_{BL} = 175\ \text{mm}$ (measured from the HSK face to the front balancing plane). The distance to the tool centre of gravity is $L_{CG} = 75\ \text{mm}$.

b) This type of tool shall be used for a finishing operation, i.e. $f_{BAL,FINE} = 0{,}2$.

c) Interface HSK-A63 → $a_M = 50\ \text{mm}$, $D_S = 63\ \text{mm}$.

d) Verification whether static or dynamic balancing is required:

— $L_{STAT,MAX} = 2{,}2 \times D_S = 2{,}2 \times 63\ \text{mm} = 138{,}6\ \text{mm}$;

— $L_{BL} = 175\ \text{mm} > L_{STAT,MAX}$.

Result: The tool shall be balanced dynamically.

e) Tool mass: $m = 1\,400\ \text{g}$.

f) Cutting edge diameter: $D_C = 30\ \text{mm}$.

