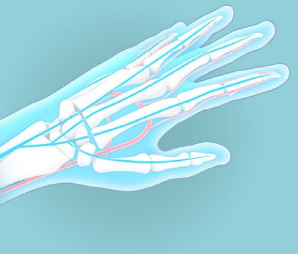
HAND ARM VIBRATION



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Institut National de Recherche et de Sécurité

Vibration emission of grinders: experiments and model

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Summary

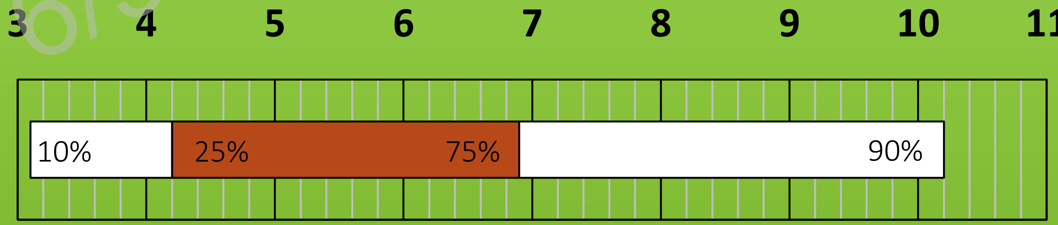
1. Introduction
2. First experimental tests: according to the standard EN 60745-2-3
3. Second experimental tests: hanged grinder
4. Numerical Model
5. Discussion

Introduction

Statistical total vibration exposure for angle grinders

Vibration magnitude a_{hv} (m/s²)

(frequency-weighted root-mean-square acceleration)

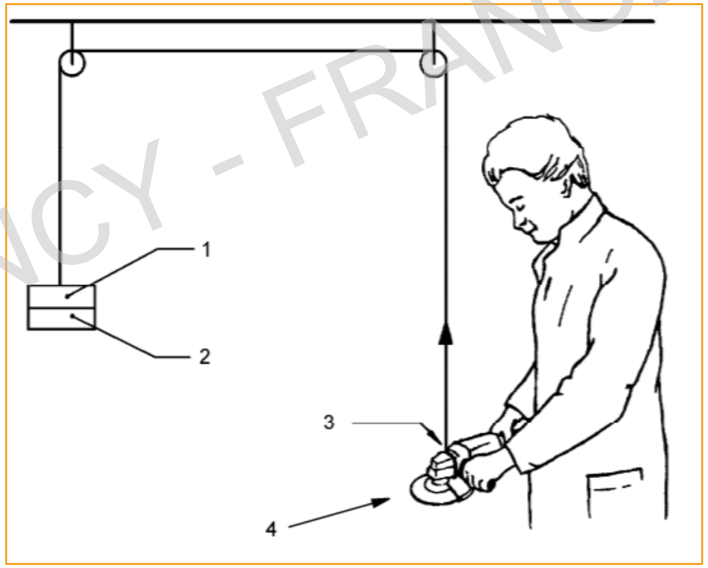


Data from CEN/TR1030-2:2016 (106 measurements between 2005-2014)

Introduction

Real condition

Standard EN 60745-2-3

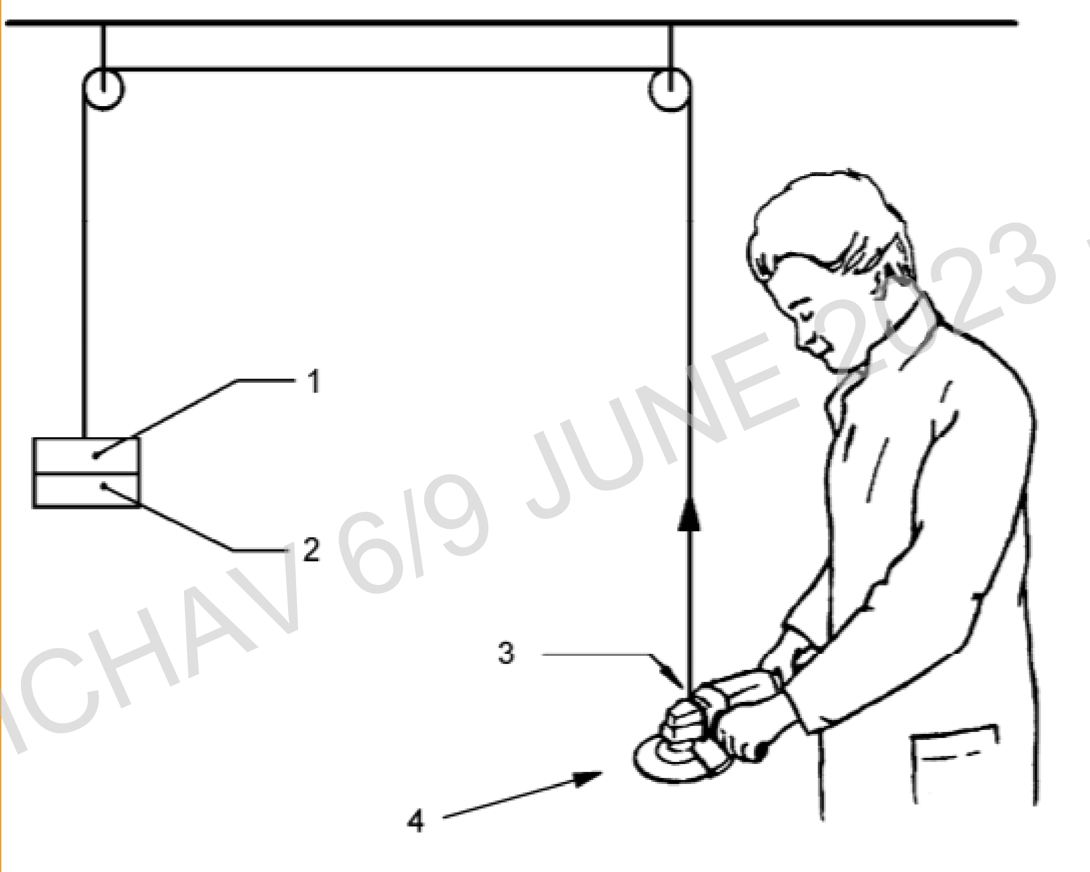


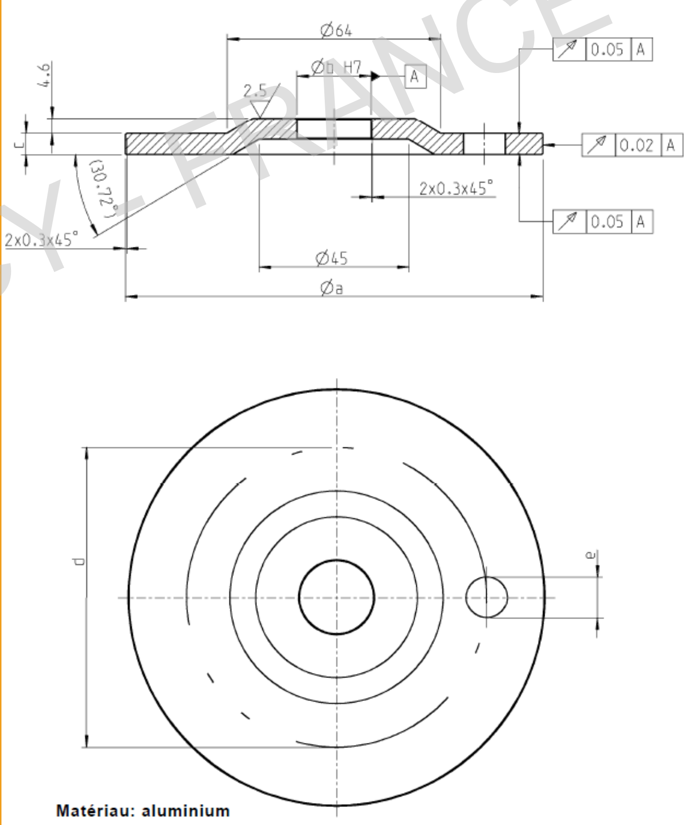
To compare vibration emission of angle grinder

➢ promote less vibrating machine

First experimental tests: according to the standard EN 60745-2-3

Standard EN 60745-2-3



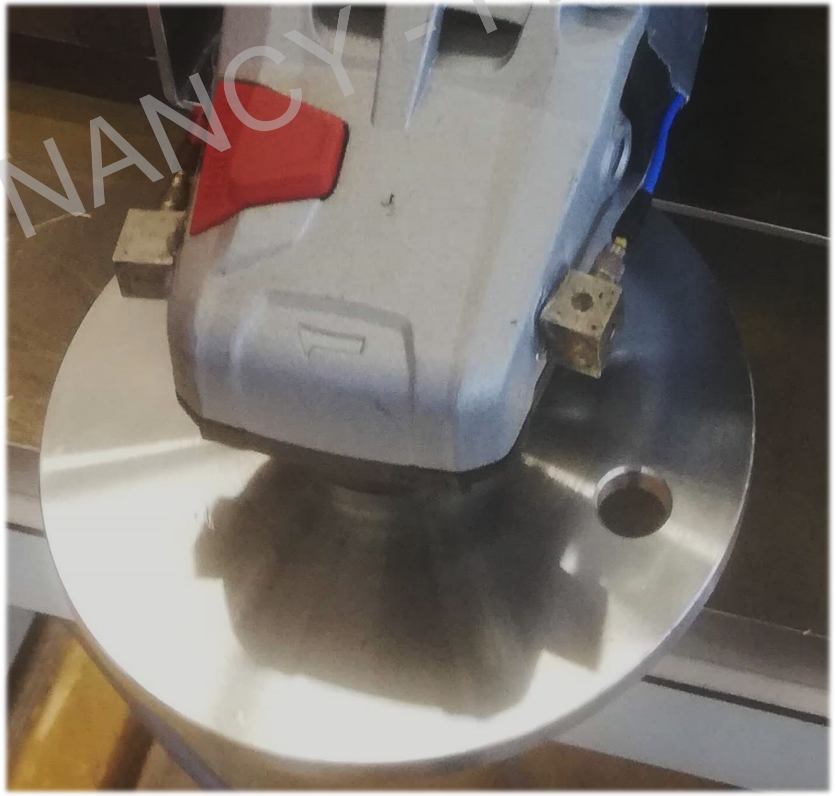


Standard disc with artificial unbalance

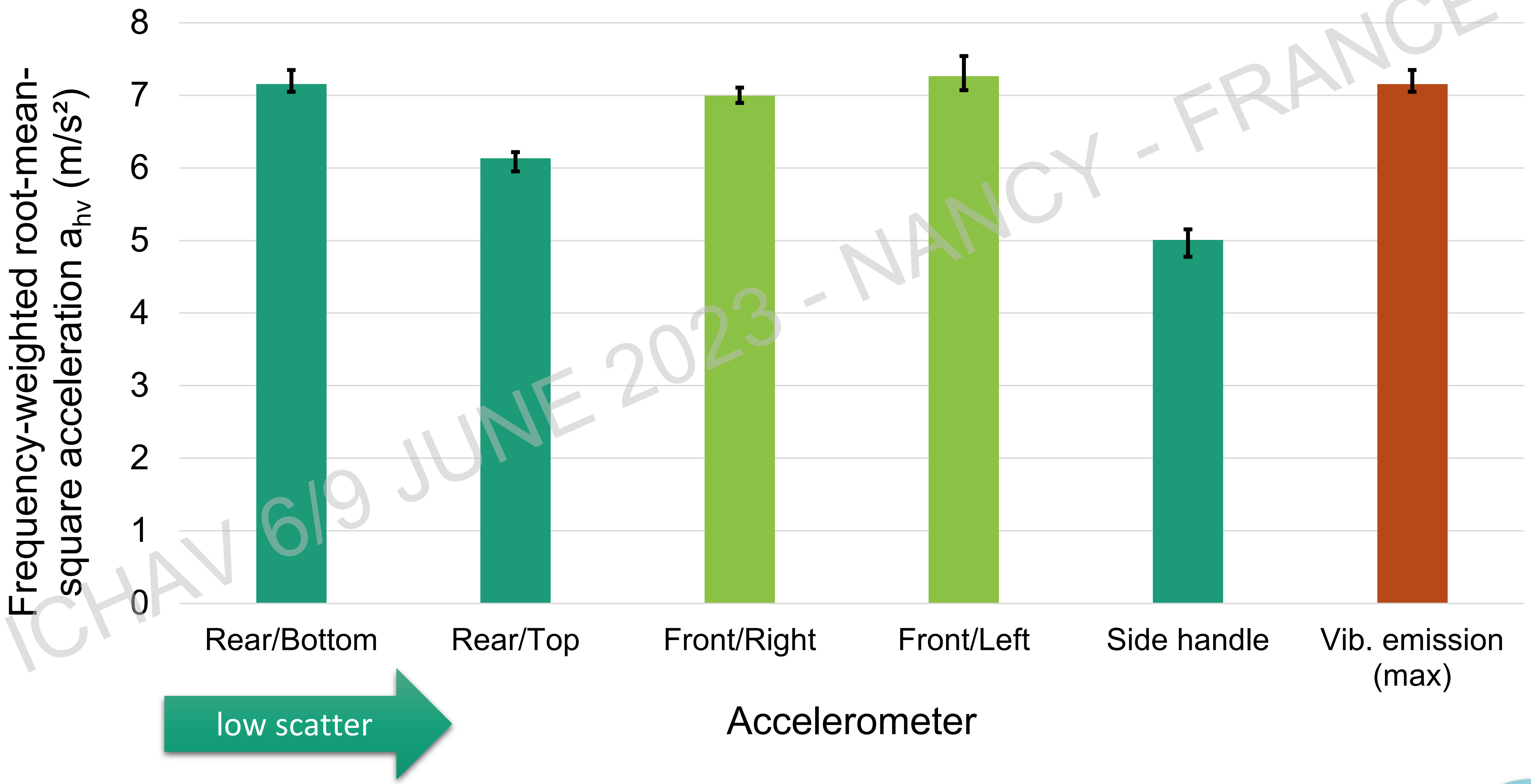
First experimental tests: according to the standard EN 60745-2-3

Metabo W12-125 Quick

5 PCB triaxle piezoelectric accelerometers

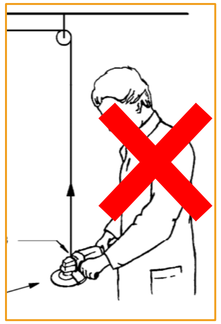


First experimental tests: according to the standard EN 60745-2-3

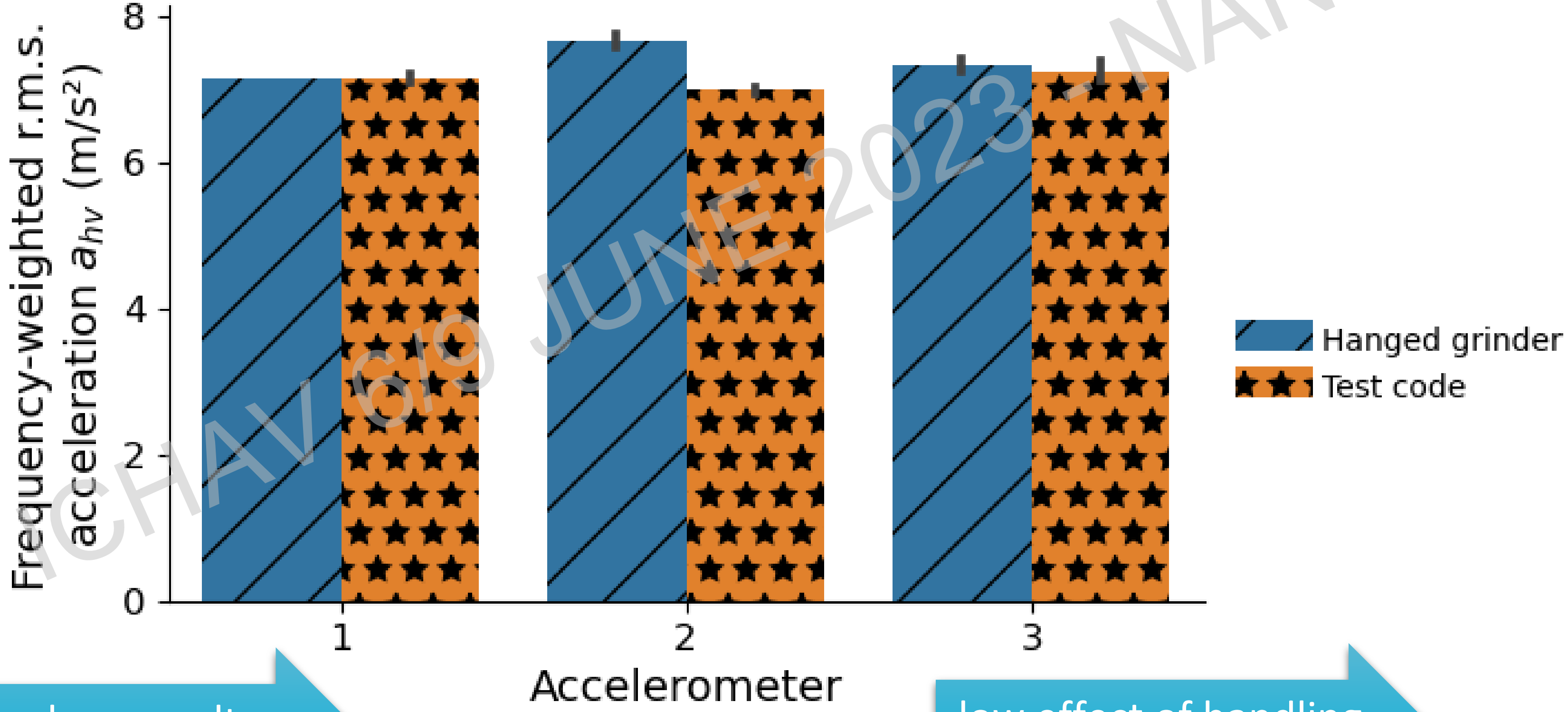


Note: error bars represents minimal and maximal values for 7 measurements

Second experimental tests: hanged grinder



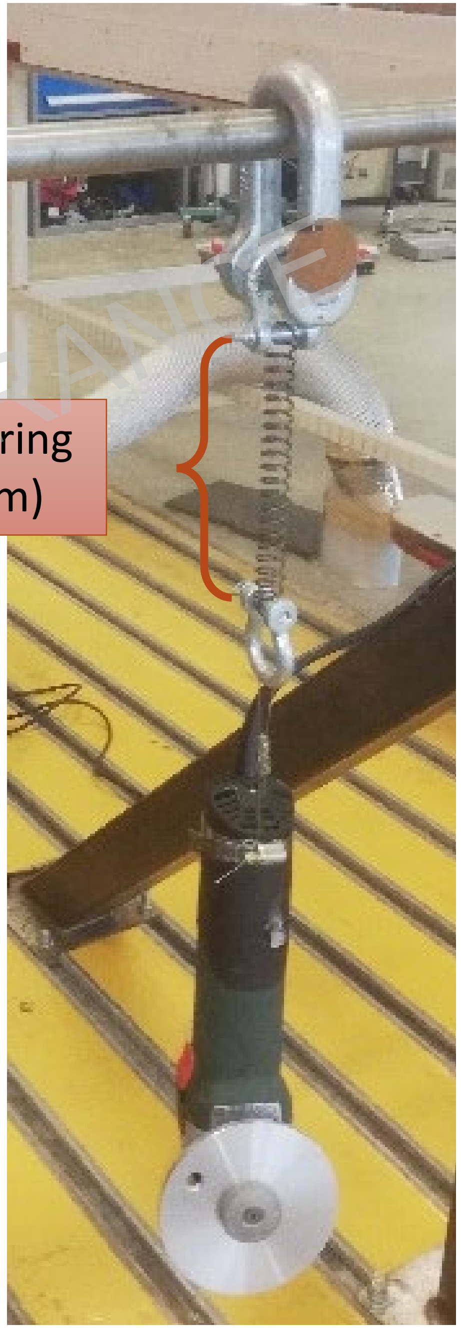
Second experimental tests: without operator and without side handle



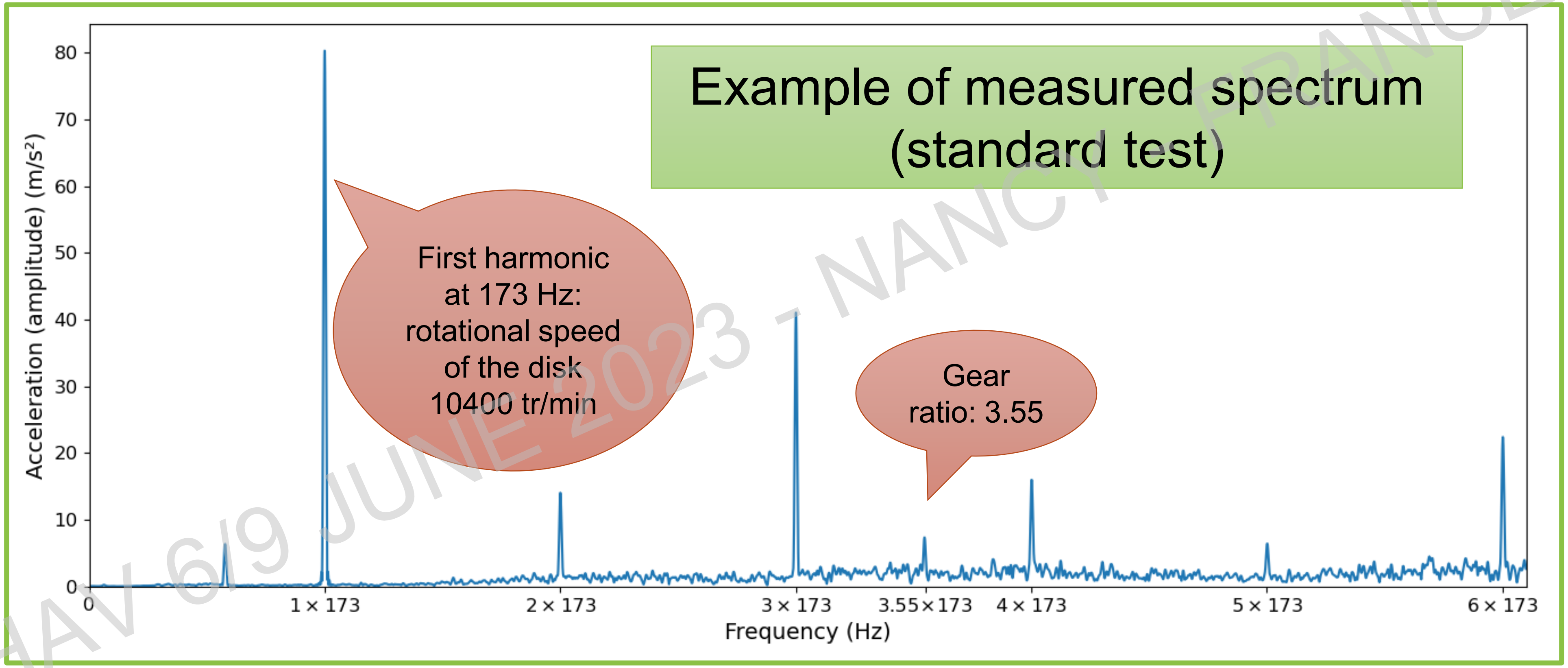
close results

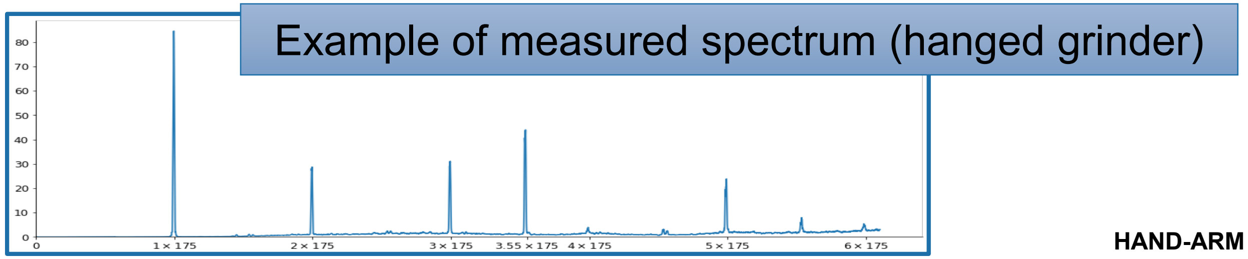
low effect of handling

Tension spring (1930 N/m)

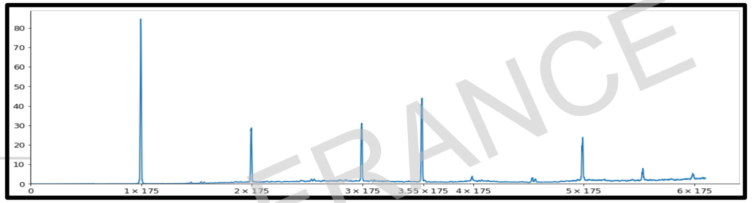


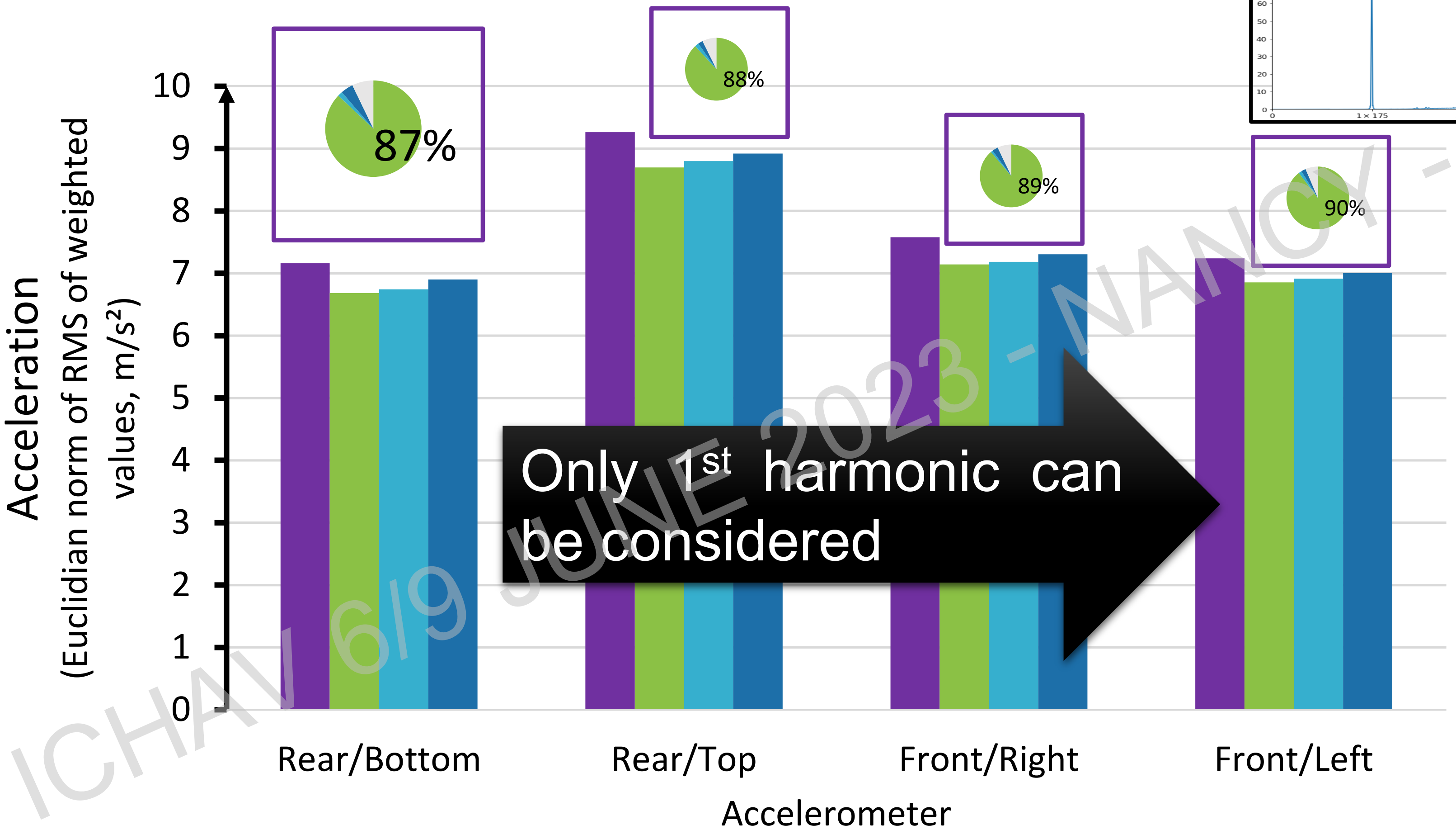
Experimental tests: frequency analysis



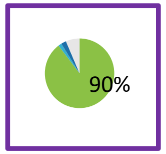


Second experimental tests: proportion of harmonics in RMS of weighted acceleration



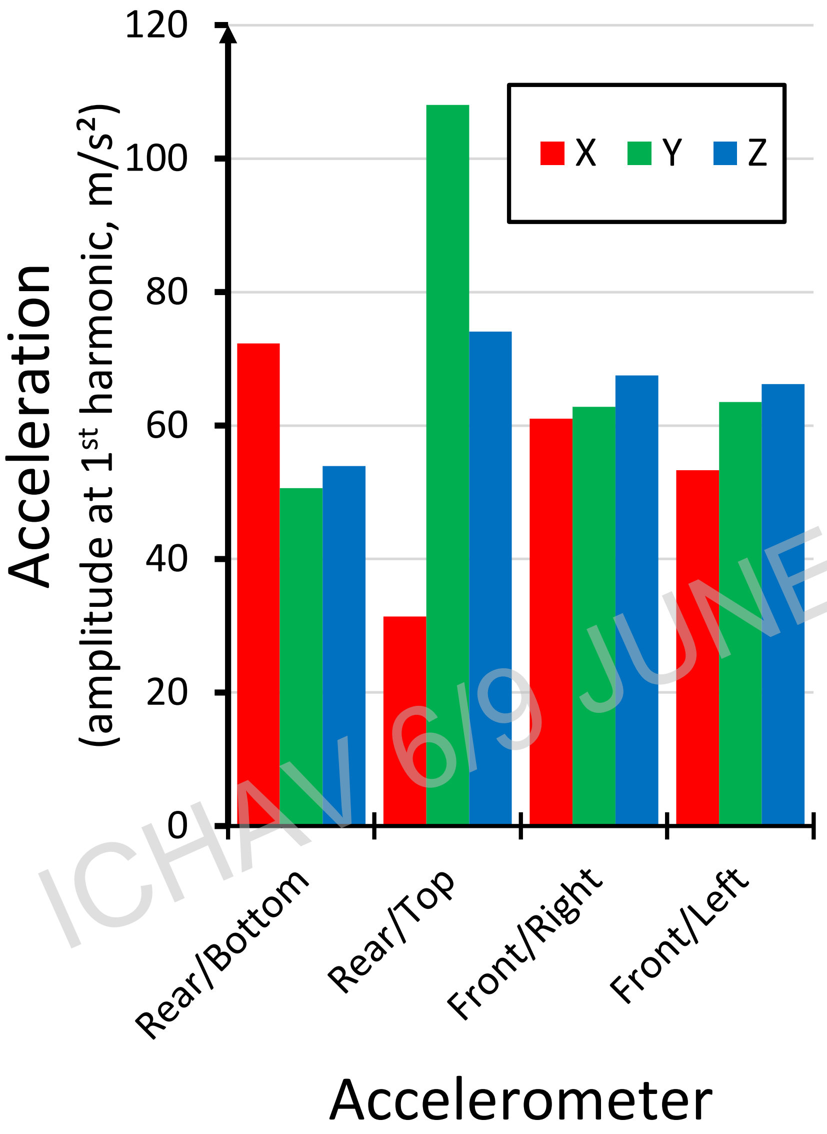


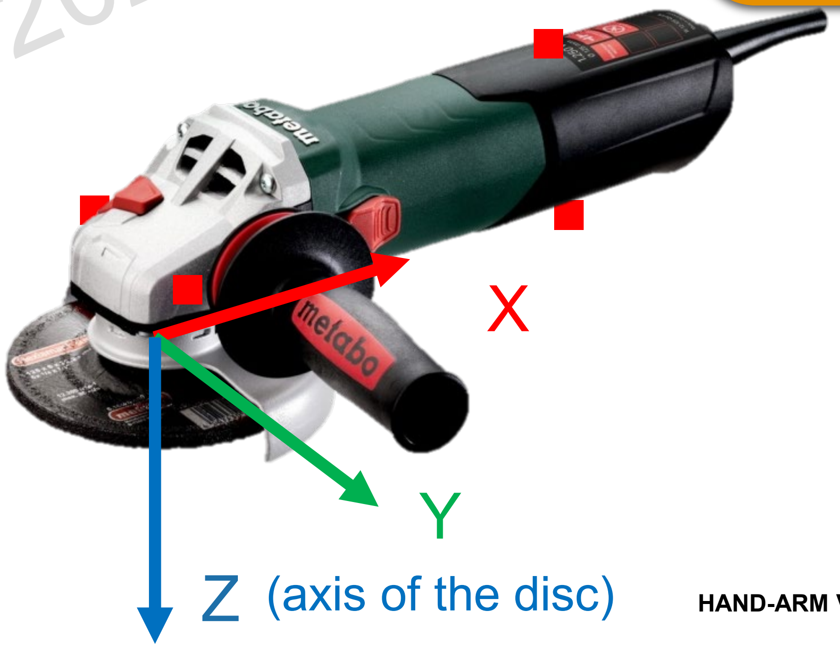
RMS of harmonics are PSD of accelerations integrated on 10 Hz-wide bands.



Proportion of mean squares of weighted acceleration ($\frac{A_i^2}{a_{hv}^2}$, (m/s²)²/(m/s²)²)

Second experimental tests: analysis of motion





Location and direction of accelerometers on grinder measured by 3D scan

Second experimental tests: analysis of motion

$\vec{\Omega}$: Instantaneous angular velocity (rad/s)

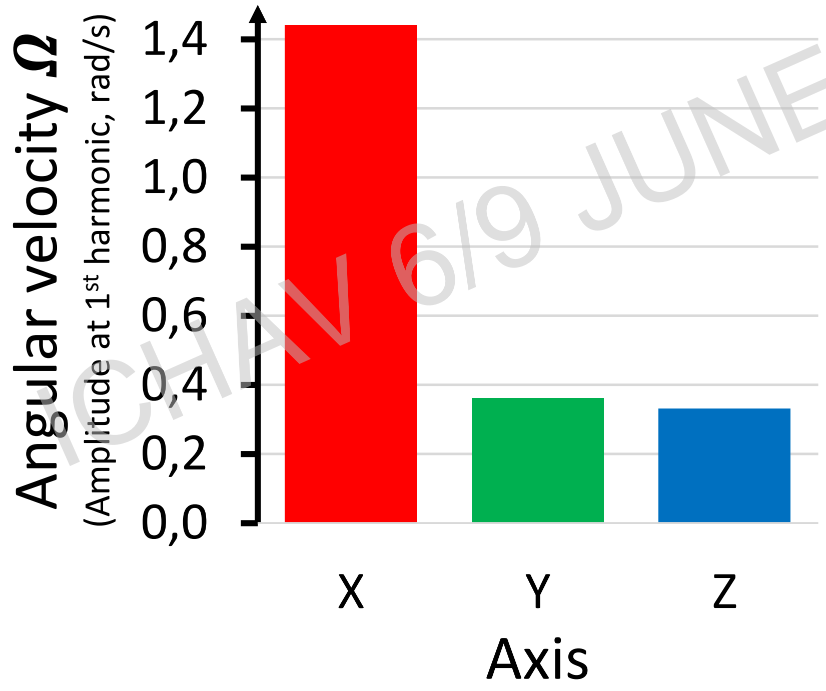
According that :

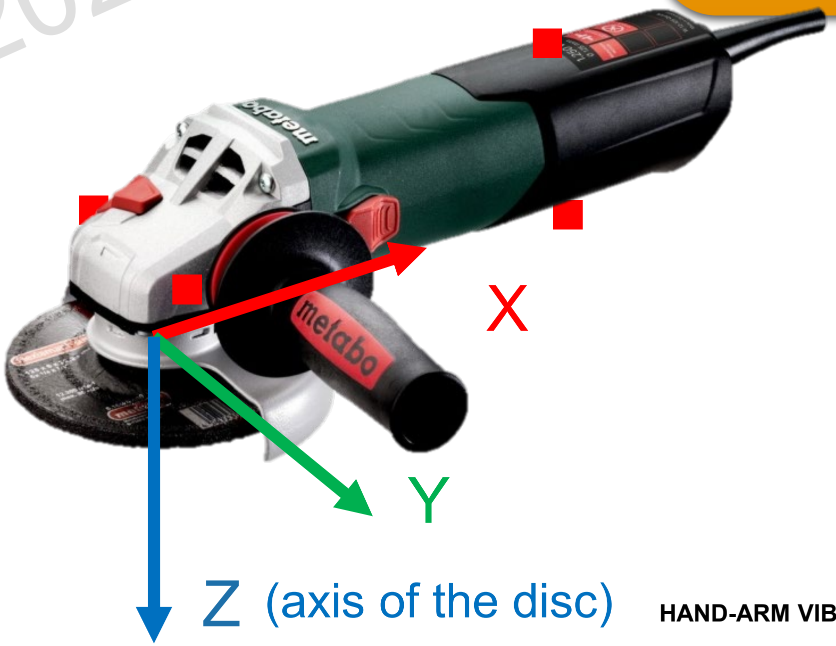
$$\vec{v}_B - \vec{v}_A = \overrightarrow{BA} \wedge \vec{\Omega}$$

for a rigid body,

$\vec{\Omega}$ obtained by minimizing the matrix form of the previous formula with 4 complex amplitudes at 1st harmonic of velocities

Position and orientation of accelerometers on grinder measured by 3D scan





Numerical Model

Two rigid bodies:
1. disc and
2. body of the grinder

1 pivot link

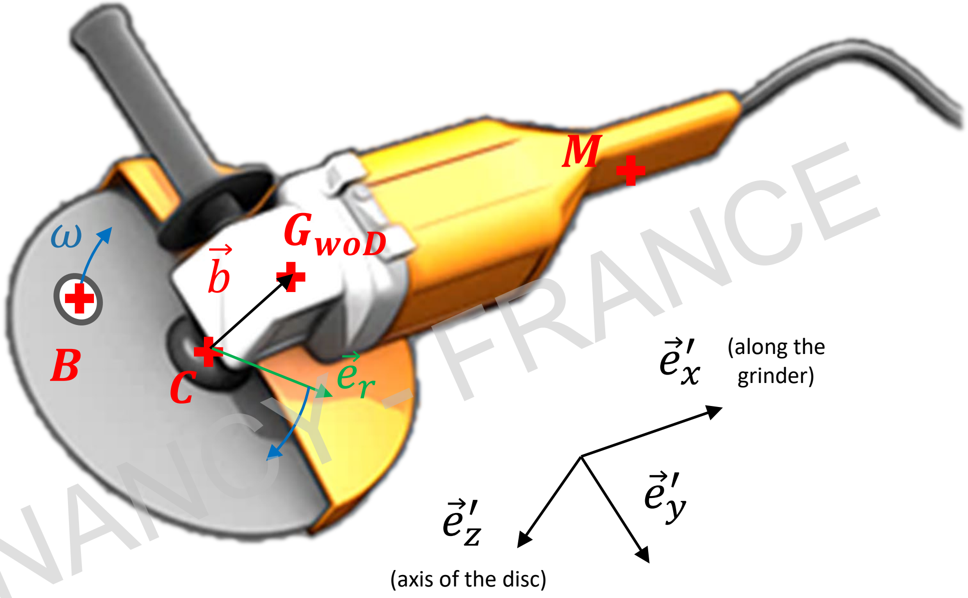
Constant rotation speed $\boldsymbol{\omega}$

1 hole:
change mass center and inertia tensor of the disc

Spring and gravity neglected:
associated with low frequency modes,
e.g.: $\frac{1}{2\pi}\sqrt{\frac{g}{L}} \lesssim 2$ Hz, $\sqrt{\frac{k}{m}} \lesssim 30$ Hz

Numerical Model

M	Point on the running grinder	$\vec{e}_r$	$-\overrightarrow{CB}/\lVert\overrightarrow{CB}\rVert$, rotating vector in the plane of the disk
C	Center of the disk	bal	Imbalance of the perforated disk
G_{woD}	Center of mass of the grinder w/o the disk	m_T	Total mass of the system
$\vec{b}$	$= \overrightarrow{CG_{woD}}$	m_G	Mass of the grinder w/o the disk
ω	Rotational speed of the disk	$\bar{\bar{I}}_T$	Sum of the inertia tensor of rigid bodies written in their center of mass
$\vec{\Omega}$	Angular velocity		



Equations of motion of the grinder

(steady-state, first harmonic, after simplifications)

($\vec{a}_M$, $\vec{\Omega}$ and $\vec{e}_r$ represent complex amplitude at the frequency $\omega/2\pi$)

Acceleration of a point M on the running grinder	$\vec{a}_M = \frac{\text{bal}}{m_T}\omega^2\vec{e}_r + j\omega\vec{\Omega} \wedge \left(\overrightarrow{CM} - \frac{m_G}{m_T}\vec{b}\right)$
Angular velocity of the grinder	$\bar{\bar{I}}_T\vec{\Omega} = j\omega m_G \frac{\text{bal}}{m_T}\vec{b} \wedge \vec{e}_r$

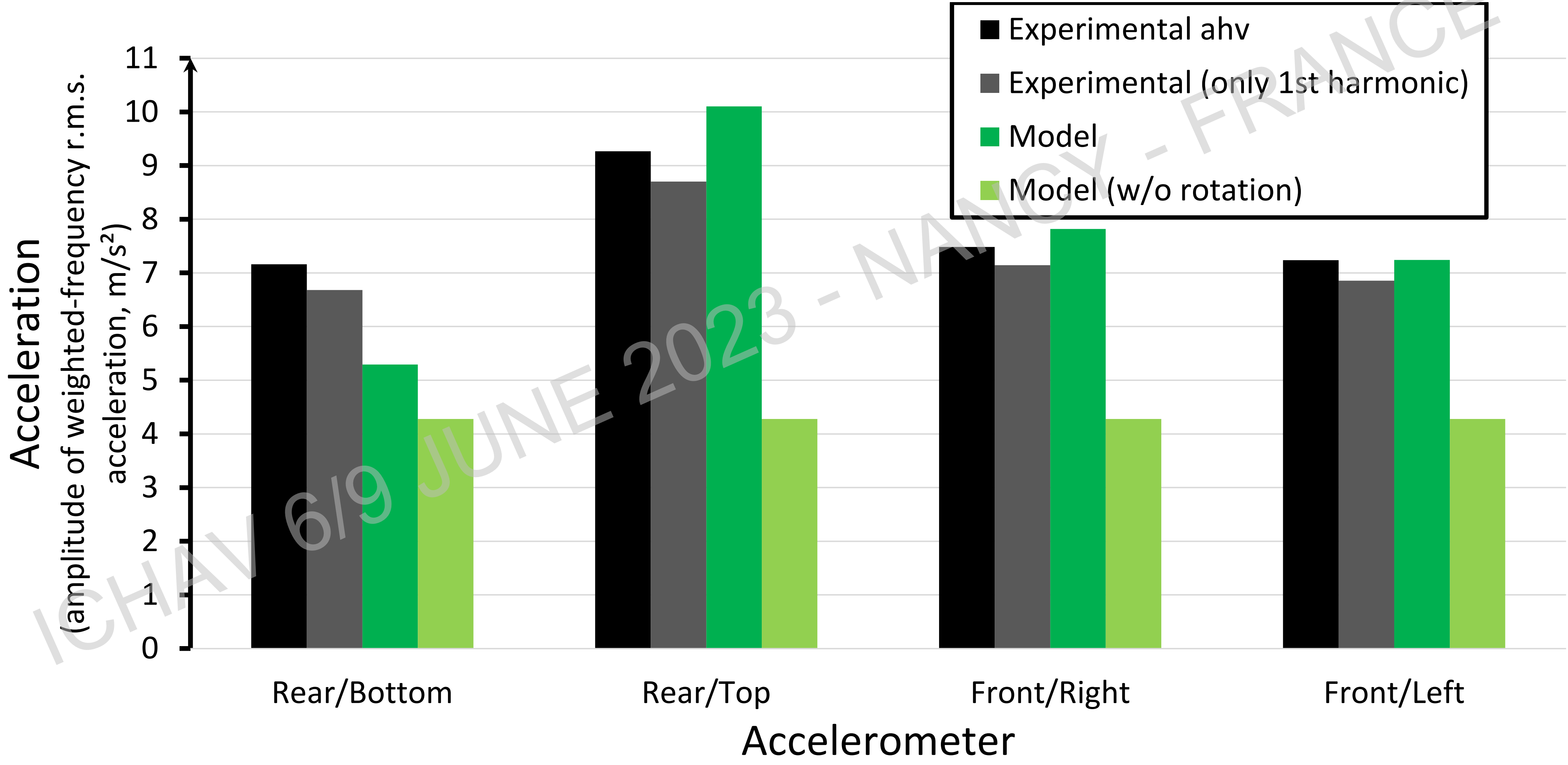
In the coordinate system linked to the grinder ($\vec{e}'_x$, $\vec{e}'_y$, $\vec{e}'_z$), we can write:

$$\vec{e}_r \equiv \begin{pmatrix} 1 \\ i \\ 0 \end{pmatrix},\ \vec{b} \equiv \begin{pmatrix} b_x \\ b_y \sim 0 \\ b_z \end{pmatrix},\ \bar{\bar{I}}_T \equiv \begin{pmatrix} I_{xx} & I_{xy} & I_{xz} \\ I_{xy} & I_{yy} & I_{yz} \\ I_{xz} & I_{yz} & I_{zz} \end{pmatrix}$$

and deduce $\vec{\Omega}$ and $\vec{a}_M$.

Numerical Model

Inertia moment tensor $\bar{\bar{I}}_B$ and center of mass $\boldsymbol{G_{woD}}$: provided by the manufacturer from their detailed CAD



Discussion: second grinder with flexible handles

Bosch GWS24-230 LVI

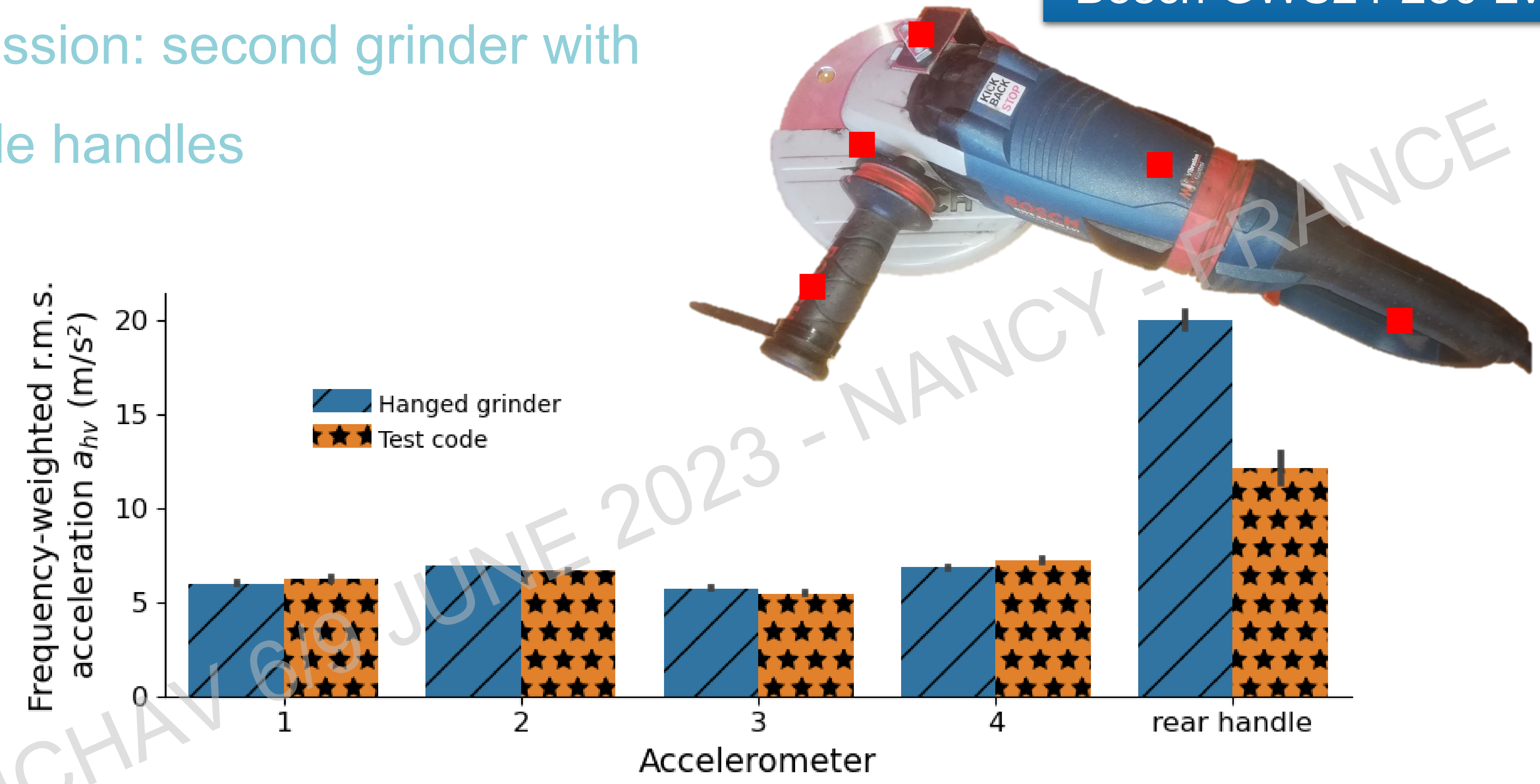
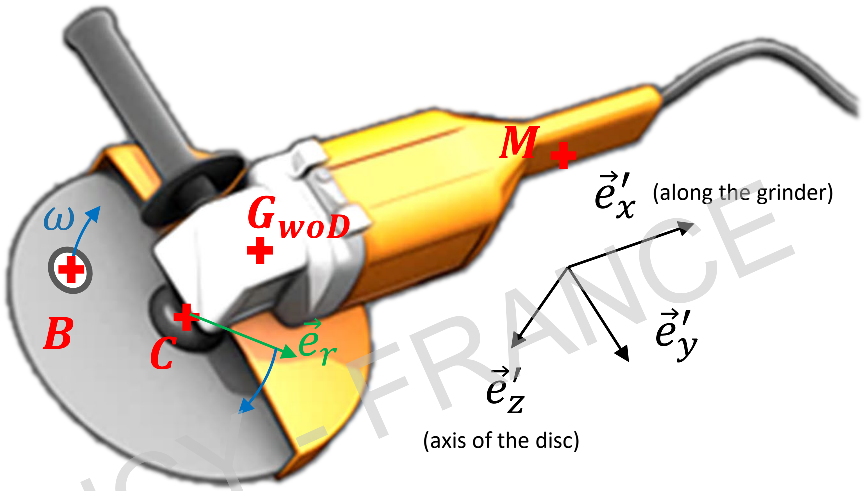


Figure 3: Frequency-weighted root-mean-square acceleration a_{hv} measured during test codes and tests with the hanging grinder. The vertical black lines indicate standard deviation.

Discussion: ways to reduce vibration

Acceleration of a point M on the running grinder	$\vec{a}_M \approx \omega^2 \frac{\text{bal}}{m_T}\left[\vec{e}_r - \bar{\bar{I}}_T^{-1}\left(\overrightarrow{CG_{woD}} \wedge \vec{e}_r\right) \wedge \overrightarrow{G_{woD}M}\right]$



1. reduce rotational speed of the disc ω,
2. increase mass of the grinder m_T,
3. increase rotational inertia $\bar{\bar{I}}_T$ by sharing mass far from mass center,
4. bring the center of mass G_{woD} closer to the hands or the center of disc C
5. use flexible part → model can be used to design flexible part

M	Point on the running grinder	$\vec{e}_r$	$-\overrightarrow{CB}/\lVert\overrightarrow{CB}\rVert$, rotating vector in the plane of the disk
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G_{woD}	Center of mass of the grinder w/o the disk	m_T	Total mass of the system
ω	Rotational speed of the disk	$\bar{\bar{I}}_T$	Sum of the inertia tensor of rigid bodies written in their center of mass