FACULTY OF MECHANICAL ENGINEERING
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Improvements in Vibroacoustic Performance – Review of Activities and Results under the WP 05 Advanced Turbochargers for Diesel Engine Applications

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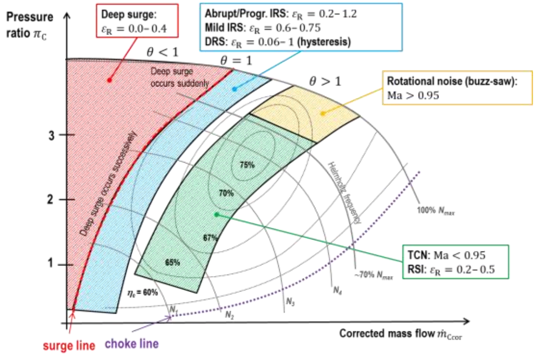
Achieved under WP 05 Advanced Turbochargers for Diesel Engine Applications in 2020.

Content

- A Review of Noise and Vibration Sources of Turbochargers
- Development of Compressor Design Enabling the Reduction of Aerodynamic Bored Noise
- Results of Activities



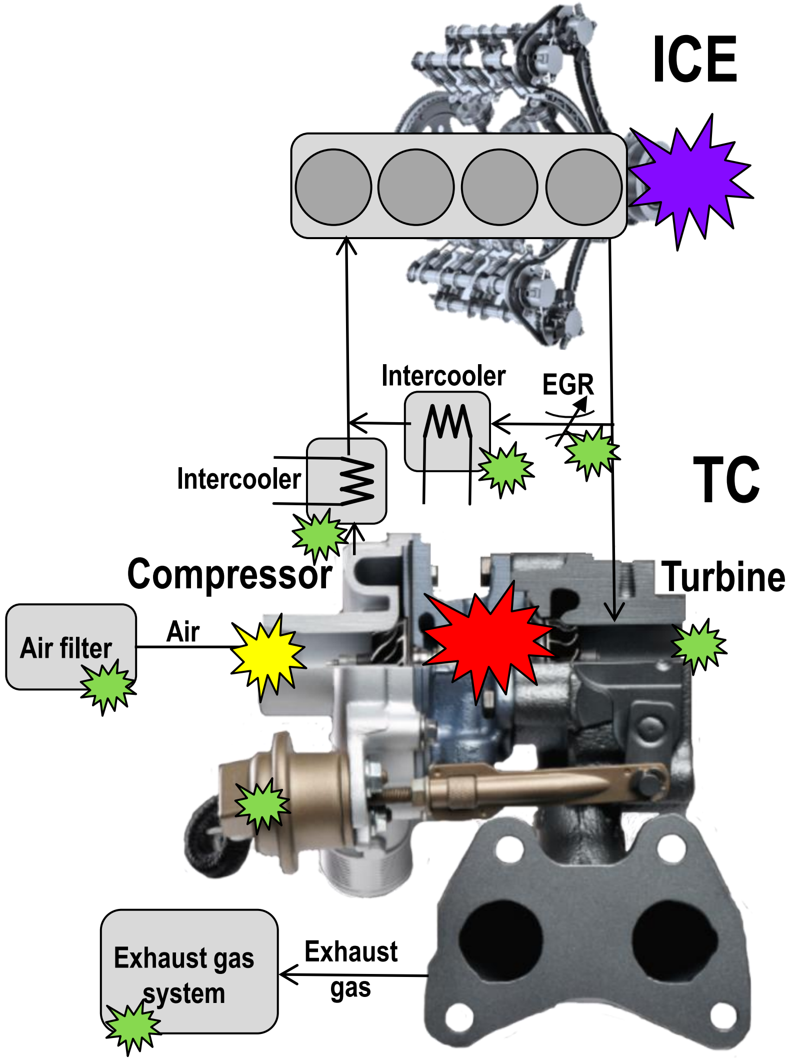
A Review of Noise and Vibration Sources of Turbochargers

Analytical Tool for the Identification of Vibration and Noise Sources in Turbochargers

Goal of the Review

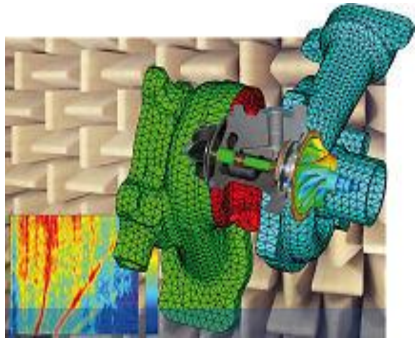
- Find the information for the analytical tool identifying the typical sources of vibration and noise generated by turbochargers (TC) and internal combustion engines (ICE) based on measured vibroacoustic data.

- Identification of noise and vibration sources:
  - Engine based noise
  - Rotordynamic noise sources
  - Aerodynamic noise sources
  - Response noise



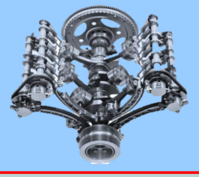
Overview of Vibroacoustic Sources of Turbochargers

Structural borne noise



Powertrain induced

Engine generated vibrations



Unbalance and gas pressure forces.
50 ÷ 500 Hz

Rotor dynamics & Oil film bearings

Constant-tone noise

Oil whirling in the oil-film radial bearing.
250÷1 000 Hz

Unbalancing whistle

Unbalanced rotor and unbalance change of the rotor after a long-term operating time.
1 000 ÷ 4 500 Hz

Rotor dynamics & Rolling bearings

Wear noise

Rolling bearings, wear defects of balls, rollers, cage, inner, and outer races.
5 000 ÷ 15 000 Hz

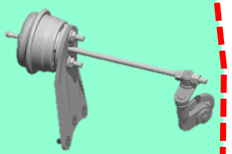
High-order harmonic noise

Rolling bearing contacts.
2 000 ÷ 15 000 Hz

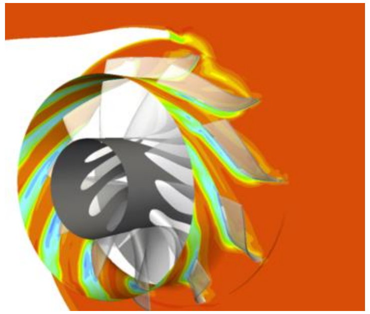
Wastegate

Crackling noise

WG vibrations about its pivot axis. 7 000÷15000 Hz



Airborne noise

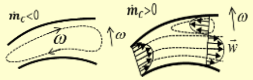


Compressor & Turbine Aerodynamics

Whining noise

Deep surge condition in the compressor wheel.
200 ÷ 2 700 Hz

Growling noise



Partial reversal of the charge air in the compressor wheel.
1 200 ÷ 3 500 Hz

Tip clearance noise

Flow through the gap between the compressor shroud and the blade tip. 500 ÷ 4 000 Hz

Pulsation whistle

Slight differences in compressor wheel chamber volumes due to milling or molding process variations.
1 200 ÷ 4 500 Hz

Rotating noise

Rotating blades of the compressor or turbine wheels.
8 000 ÷ 18 000 Hz

Vibrations generated by the ICE

❑ Cranktrain gas pressure forces

➔ Engine frequency ratio of excitations by **four-stroke multi-cylinder engines**

$$\boxed{\varepsilon_{\mathrm{E}} = \frac{f}{f_{\mathrm{n}}} = \kappa i_{\mathrm{v}}}$$

$$\boxed{\kappa = \boxed{0,5;} \quad 1; \quad 1,5; \quad 2 \ldots}$$

$\kappa = 0,5$ ⇨ <u>TCNOISE: </u>firing frequency dominantly

i_{v}	– number of cylinders [-]
κ	– harmonic order [-]
f_{n}	– rotational frequency [Hz]
n_{E}	– engine speed [min^{-1}]

❑ Cranktrain inertia forces

➔ Engine frequency ratio of excitations by unbalance inertia forces and moments

$$\boxed{\varepsilon_{\mathrm{E}} = \kappa}$$

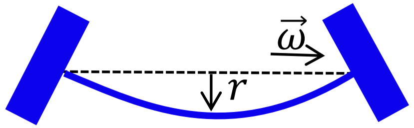
<u>TCNOISE: automatically generated based on:</u>

Number of cylinders
Number of cylinder rows (in-line, V-engines)
Firing order

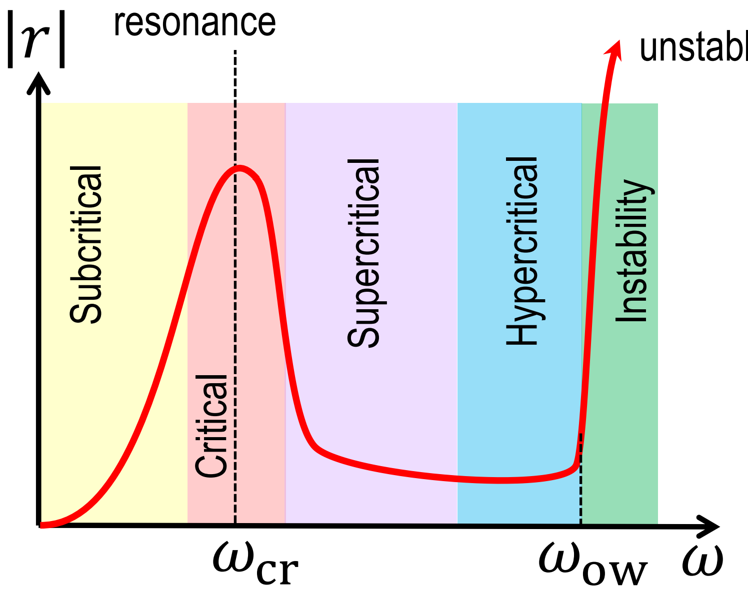
Unbalance Whistle

- Typical noise due to turbocharger rotor unbalance
- Rotor frequency ratio

$$\boxed{\varepsilon_{\mathrm{R}} = \frac{f_{\mathrm{uw}}}{f_{\mathrm{R}}} = 1}$$



- Subdivision
  - Subcritical rotor speeds $(\omega \ll \omega_{\mathrm{cr}})$
  - Critical rotor speed $(\omega = \omega_{\mathrm{cr}})$
  - Supercritical rotor speeds $(\omega > \omega_{\mathrm{cr}})$
  - Hypercritical speeds $(\omega \gg \omega_{\mathrm{cr}})$
  - Speeds of instability $(\omega = \omega_{\mathrm{ow}})$

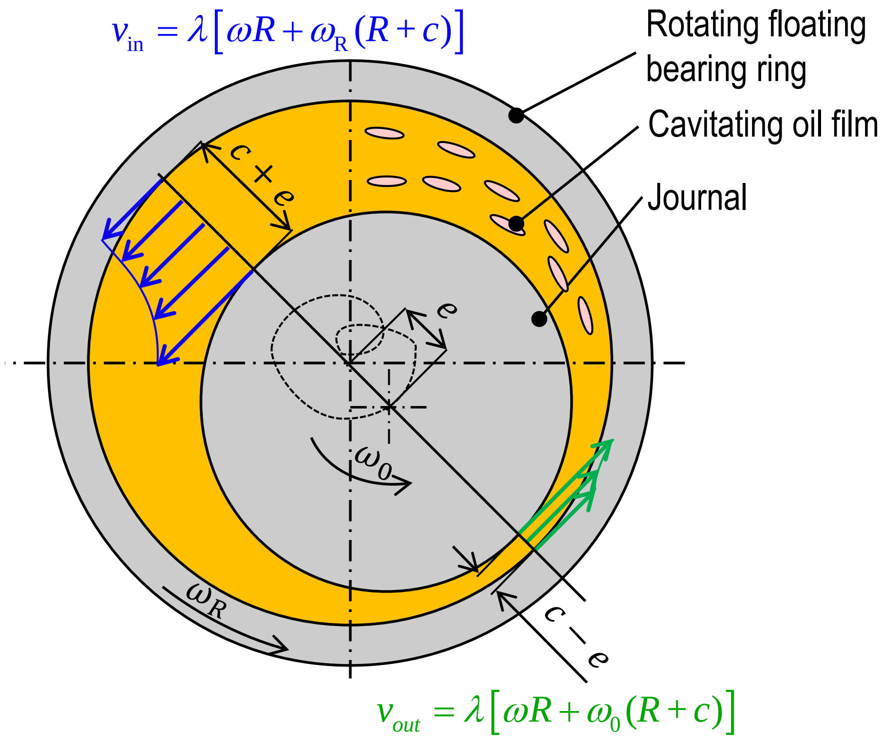


f_{uw} – unbalance whistle frequency [Hz]
f_{R} – rotor speed frequency [Hz]
ω – angular velocity [rad.s^{-1}]

[1]NGUYEN-SCHÄFER, H. Rotordynamics of Automotive Turbochargers. Second Edition. Ludwigsburg, Germany: Springer, 2015. ISBN 978-3-319-17643-7.

Constant Tone Noise

- **Oil whirl** is a kind of self-excited instability with a sub-synchronous frequency.
- Instability of the oil whirl at the threshold rotor speed (ω_{ow}) leads to **oil whip**.



- Semi-floating ring or fixed bearings

$$\varepsilon_R = \frac{\omega_w}{\omega} \doteq \lambda \doteq 0,5$$

- Inner oil film of automotive TC

$$\varepsilon_R \doteq \lambda_{in}(1 + x_{RSR})$$

Typically at low speeds: $\varepsilon_R \doteq 0.6$

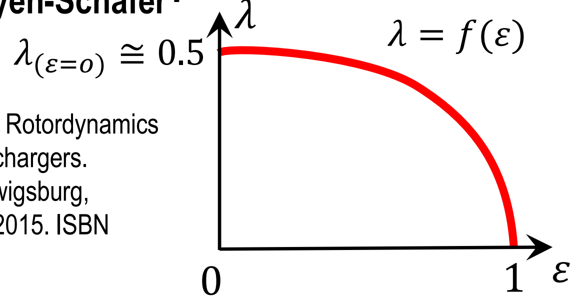
Typically at high speeds: $\varepsilon_R \doteq 0.3$

- Outer oil film of automotive TC

$$\varepsilon_R \doteq \lambda_{out} x_{RSR}$$

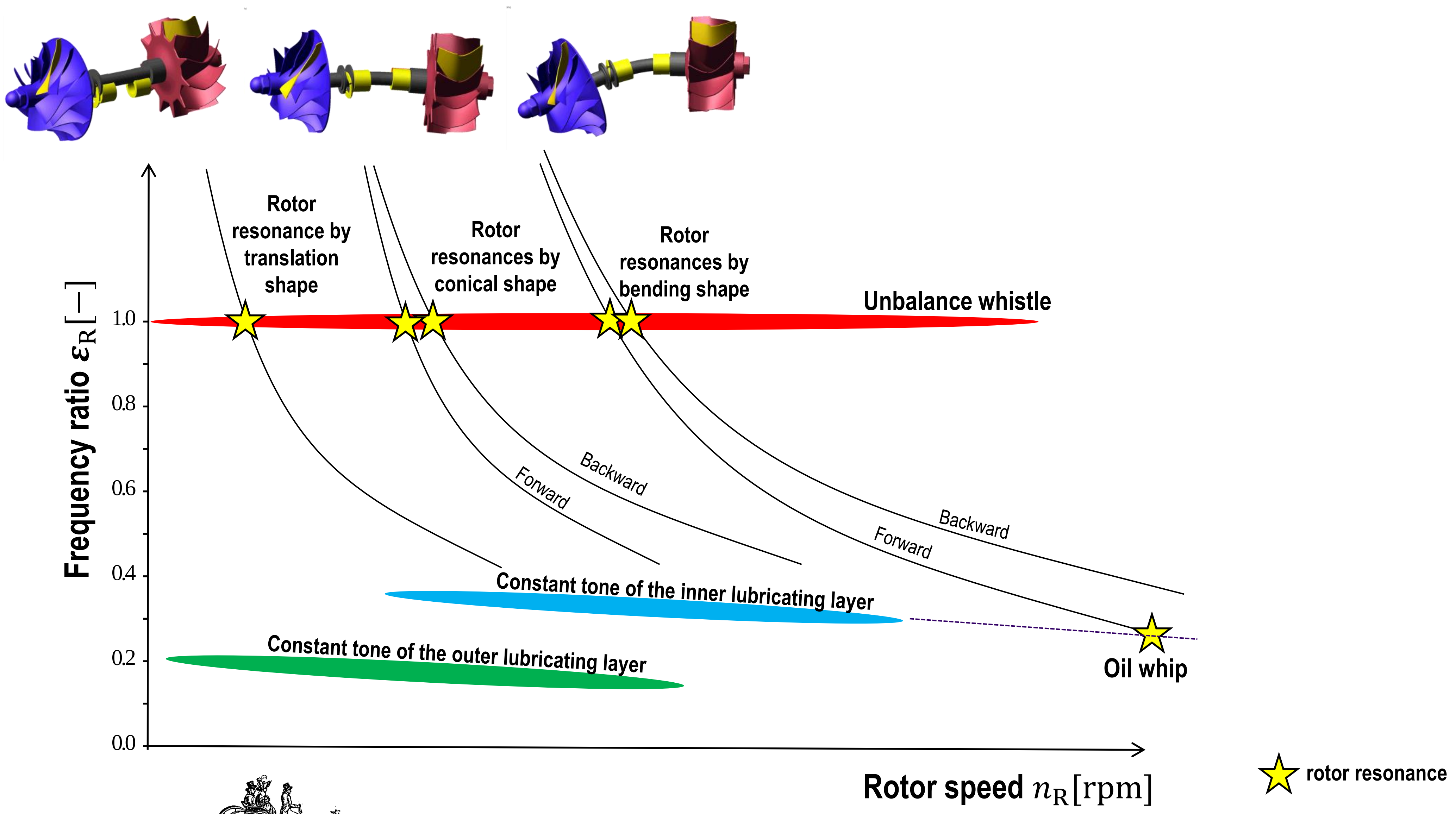
Typically: $\varepsilon_R \approx 0.1$

Fluid circumferential average velocity ratio λ velocity according Nguyen-Schäfer[1]



[1] Nguyen-Schäfer, H. Rotordynamics of Automotive Turbochargers. Second Edition. Ludwigsburg, Germany: Springer, 2015. ISBN 978-3-319-17643-7.

Campbell Plot describing Mechanical Noise of Turbochargers



Compressor & Turbine Aerodynamic Noise

- Number of rotating stall cells: $n \geq 1$
- Frequency ratio of rotation of rotating stall: $\varepsilon_S = \dfrac{f_S}{f_R} = \dfrac{\varepsilon_R}{n}$
- Relative amplitude of oscillations: $A = 0\text{–}1$
- Compressor pressure ratio $\pi_c = p_2/\, p_1$
- Surge line mass flow ratio $\theta = \dot{m}_C/\dot{m}_{Csurge}$

- Rotor–stator Interaction (RSI) according to Marshall and Sorokes[3]

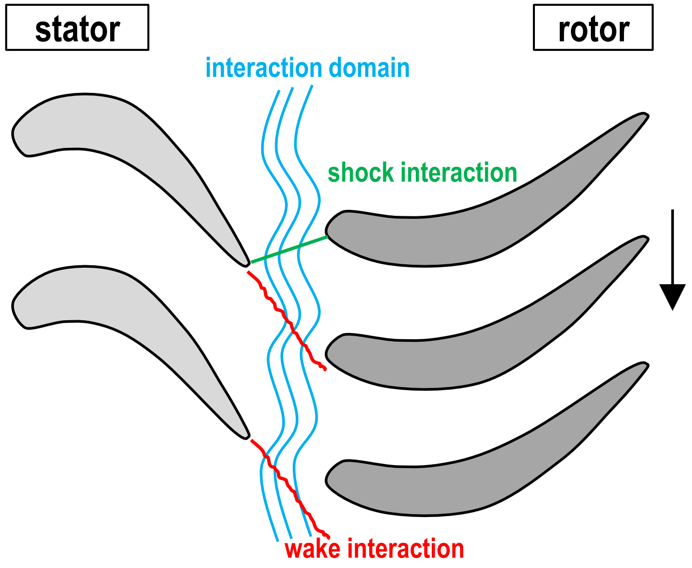
Diffusor $f_{dif} = k_R\ z_R\ f_R$

Impeller $f_{imp} = k_D\ z_D\ f_R.$

$$n = k_R\, z_R \pm k_D\, z_D$$

k_R, k_D harmonic orders associated with rotor and diffuser
z_D the number of diffuser vanes
z_R the number of compressor blades [Hz]

$$\varepsilon_S = \frac{f_{RSI}}{f_R} \doteq 0.20\text{–}0.50$$



[3] Marshall F. and J. Sorokes, A Review of Aerodynamically Induced Forces Acting on Centrifugal Compressors, and Resulting Vibration Characteristics of Rotors, Proceedings of the 29th turbomachinery symposium, pp. 263-280, 2000.

Compressor Aerodynamic Noise under Design Operating Conditions

- **Tip clearance noise (TCN) according Raitor and Neise[2], for $\mathrm{M_a} < 0.95$ and $\theta > 1$**

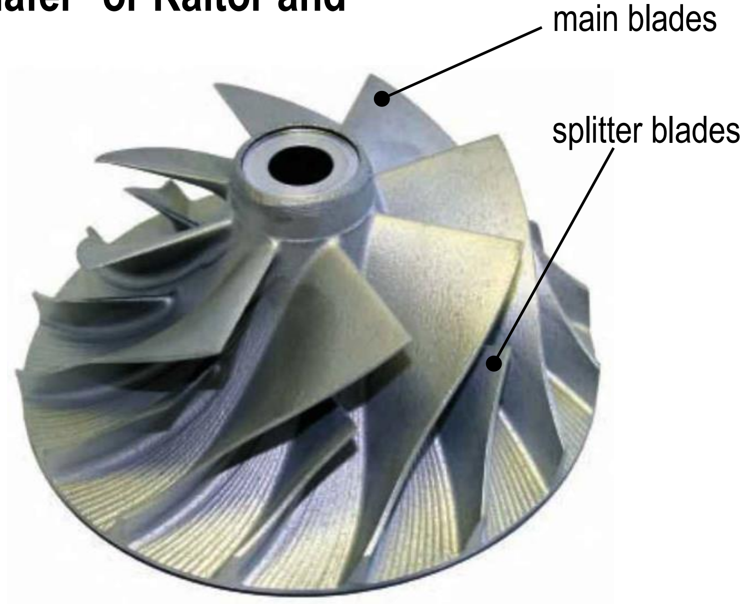
$$\varepsilon_{\mathrm{R}} = \frac{f_{\mathrm{TCN}}}{z_R f_{\mathrm{R}}} \doteq 0.36 - 0.64$$

M_{a} – Mach number $[-]$ f_{R} – rotor speed frequency $[\mathrm{Hz}]$ f_{TCN} – TCN frequency $[\mathrm{Hz}]$ f_{BPF} – blade passing frequency $[\mathrm{Hz}]$ ε – integer frequency ratio [-]

- **Rotational noise or also buzz-saw noise according Nguyen-Schäfer[1] or Raitor and Neise [2], for $\mathrm{M_a} > 0.95$ and $\theta > 1$**

$$\varepsilon_{\mathrm{R}} = \frac{\varepsilon f_{\mathrm{BPF}}}{f_{\mathrm{R}}} = \varepsilon z_R$$

$$\varepsilon = 1,2,3, \ldots$$



[1]Nguyen-Schäfer, H. Rotordynamics of Automotive Turbochargers. Second Edition. Ludwigsburg, Germany: Springer, 2015.
[2]Raitor T. and W. Neise. Sound generation in centrifugal compressors. Journal of Sound and Vibration, vol. 314, pp. 738-756, 2008.

Compressor Noise under Off-Design Operating Conditions

❑ **Impeller Rotating Stall (IRS) – flow separation characterized by rotating stall cells**

Mild IRS:

$$\varepsilon_{\mathrm{R}} = \frac{f_{\mathrm{Mild\ IRS}}}{f_{\mathrm{R}}} \doteq 0.60 - 0.75$$ Frigne and Van Den Braembussche[4]

$$A = 0.07,\ \theta \cong 1$$

Progressive IRS:

$$\varepsilon_{\mathrm{R}} = \frac{f_{\mathrm{Prg\ IRS}}}{f_{\mathrm{R}}} \doteq 0.67 - 0.82$$ Frigne and Van Den Braembussche[4]

$$\varepsilon_{\mathrm{R}} = \frac{f_{\mathrm{Prg\ IRS}}}{f_{\mathrm{R}}} \doteq 0.50 - 0.80$$ Liśkiewicz et al.[5]

$$A = 0.1 - 0.3,\ \theta \cong 1$$

Abrupt IRS:

$$\varepsilon_{\mathrm{R}} = \frac{f_{\mathrm{Abr\ IRS}}}{f_{\mathrm{R}}} \doteq 0.26 - 0.93$$ Frigne and Van Den Braembussche[4]

$$\varepsilon_{\mathrm{R}} = \frac{f_{\mathrm{Abr\ IRS}}}{f_{\mathrm{R}}} \doteq 0.20 - 1.20$$ Liśkiewicz et al.[5]

$$A = 0.1 - 0.3,\ \theta \cong 1$$

[4] Frigne P. and R. Van Den Braembussche, Distinction Between Different Types of Impeller and Diffuser Rotating Stall in a Centrifugal Compressor With Vaneless Diffuser, J. Eng. Gas Turbines Power, vol. 106, no. 2, pp. 468-474, 1984.
[5] Liśkiewicz G., L. Horodko, M. Stickland and W. Kryłowicz, Identification of phenomena preceding blower surge by means of pressure spectral maps, Experimental Thermal and Fluid Science, no. 54, pp. 267-278, 2014.

Compressor Aerodynamic Noise under the Off-Design Conditions

- **Diffuser Rotating Stall (DRS)** for $\theta \cong 1$

DRS:	$\varepsilon_{\mathrm{R}} = \dfrac{f_{\mathrm{DRS}}}{f_{\mathrm{R}}} \doteq 0.06\text{–}0.99$	Marshall and Sorokes[3]
Low speed DRS:	$\varepsilon_{\mathrm{R}} = \dfrac{f_{\mathrm{Low\ speed\ DRS}}}{f_{\mathrm{R}}} \doteq 0.26\text{–}0.32$	Frigne and Van Den Braembussche[4]
High speed DRS:	$\varepsilon_{\mathrm{R}} = \dfrac{f_{\mathrm{high\ speed\ DRS}}}{f_{\mathrm{R}}} \doteq 0.51\text{–}0.63$	Frigne and Van Den Braembussche[4]

$$A = 0.07$$

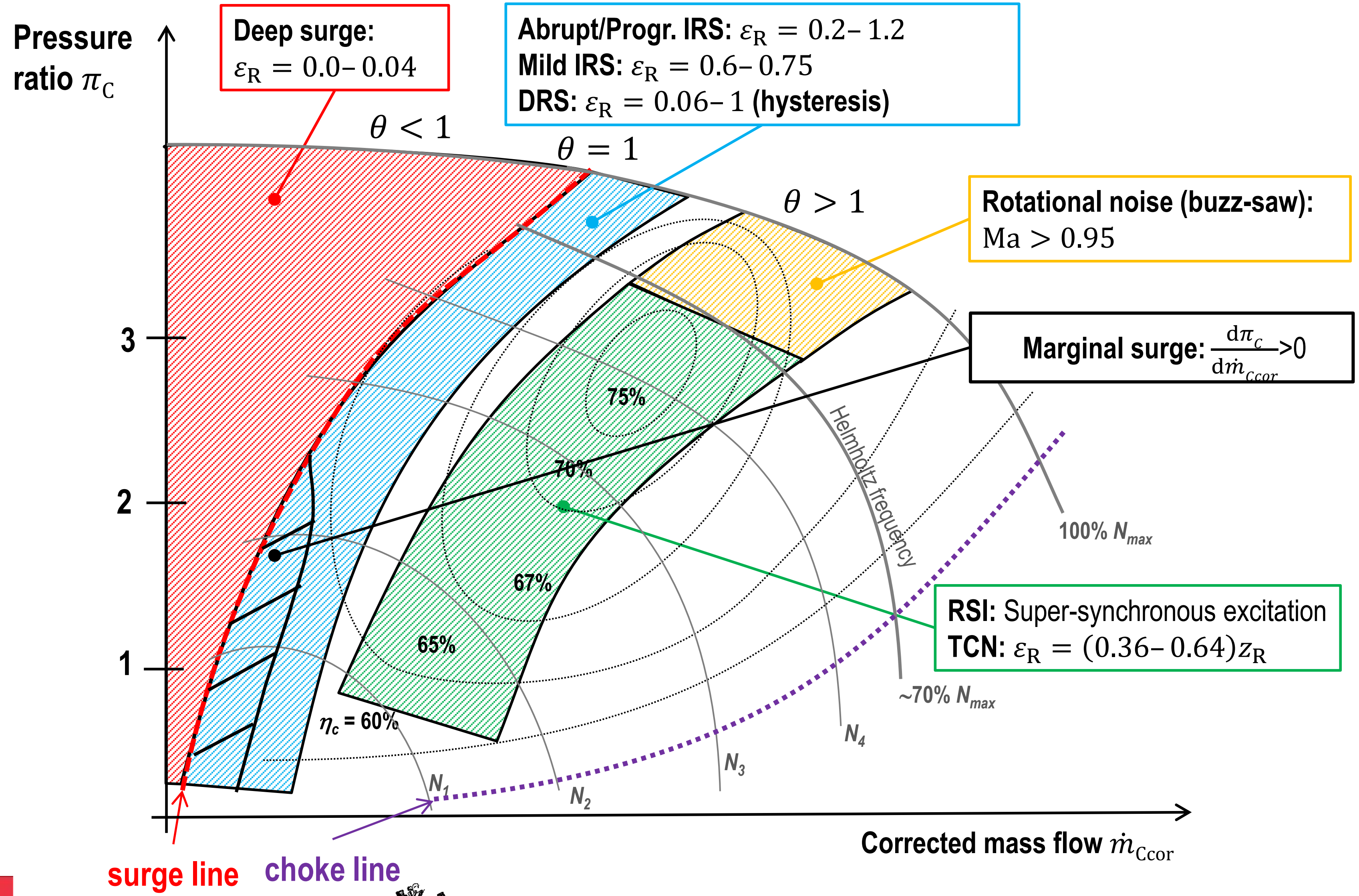
- **Whoosh Noise**, denoted also as **surge noise** for $\theta < 1$

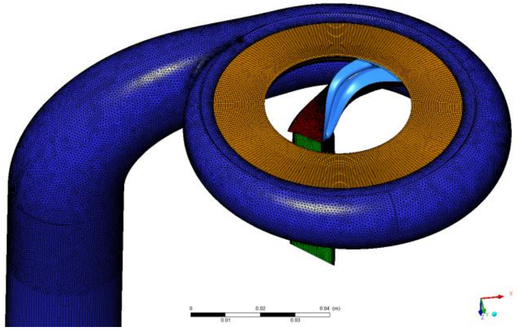
$\varepsilon_{\mathrm{R}} = \dfrac{f_{\mathrm{surge}}}{f_{\mathrm{R}}} \doteq 0.90$	Kämmer and Rauetenberg[6]
$\varepsilon_{\mathrm{R}} = \dfrac{f_{\mathrm{surge}}}{f_{\mathrm{R}}} \doteq 0.80\text{–}0.90$	Lawless and Fleeter[7]
$\varepsilon_{\mathrm{R}} = \dfrac{f_{\mathrm{surge}}}{f_{\mathrm{R}}} \doteq 0.70\text{–}1.70$	Torregrosa et al.[8]

$$A = 1$$

[6] Kämmer N. and M. Rautenberg, An Experimental Investigation of Rotating Stall Flow in a Centrifugal Compressor, ASME 1982 International Gas Turbine Conference and Exhibit: Paper No: 82-GT-82, vol. 1, pp. 1-9, 1982.
[7] Lawlesss P. and S. Fleeter, Prediction of Active Control of Subsonic Centrifugal Compressor Rotating Stall, AIAA JOURNAL, vol. 35, no. 12, pp. 1829-1836, 1997.
[8] Torregrosa A., A. Broatch, R. Navarro and J. Jorge García-Tíscar, Acoustic characterization of automotive turbocompressors, International Journal of Engine Research, vol. 16, no. 1, pp. 31-37, 2014.

Compressor Map describing Aerodynamic Noise of Turbochargers





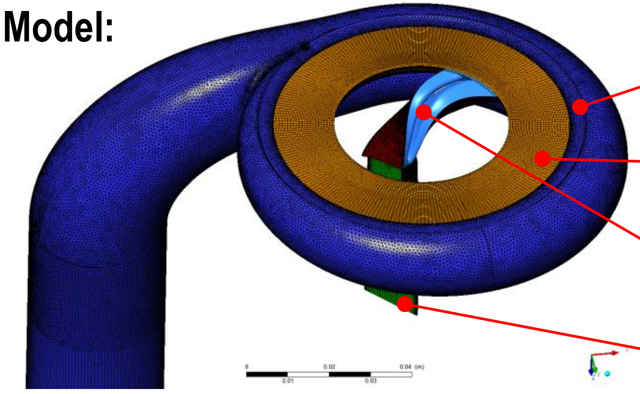
Development of Compressor Design Enabling the Reduction of Aerodynamic Bored Noise

Improvements in Vibroacoustic Performance – Review of Activities and Results under the WP 05 Advanced Turbochargers for Diesel Engine Applications

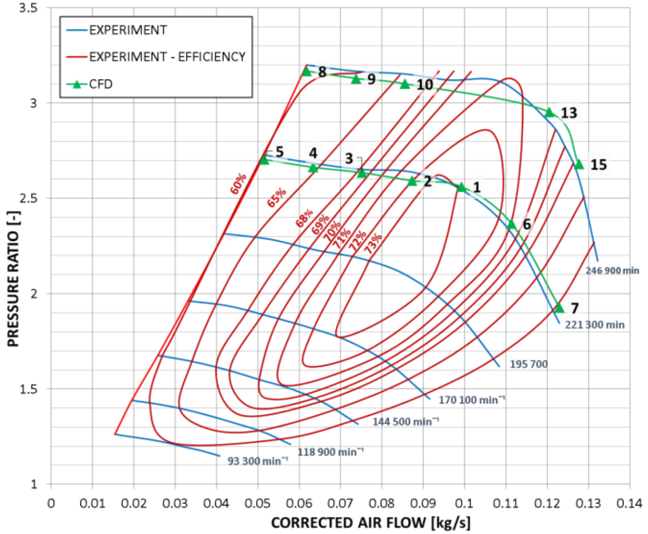
Model Description

- CFD Model is created in software CFX.
- Several operational conditions of the compressor characteristic were analysed.
- Correlation with experimental data was done.
- Steady-state and transient analyses were performed.
- Searching for noise sources was performed with the aim to propose a geometric modification leading to suppression of the generated noise.

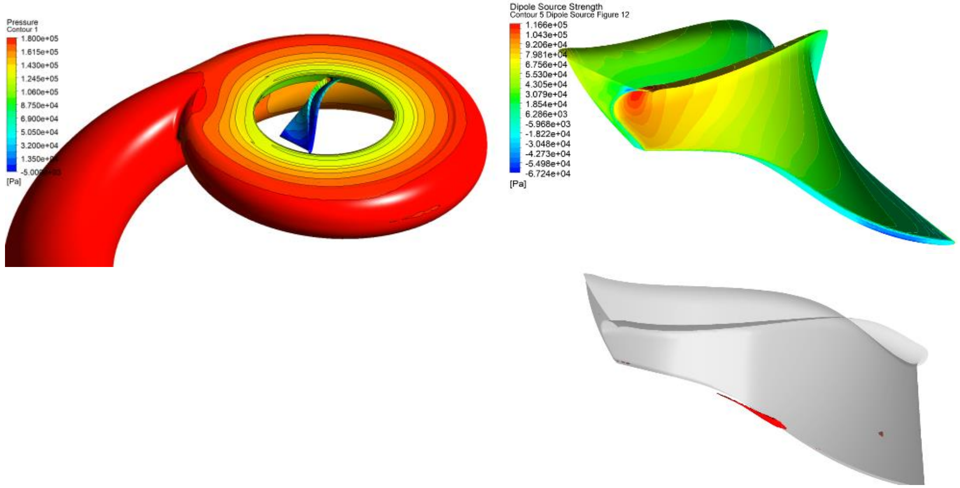
CFD Model:



Calculated results connected to compressor map:



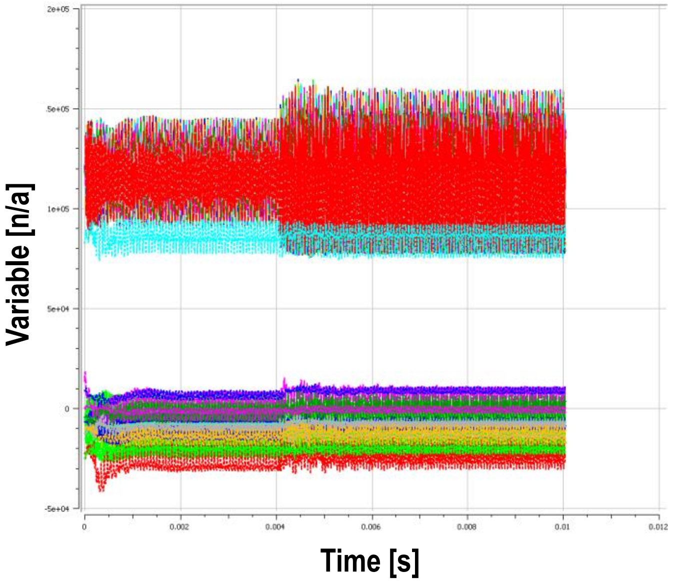
CFD Results:



Evaluation of Aerodynamic Noise Calculations

- Transient analysis included more than 30 impeller revolutions.
- Frequency range approximately 300 Hz–100 000 Hz was captured $\Rightarrow \varepsilon_{\mathrm{R}} = 0.08\text{–}36.$
- A geometric modification was created based on the CFD results including both acoustic analogy and direct evaluation of sound parameters from transient analysis.

Monitored quantities (pressures and velocities): → **Acoustic power vs. time**



Matlab:

For each node of control surface S and for each analysed time step the fluctuating component was obtained to compute acoustic power

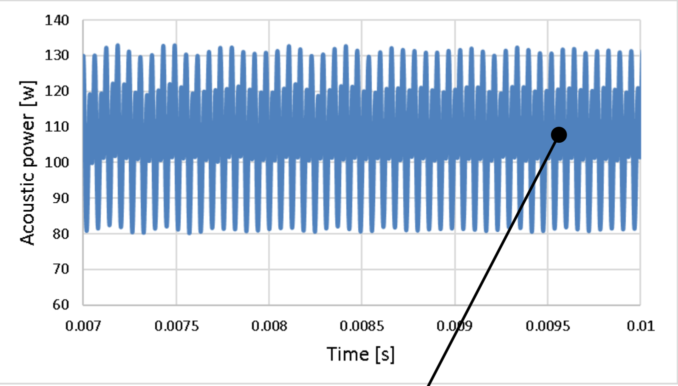
$$P(time) = \iint_S p' \, v_i' \, n_i' \; dS,$$

where p', v_i' and n_i' is fluctuating pressure, fluctuating velocity and normal vector respectively. i=1, 2, 3.

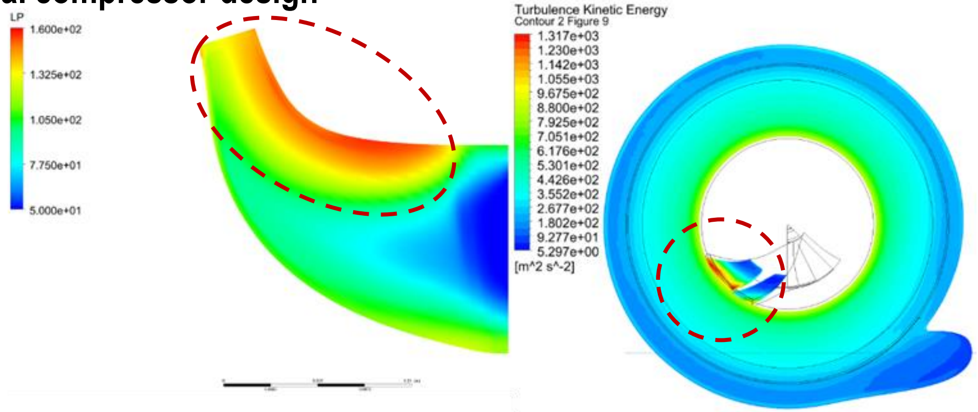
Over 300 million values were post-processed for one operational point.

Evaluation of Compressor Design Changes

- Proposed modification showed acoustic power improvement.
- Suppression of buzz–saw noise is expected with even positive impact on the compressor efficiency.
- Sound power level decrease by approximately 3 dB may be expected.

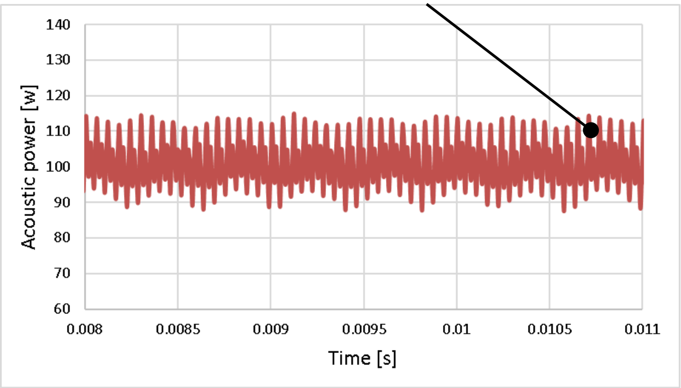


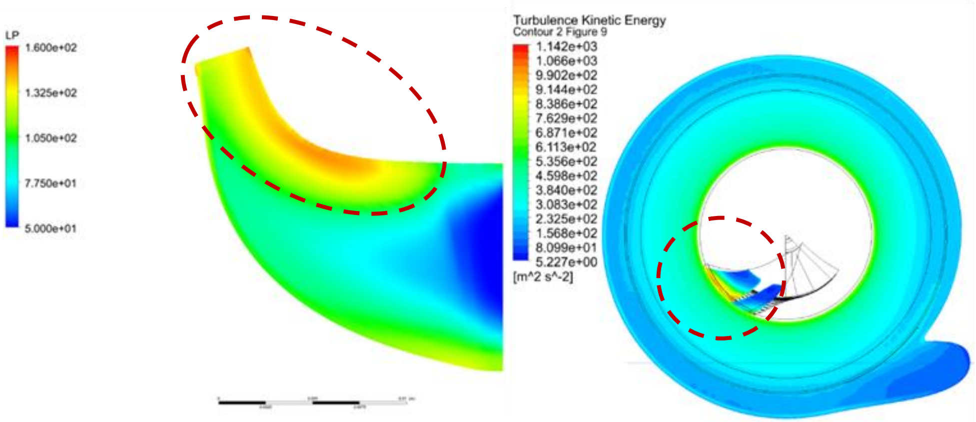
Nominal compressor design



Both mean value and amplitude were decreased.

Modified compressor design





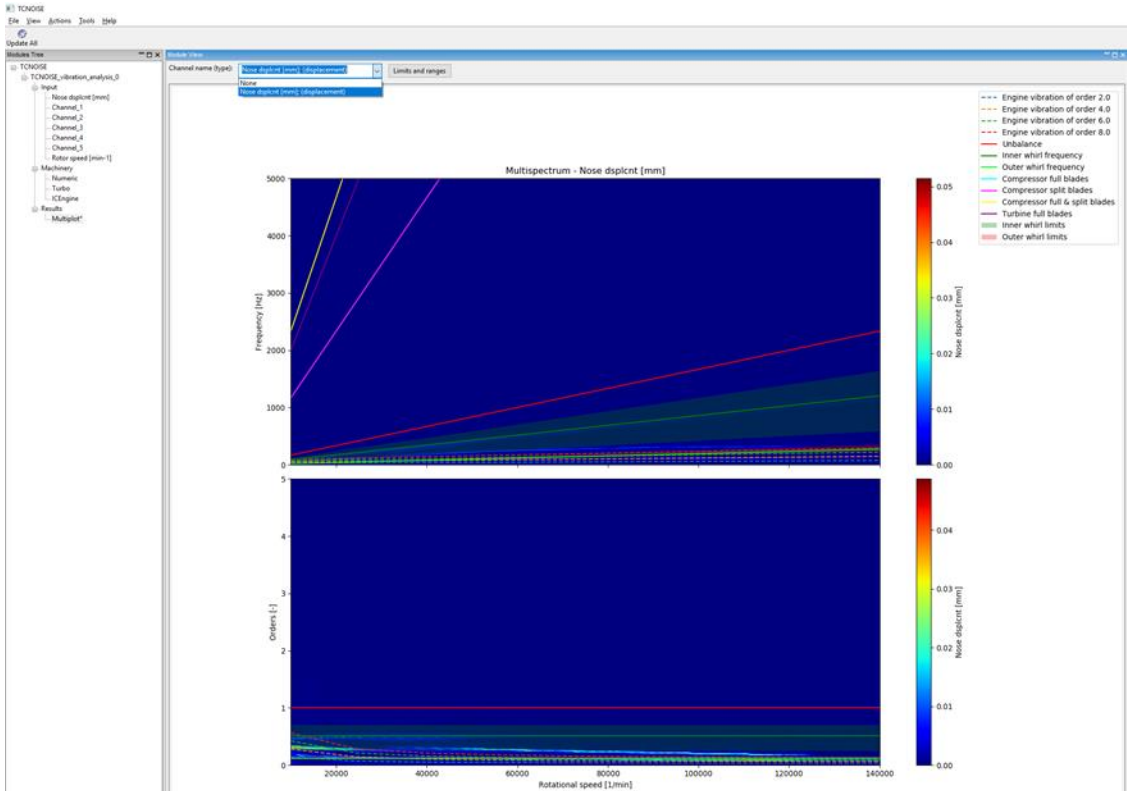
Results of Activities

Analytical Tool for the Identification of Vibration and Noise Sources in Turbochargers

TC NOISE Software

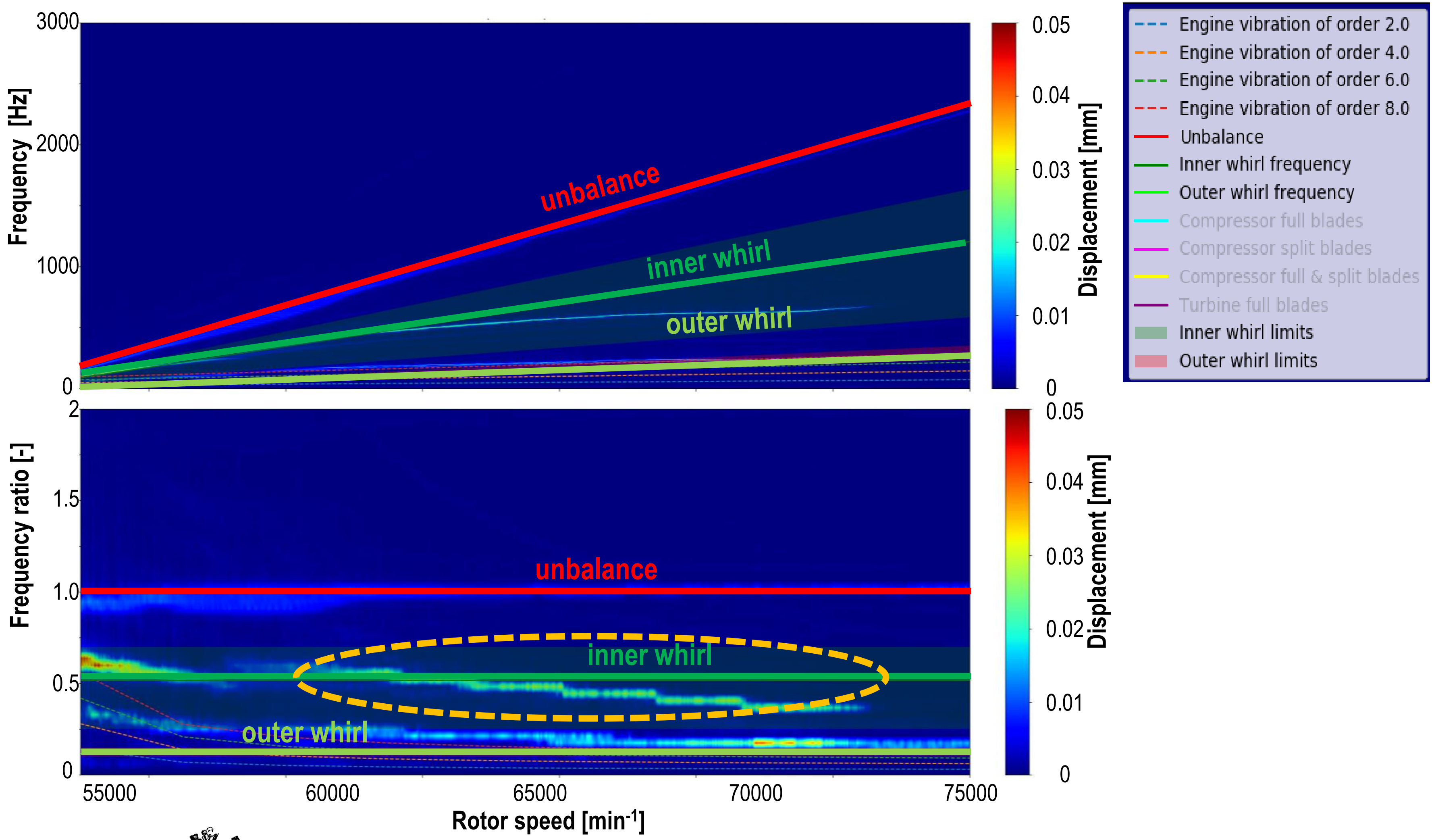
- Analytical methodology is implemented in the software TC NOISE, release 2.0
- Software capabilities: The analytical tool can identify probable vibroacoustic sources based on measurements (kinematic and acoustic quantities).
- Software restrictions: Some processes may overlap in frequency. Subsequently, more advanced tools such as FEM, CFD, Multibody etc. can be used to identify the issue.
- TCNOISE v. 2.0 software is free for non-commercial use.

GUI of TCNOISE v. 2.0



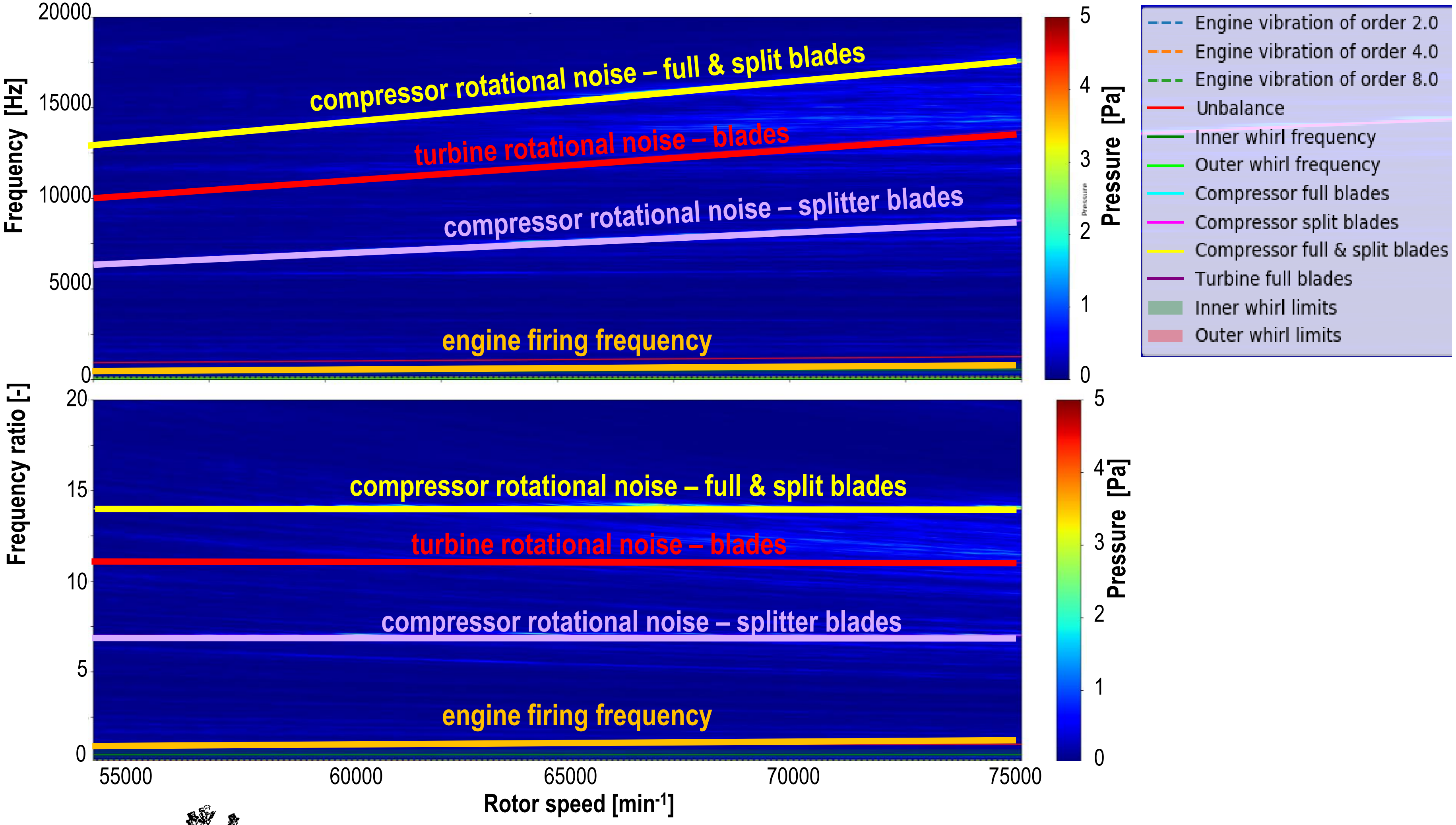
Example of Use: Vibration of TC Rotor Nose of Tractor Diesel Engine

- Distance measurement by eddy current sensors



Example of Use: Noise of TC Compressor of Heavy-Duty Vehicle Diesel Engine

- Noise measurement by acoustic camera



Acknowledgement

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Thank you for your attention

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