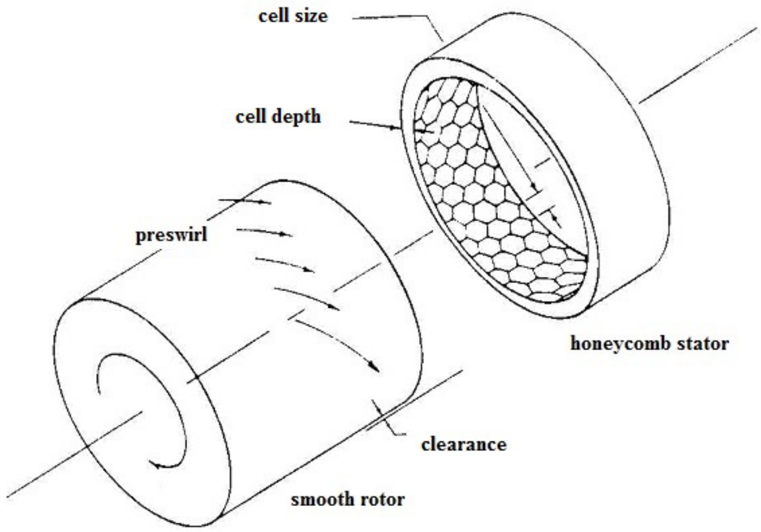
Honeycomb Seals in MADYN 2000 Version 4.6

In version 4.6 Honeycomb seals as shown in the following sketch were introduced (see /1/). They have a special characteristic, which cannot simply be modelled by rotordynamic coefficients as other seals. Hole pattern seals have a similar behaviour. In the following the rotordynamic behaviour of an example of a compressor with honeycomb seals is described. The description for the modelling of the seal can be found in the documentation of version 4.6 in chapter II.15 (see /1/). It is not subject of this document.



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1. Description of the Rotor

The rotor with the seals (green symbols) is shown in figure 1.1. The seals with the largest influence are considered in the example, they are the labyrinth seals with the highest pressure in section 1 and 2 of the compressors, the interstage honeycomb seal between section 1 and 2 as well as the honeycomb balance piston seal.

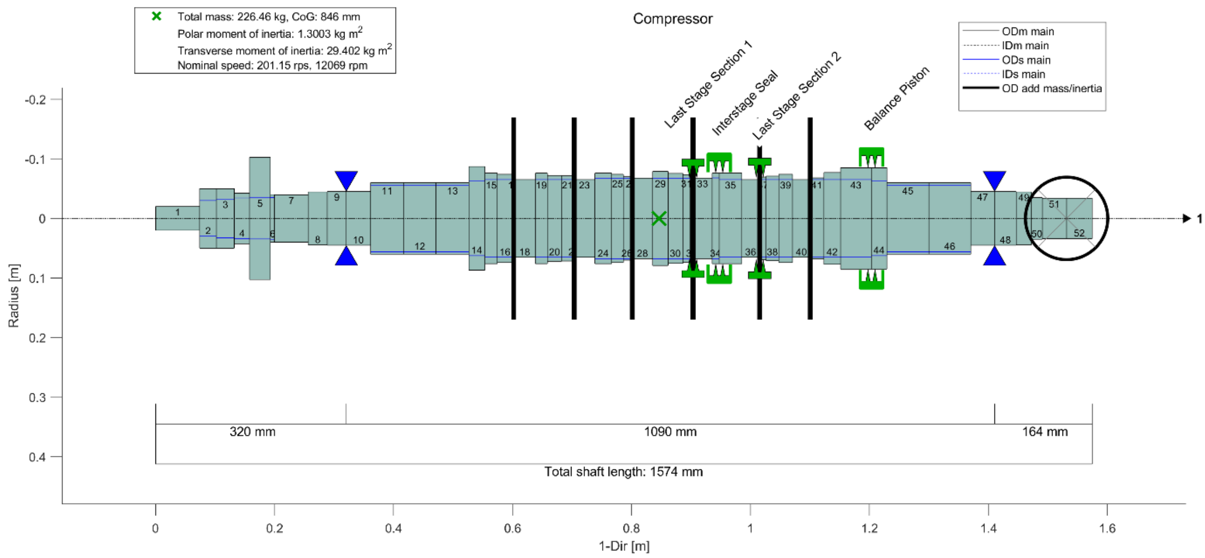


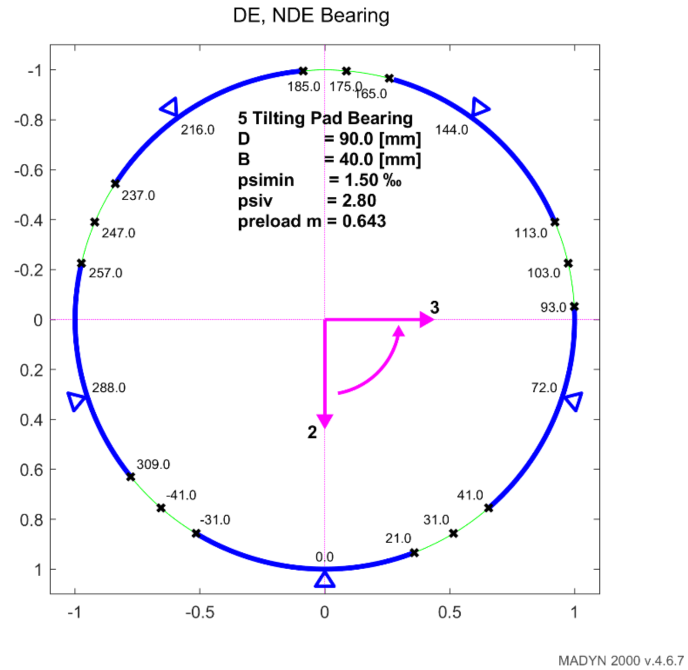
Fig. 1.1: Compressor rotor with seals

1.1 Bearings

The main bearing data are listed in table 1.1. The bearings are 5 tilting pad bearings with load on pad as shown in figure 1.2.

Table 1.1: Main bearing data

	Bearings (DE and NDE identical)
Bearing type	5-Tilting pad, 60% offset, load on pad-pivot
Diameter D	90 mm
Width B	40 mm
Relative Width B/D	0.43
Rel. clearance Ψ_{min}	1.5 ‰
Preload	0.643
Oil inlet temperature (min. / nominal / max.)	45°C
Peripheral speed at N_{mc}	56.9 m/s
Bearing load due to weight (see figure 2.1)	DE: 1150 N (3.2 bar) NDE: 1072 N (3.0 bar)



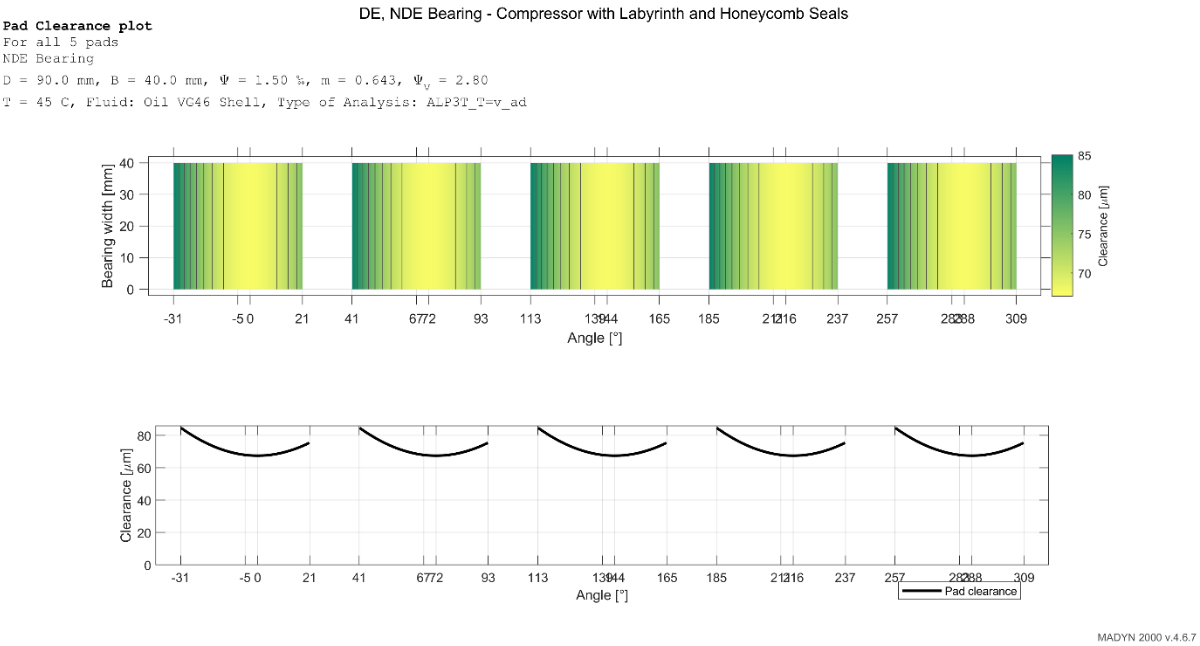


Fig. 1.2: Bearings, geometry and clearance for centred position

1.2 Seals

The main data of the labyrinth and honeycomb seals are listed in table 1.2. In the table the data at nominal speed are shown. Speed dependent characteristics were considered, assuming that the pressure difference is a quadratic function of speed.

Table 1.2: Main seal data

	Section 1 Laby. Seal	Section 2 Laby. Seal	Honeycomb Interstage	Honeycomb Bal. Piston
Type	See through, stator	See through, stator	Cell depth 2mm	Cell depth 2mm
Number of laby. strips, length [mm]	6, 28	6, 28	70	80
Diameter [mm]	230	230	170	203
Diametral clearance [mm]	0.2	0.2	0.4	0.4
Relative inlet swirl*	70%	70%	70%	70%
Molecular weight [kg/kmol]	43.4	43.4	43.4	43.4
Inlet pressure at nom. speed [bar]	87.8	150	150	87.8
Outlet pressure at nom. speed [bar]	62	115	87.8	22.7

* Swirl in front of the seal assuming that there are no swirl brakes.

The linear tangential and radial forces on the rotor per displacement at nominal speed for a circular orbit with different whirling frequency are shown in figure 1.3 for the honeycomb seals and in figure 1.4 for the labyrinth seals.

The characteristic for the honeycomb seals in figure 1.3 calculated according to the theory described in /2/ cannot be described by stiffness, damping and mass coefficients as for other seals. They are captured with polynomials in the Laplace domain (see /1,3/).

For the labyrinth seals the forces are calculated with a CFD analysis at 4 frequencies. They are marked with a cross in figure 1.4. From these results stiffness, damping and mass coefficients are determined /1,2/. The solid curves in the figure show the forces calculated with these coefficients. The dashed curves show the forces just using stiffness and damping coefficients calculated form the result of 2 whirling frequencies.

In the figures 1.3 and 1.4 the Swirl Frequency Ratios SFR are highlighted. It is the frequency relative to the speed with a zero tangential force. The SFR is measure for the stability: modes with a frequency relative to the speed lower than the SFR are destabilised, those with a higher relative frequency are stabilised.

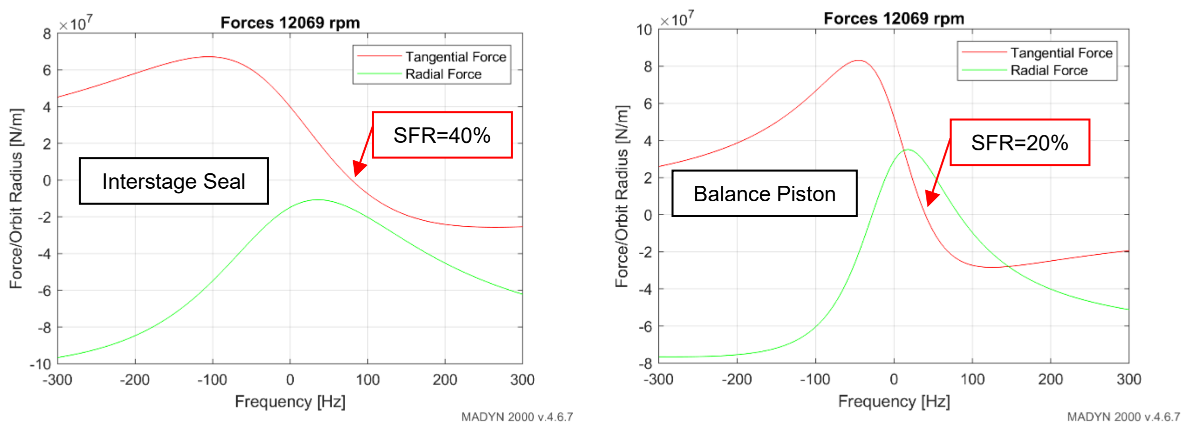
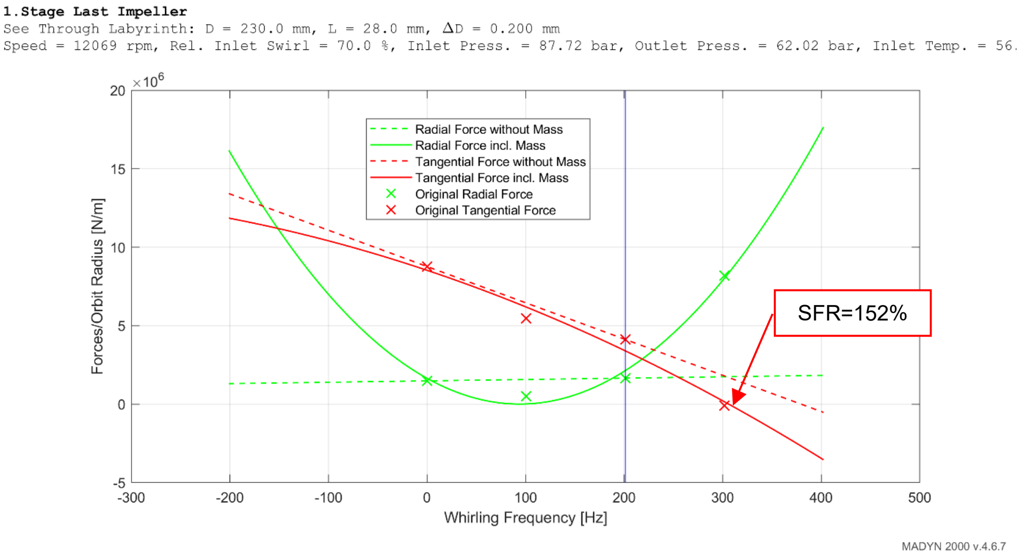


Fig. 1.3: Radial and tan. forces of the honeycomb seals on the rotor for circular whirling, 100% speed



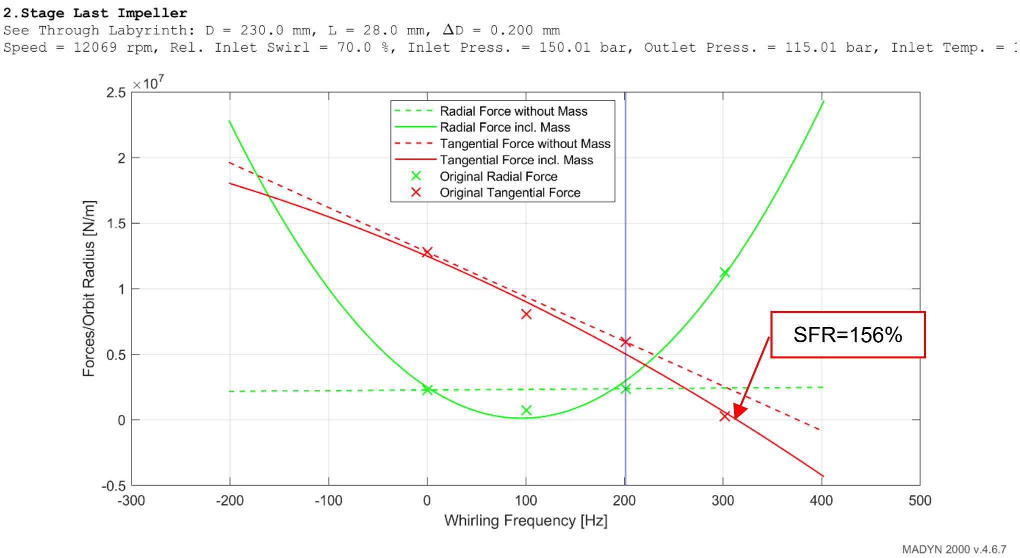


Fig. 1.4: Rad. and tan. forces of the labyrinth seals on the rotor for circular whirling, 100% speed

2. Static Analyses

The rotor deformation and the forces at nominal speed including the bearing forces can be seen in figure 2.1 without the influence of any seals (unloaded condition). In figure 2.2 they are shown with the influence of the labyrinth seals only and in figure 2.3 with the influence of all seals (loaded condition).

The rotor is supported on 2 bearings, thus it would be statically determined, if the seals do not carry any load. However, as can be seen from the results the honeycomb seals carry a large portion of the weight load. For this reason, a hyperstatic nonlinear[1] static analysis must be carried out to get the correct results and bearing loads, which determine the speed and load dependent bearing characteristics.

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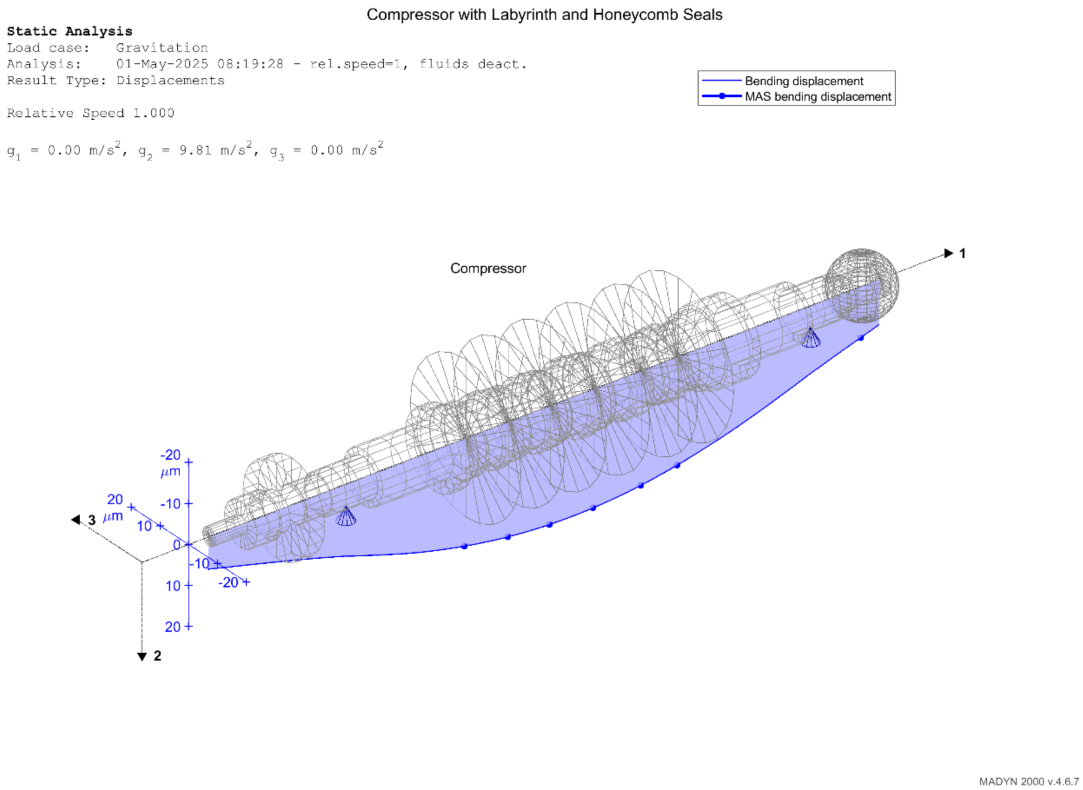
[1] Bearings are nonlinear, for the seals the linear stiffness is used.



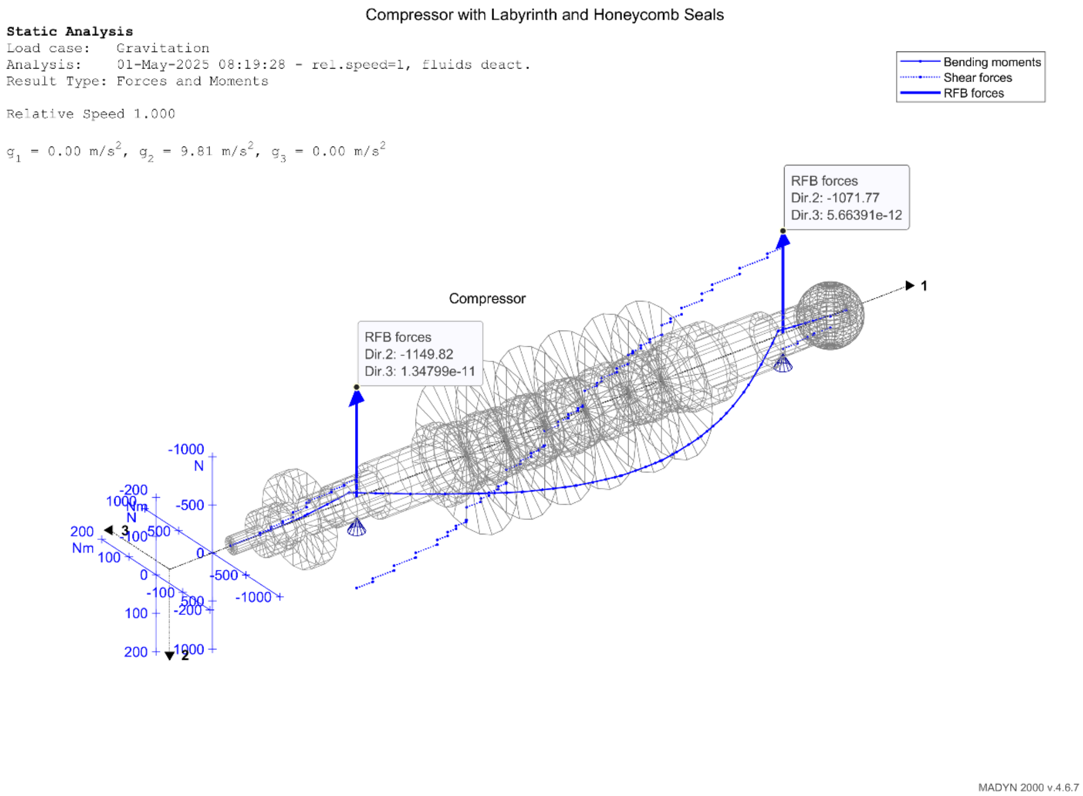
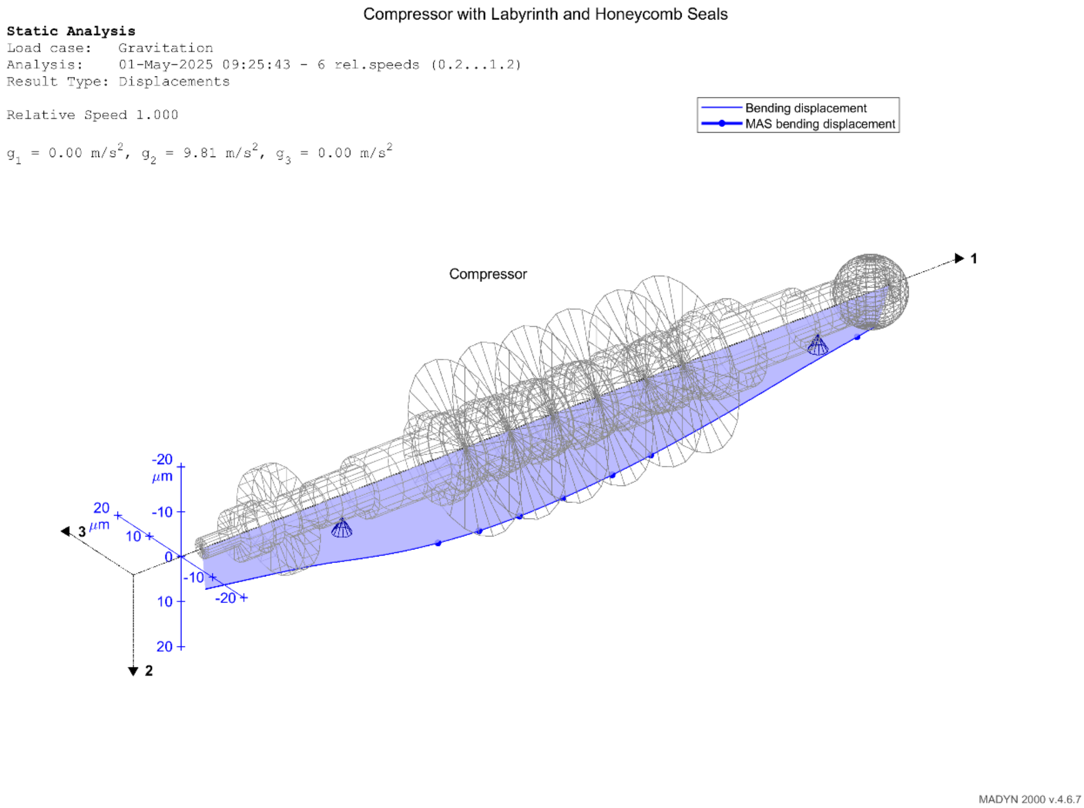


Fig. 2.1: Static displacement and forces at 100% speed without seals (no load condition)



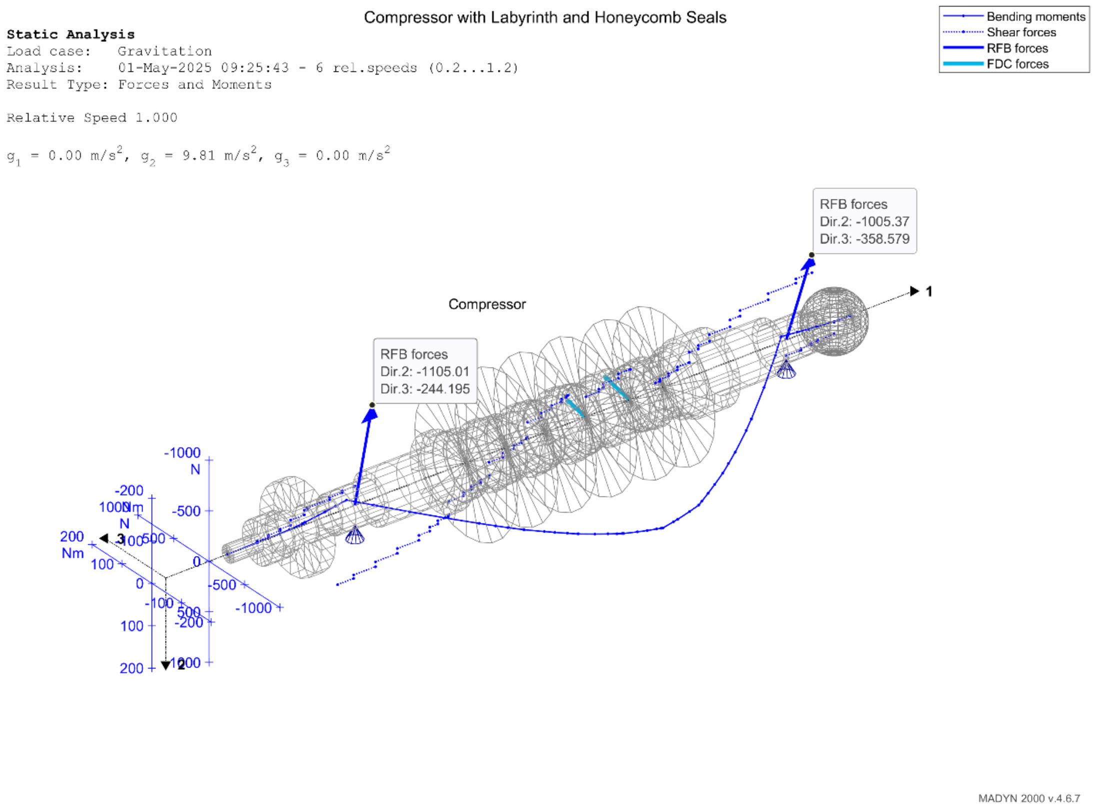
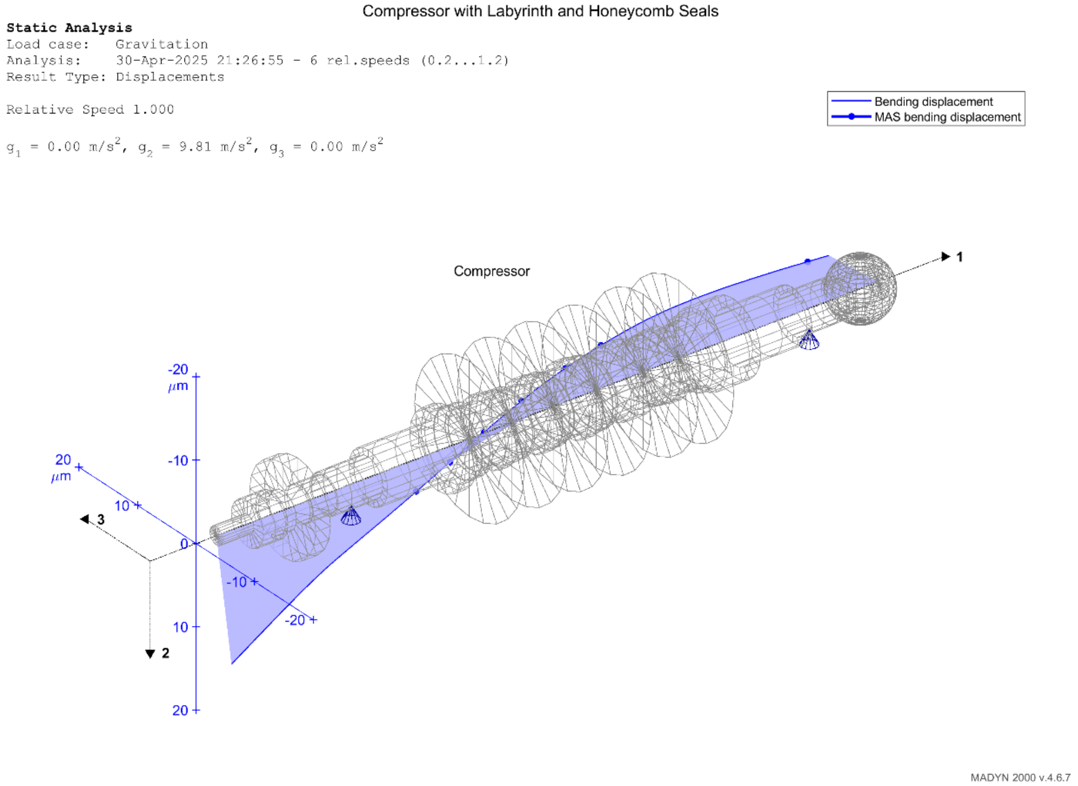


Fig. 2.2: Static displacement and forces at 100% speed with labyrinth seals only



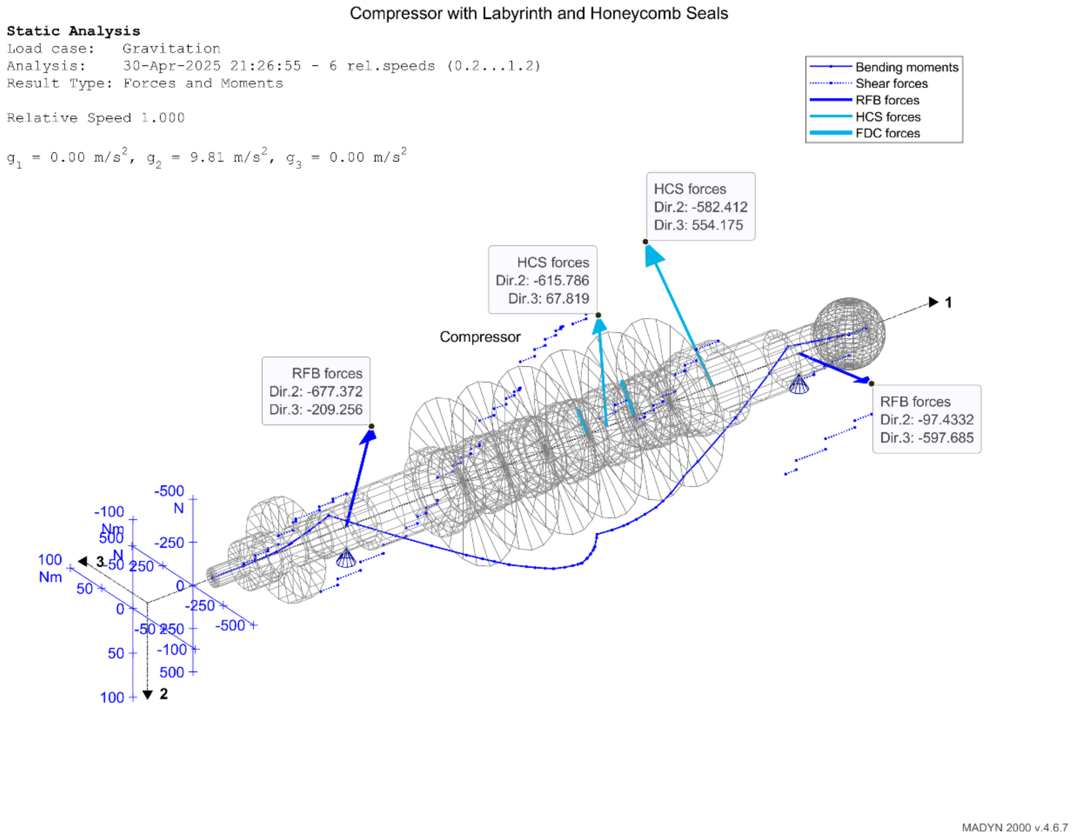


Fig. 2.3: Static displacement and forces at 100% speed with all seals (loaded condition)

3. Campbell Diagrams, Stability

3.1 Campbell Diagram without Seal Influence

In figure 3.1 the static results of the bearing analyses are shown for the bearing loads without seal influence, i.e. corresponding to the bearing forces with only the bearings carrying the weight load. The corresponding rotordynamic stiffness and damping coefficients are shown in figure 3.2.

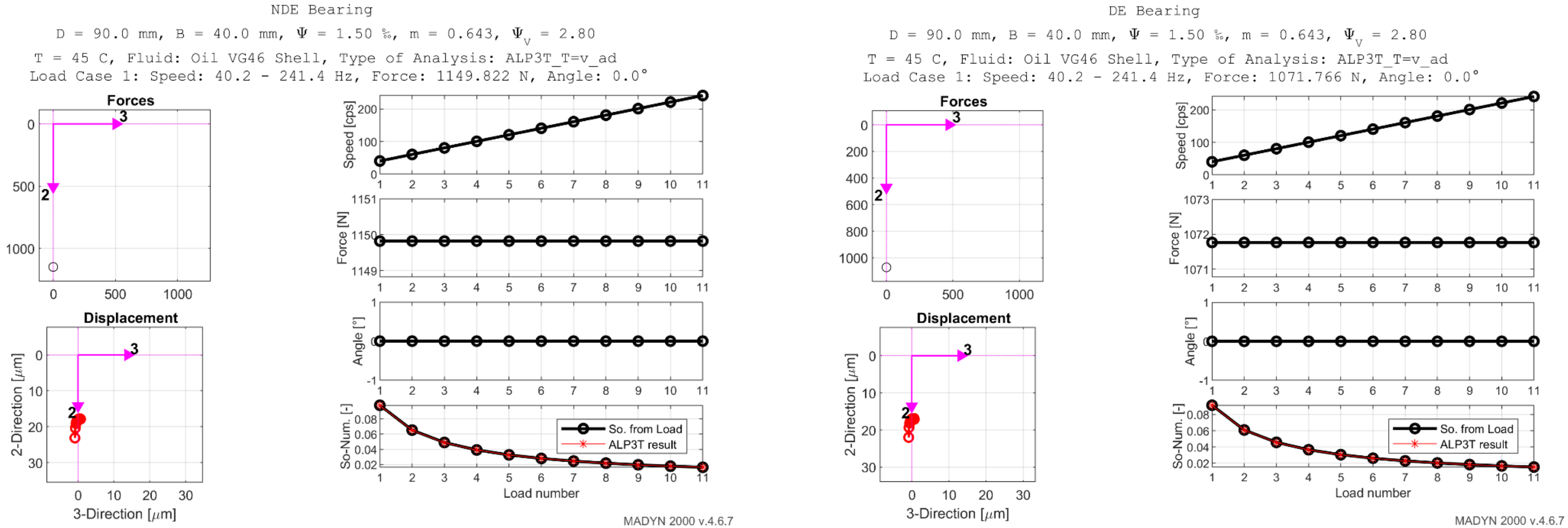


Fig. 3.1: Static results of the bearing analysis without seal influence

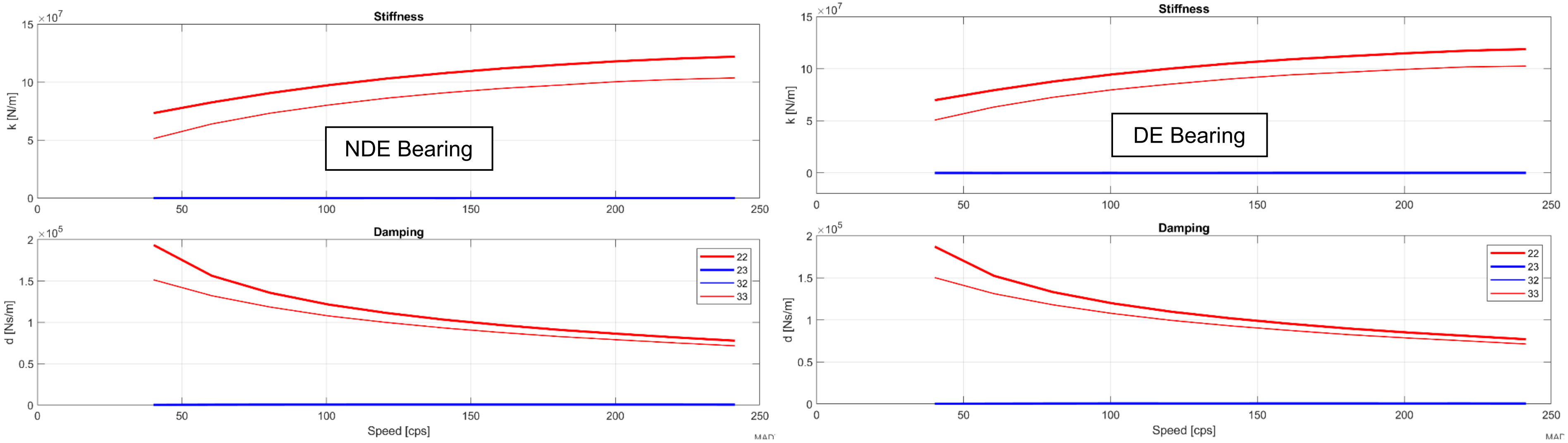


Fig. 3.2: Rotordynamic stiffness and damping coefficients for the static loads without seal influence

The resulting Campbell diagram can be seen in figure 3.3. The first critical speeds (horizontal and vertical bending) are at 6’923 and 7’171 rpm, respectively, i.e. at 57% and 59% of the nominal speed. They have a rather good damping of 18.7% and 15.2% damping ratio. The damping of these modes drops to 11.9% and 10.5% at nominal speed. The tilting modes in the vicinity of the nominal speed have high damping ratios above 40%.

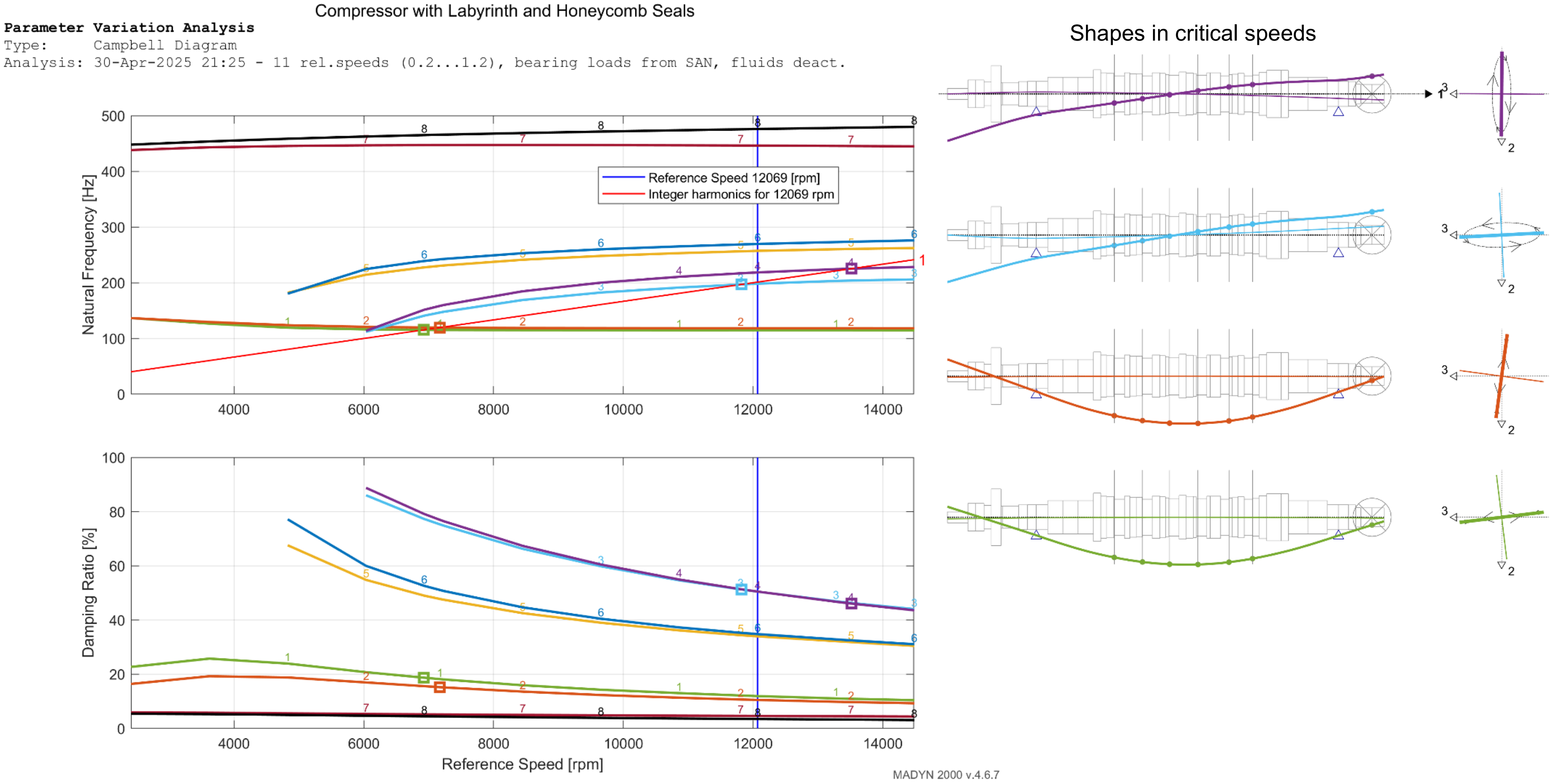


Fig. 3.3: Campbell diagram without seals with mode shapes in critical speeds (no load condition)

3.2 Stability at Nominal Speed

The eigenvalues (natural frequency and damping ratio) at nominal speed for the following 3 conditions are shown in figure 3.4:

- No seal influence (unloaded condition)
- Only labyrinth seals are considered
- All seals are considered (loaded condition)

The mode shapes for the case without seals and with all seals are shown in figure 3.5. With the seals they become more circular.

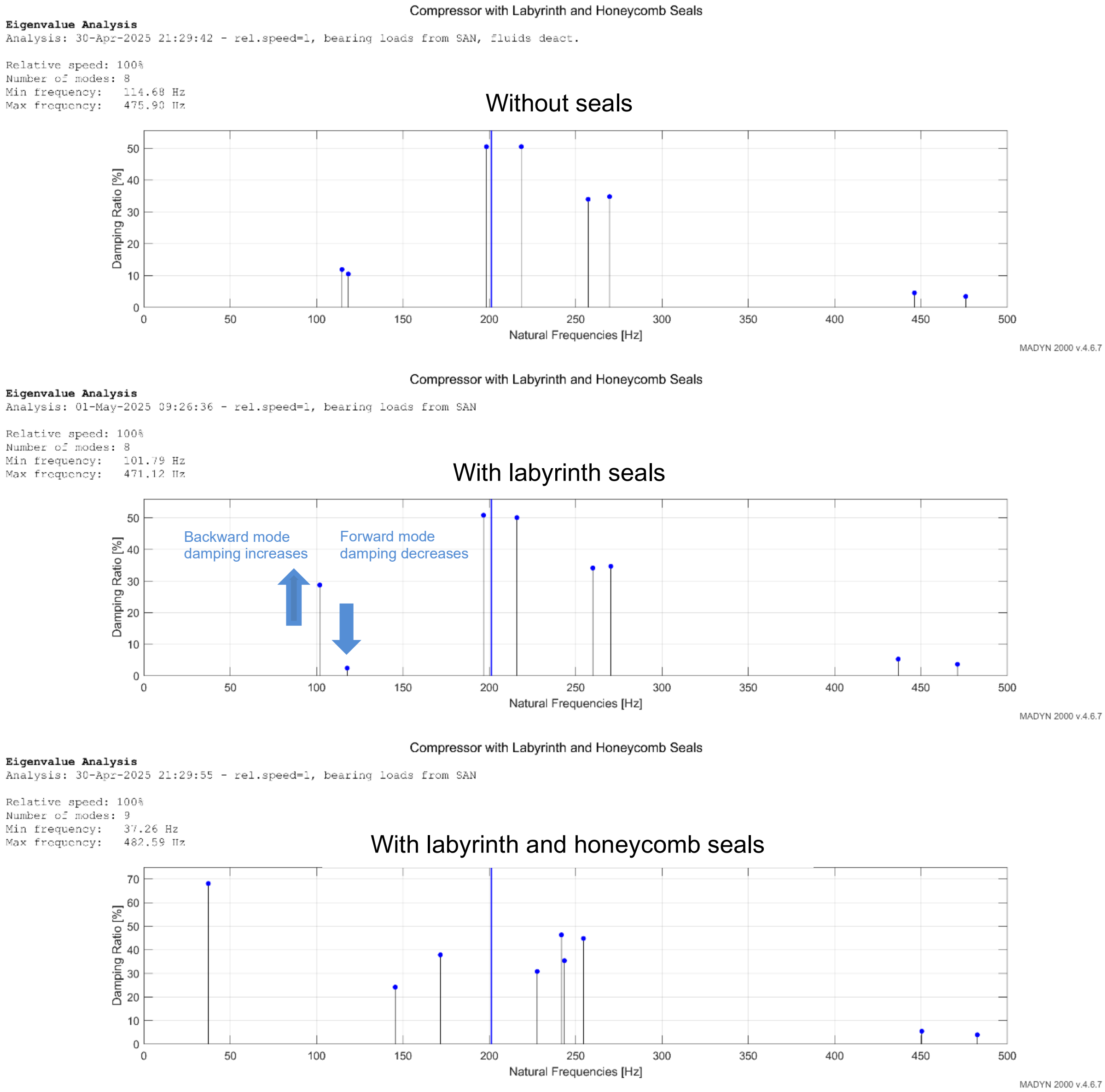


Fig. 3.4: Eigenvalues at 100% speed, without seals, with labyrinth seals only, with all seals

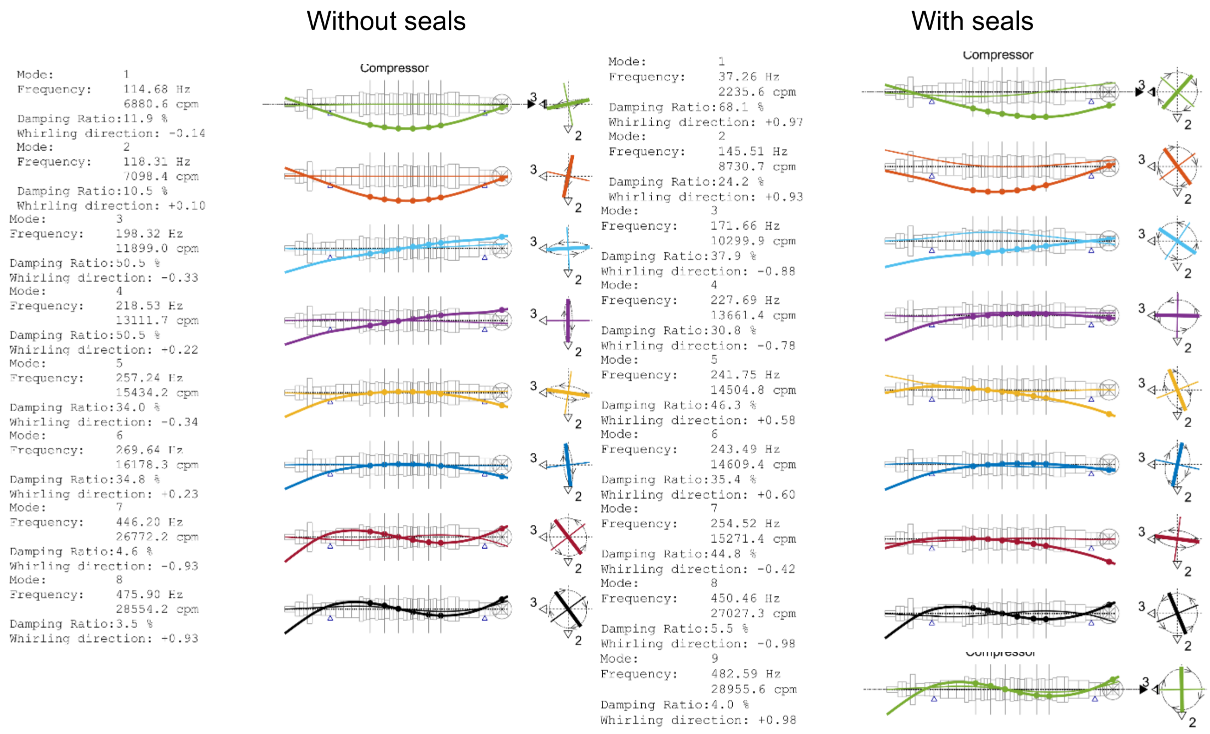


Fig. 3.5: Natural modes at 100% speed, with and without seals

A comparison of the eigenvalues without seals and with labyrinth seals shows the that the 1st bending forward mode is destabilised by the labyrinth seals and the backward mode stabilised. This can be expected due to the high SFR of the labyrinth seals. The natural frequency of the bending modes changes to some extent (lower frequency for the backward mode).

For the case that all seals are considered all modes have a higher damping and natural frequencies shift considerably due to the influence of the honeycomb seal. Especially the first bending mode increases, which is the reason why it is much better damped.

3.3 Campbell Diagram with Seals

In figure 3.6 the static results of the bearing analyses are shown considering all seals. As can be seen the forces are considerably lower and change direction with the speed, due to the influence of the seals, which also carry part of the weight load. The corresponding rotordynamic stiffness and damping coefficients are shown in figure 3.7. They are lower than without seal influence and more isotropic. The results at nominal speed were also used in the stability analysis described in chapter 3.2. The coefficients are calculated automatically in MADYN 2000 when needed, using the speed dependent static loads determined in the static analysis (see chapter 2, where the results for nominal speed are shown).

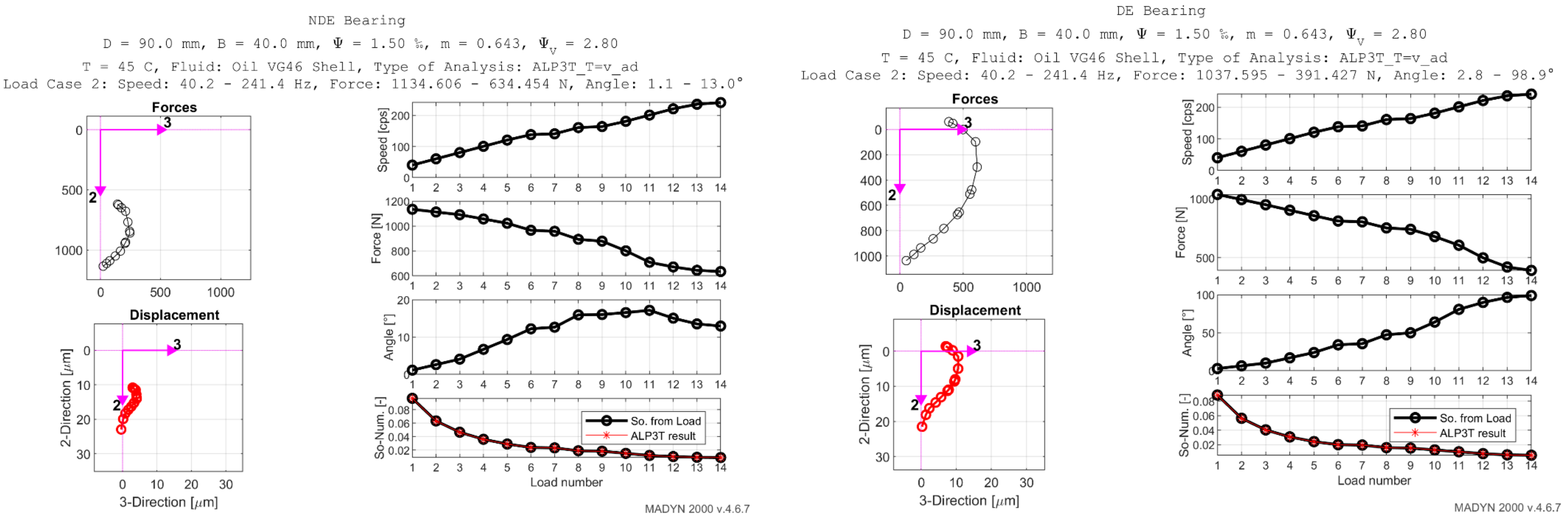


Fig. 3.6: Static results of the bearing analysis with seal influence

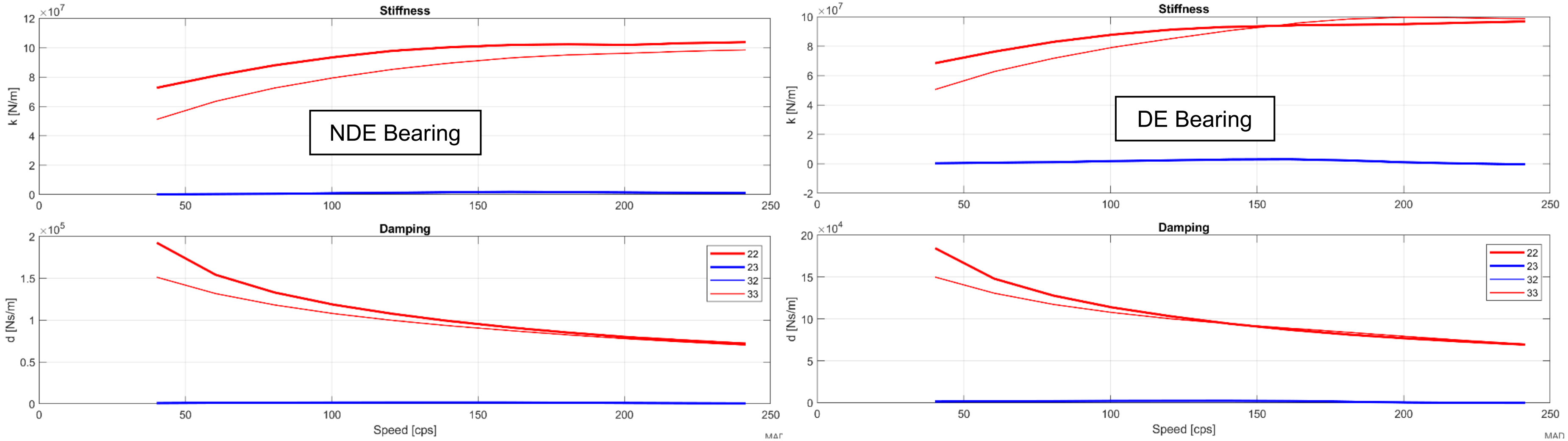


Fig. 3.7: Rotordynamic stiffness and damping coefficients for the static loads without seal influence

The resulting Campbell diagram can be seen in figure 3.8. As already noted for the results of the stability analysis the damping increases and the frequencies of modes shift considerably due to the influence of the honeycomb seal.

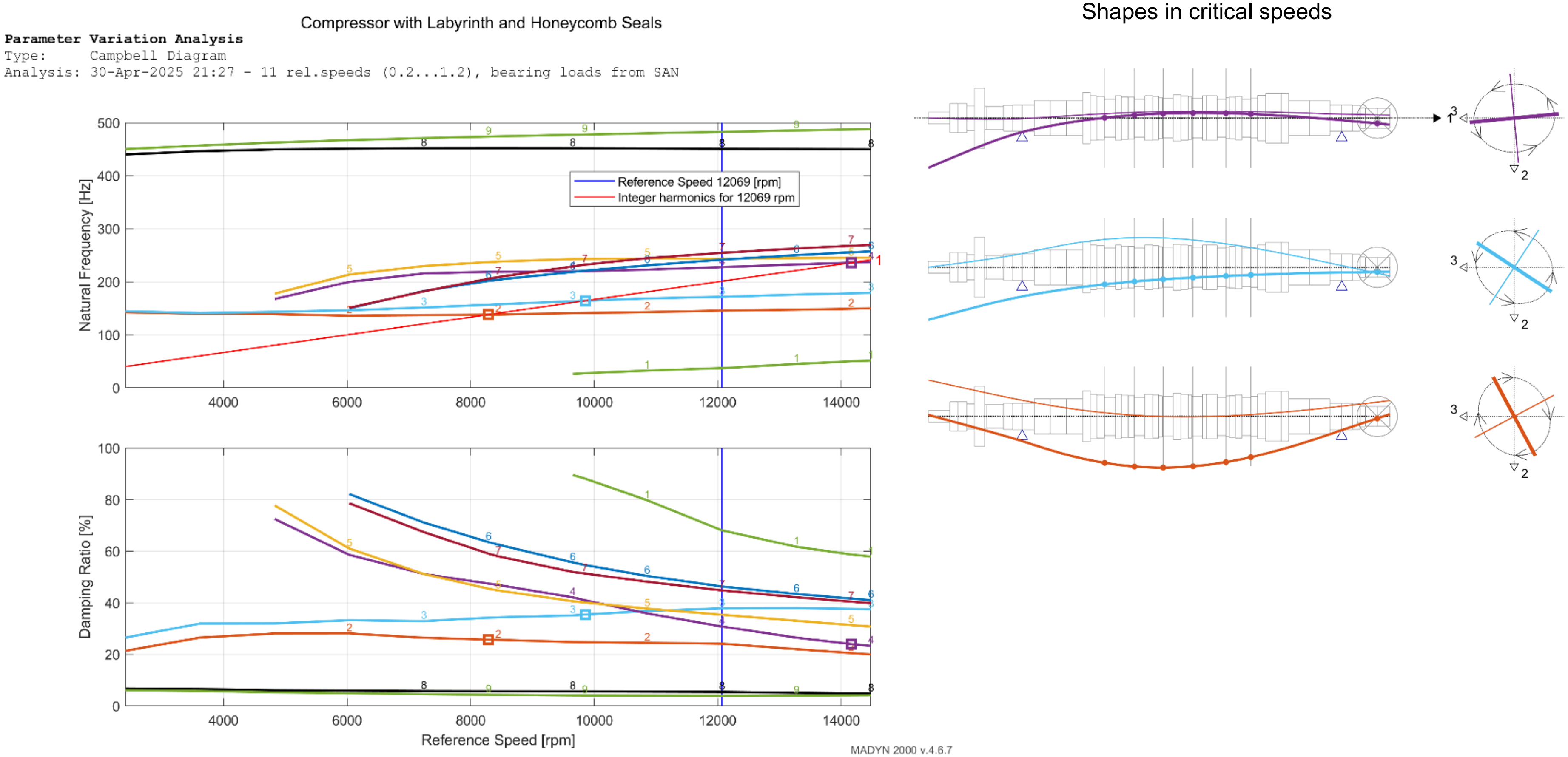


Fig. 3.8: Campbell diagram with seals

4. Summary

The behaviour of a high-density compressor supported on tilting pad fluid film bearings with labyrinth and honeycomb seals is described.

The consideration of honeycomb seals with their strongly frequency dependent behaviour is new in version 4.6 of MADYN 2000.

The seal characteristics are described by the radial and tangential forces on the rotor for a circular orbit whirling with different frequencies. The Swirl Frequency Ratio SFR as the frequency for zero tangential forces is determined for all seals. For the seals no swirl brakes are applied, resulting in high SFR values, especially for the labyrinth seals. Modes with a relative frequency to the speed below SFR are destabilised.

It is shown that the honeycomb seals carry a great portion of the weight load and therefore have a high influence on the static bearing loads, which changes the dynamic characteristic of a fluid film bearing. The rotor can no longer be considered as statically determined. Therefore, a hyperstatic nonlinear static analysis is necessary.

The labyrinth seals also have some influence on the bearing loads, but by far not as much as the honeycomb seals. In most compressors this influence is negligible.

The Campbell diagram and eigenvalues at nominal speed were calculated for the following conditions: no seal influence (unloaded condition), considering the labyrinth seals only and considering all seals. The labyrinth seals destabilise the 1st forward bending mode of the compressor (critical speed at about 60% speed without seal influence) due to the high SFR of the labyrinth seals of about 150%. The damping ratio drops from 10.5% to 2.4%. The Honeycomb seals have a SFR of 40% and 20%, respectively, and a considerable stiffening effect, increasing the frequency of the 1st forward bending mode to almost 70% of the nominal speed. This results in a high damping of this mode (increase from 10.5% damping ratio to 24.2%).

5. References

/1/ MADYN 2000 Documentation Version 4.6.7, March 2025.

/2/ Joachim Schmied et. al.: Influence of Seals with Frequency Dependent Characteristics on the Rotordynamic Behaviour of High Pressure Compressors. Conference Fluid-Struktur-Wechselwirkung, VDI, Wiesloch, Heidelberg, Germany, 2002.

/3/ George F. Kleynhans, Dara W. Childs: The Acoustic Influence of Cell Depth on the Rotordynamic Characteristics of Smooth Rotor / Honeycomb-Stator Annular Gas Seals. ASME paper No. 96-GT-122, presented at the ASME Turbo Expo, Birmingham, UK, 1996.