Vibration resistance of air bearing turbo compressors

Christian Loosli, Fabian Dietmann, Patrik Fröhlich, Christof Zwyssig

Air bearing radial turbo (also called centrifugal) compressors prevail in most mobile fuel cell air supply applications due to the small size and weight, the high efficiency and the oil- and maintenance free operation. An important aspect in mobile fuel cell applications is the vibration resistance of all system components, including the compressor, with vibration requirements up to 20 g in automotive applications.

Vibration resistance in air bearing compressors is different to ball bearing compressors: a touchdown of the rotating component in an air bearing has to be avoided and following this, no short-term bearing overload is possible. On the other side, vibrations within the specifications do not shorten the lifetime of an air bearing as compared to ball bearings. Vibration resistance in air bearing turbo compressors depends on inlet temperature and pressure of the fluid, the rotational speed, the operating point in the compressor map, and the compressor and bearing design itself. Therefore, the vibration requirement, beside all other requirements, has to be included in the design of the air bearings. This can be used as an advantage, as the vibration resistance of an air bearing turbo compressor can be predicted with the design tools without extensive vibration testing. In developing new compressors, the vibration resistance can be included and verified with tools. For the fuel cell system integrator it is an advantage, as the vibration resistance of an existing turbo compressor can be analyzed in different operating points with the respective tools and e.g. mounting options can be designed properly.

This paper gives the background of the air bearing vibration characteristics, depicting the dependencies of vibration resistance on inlet conditions and operating points. The critical operating conditions concerning vibration resistance are identified, and it is outlined how vibration requirements can be included in the design process of an air bearing turbo compressor. A visualization of vibration resistance in the commonly used compressor map is presented, allowing the fuel cell system integrator to take qualified decisions for the mechanical integration of the compressor concerning vibrations.

KEYWORDS: Shock/vibration resistance, centrifugal compressor, air/gas bearing

1 Introduction

To combat climate change, there is a political and social demand for sustainable, storable and competitive energy. Hydrogen is identified as one of the key enablers, with fuel cells as key technology to convert hydrogen into electrical energy. A quantified goal is formulated in the European Union's Green Deal, of which hydrogen is an integral module, by reducing greenhouse gas emissions by at least 55% by 2030 [1].

For increasing the fuel cell's efficiency and reduce the overall fuel cell system volume and weight, which is specifically important in mobile applications, compressed ambient air is provided by a compressor. Due to its power consumption and cost share, the compressor is one of the most important components in the fuel cell system beside the stack, also denoted as balance of plant (BoP) [2].

Centrifugal compressors, also called radial turbo compressors, achieve the same performance (mass flow and pressure ratio) compared to conventional compressor technologies such as piston or scroll compressors with up to 50 times less volume and

weight, and are therefore a preferred choice for mobile fuel cell systems [3]. The high speed of turbo compressors on the one side allow miniaturized aerodynamics and motors resulting in high power density and on the other side require contact free bearings to cope with the lifetime requirements of several thousand to ten thousand hours operation time. In addition, due to the fuel cell's sensitivity towards impurities such as particles and in particular oil, the entire compressor, and specifically the bearings, are required to avoid any contamination of the compressed air. Air bearings are best suited to cope with these requirements by simply utilizing the inlet air as bearing fluid, thus having no wear during operation and avoiding any contamination of the fuel cell with oil such as in oil or ball bearings.

BoP components for mobile applications face requirements for shock and vibration resistance. Vibration resistance in air bearing compressors is different to ball bearing compressors. Vibration levels and durations affect the lifetime of ball bearings, whether they are within the specifications or outside, and some short-term overload is usually possible. In contrary, in air bearings, the air film has a defined load capacity, and if this load capacity is exceeded by the vibration the rotor touches down, usually leading to a disintegration of the rotor. Therefore, such a touchdown has to be avoided and no short-term overload is possible. On the other side, vibrations within the specifications do not shorten the lifetime of air bearings, as it is the case of ball bearings. As there is no contact in air bearings, also in case of vibrations, just the air film is compressed. No wear and no additional stresses occur in air bearings seeing vibrations within the specifications.

For these reasons, it is critical to include the vibration requirements into air bearing compressor designs, and to know the safe operating range of the air bearing turbo compressors concerning vibration resistance. To achieve this, this paper gives the theoretical background and presents tools for visualization of the vibration resistance of air bearing turbo compressors. The dependencies of load capacity of air bearings on different operating conditions and inlet conditions of the compressor are described in chapter 2. Chapter 3 describes the forces applied onto the air bearings by operation of the air bearing within a turbo compressor. The difference of the load capacity as described in chapter 2 and the forces applied during operation as described in chapter 3 results in the vibration resistance of the entire air bearing turbo compressor, which is visualized in the commonly used compressor map shown in chapter 4. This allows the fuel cell system integrator to take qualified decisions for the mechanical integration of the compressor concerning vibrations.

2 Load capacity and acceleration capability in air bearings

An air bearing turbo compressor system consists primarily of a rotor with air bearings, an impeller and a permanent magnet surrounded by a housing with the cooling system and the stator of the electric motor. Figure 1 shows a simplified sectional view of such a system. The impeller is attached to the left-hand side of the rotor, followed by a disc-shaped part, which is the axial bearing, and a long cylindrical rotor part in which the two radial bearings are arranged. The permanent magnet, which is used to drive the rotor, is attached to the right end of the rotor. The bearing components of the rotor are surrounded by the bearing counterparts of the stator. Between the rotor and the stator bearing parts is a bearing gap, which allows the rotation of the rotor.

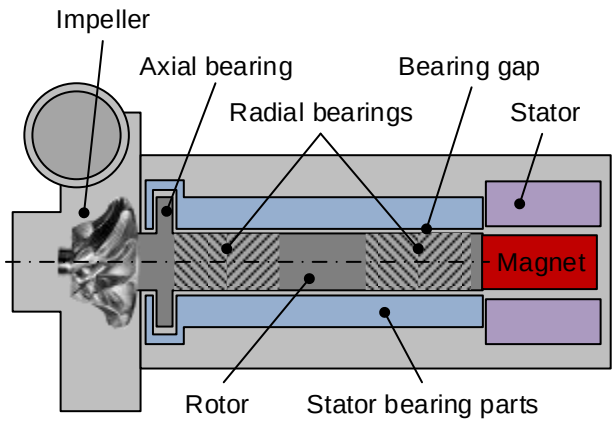

Figure 1: Intersection trough a sample air bearing compressor.

Design parameters of an air bearing system are, besides others, the geometric dimensions such as lengths, diameters of bearing surfaces, bearing gaps, as well as geometry and depth of the bearing structure.

The aerodynamic design defines a torque requirement, which is basis for the electromagnetic motor design and finally defines the magnet volume. Together with the operating requirements, an overall optimum air bearing rotor system is targeted. Important optimization parameters are the stability of the air bearing as well as its load capacity and the losses. Stability refers to the response of the air bearing system to excitation in all possible operating conditions. According to theory, very small displacements around the center position are assumed for the radial bearing [4]. A stability margin is defined in order to compensate margins for effects such as surface structure deviations and shape and position tolerances in the air bearing. The air bearing stability thus is a limiting criterion. This is in contrast to, for example, the load capacity, which shall be maximized. The load capacity describes the “strength” of the air bearing. The load capacity can be optimized and there are trade-offs with other optimization parameters, and thus increased, for example, by accepting higher losses.

With the help of Celeroton's proprietary design tools, which are shown in detail in [5], an overall optimization of the system can be calculated. Short calculation times per calculation loop allow the algorithm to converge efficiently with regard to the optimization parameters defined.

A major challenge for this calculation process is the range of design parameters, e.g. the manufacturing tolerances and the range in operating conditions. This includes the air bearing pressure and temperature range, depending on the ambient pressure and temperature range and the thermal and fluid dynamic compressor design. It further includes the speed range and the manufacturing tolerances of the different air bearing dimensions. Finally, the minimum load capacity required to compensate vibration, depending on operating conditions and rotational speed.

The air bearing design tools calculate the load capacity depending on the bearing geometry, the rotational speed, the air pressure and temperature in the bearing, and the relative displacement of the rotor in the bearing gap, where 0% is referencing a rotor in the center of

the bearing, and 100% referencing a touchdown of the bearing. Significant differences in calculation as well as load capacity characteristic are present between the operating characteristics of the axial and radial bearings, e.g. in the axial bearing the characteristic can be different in the two axial directions as the axial bearing has two sides which can have different designs. The analysis presented in this paper is based on the Celeroton air bearing turbo compressor CT-25-10000.GB, see Figure 2.

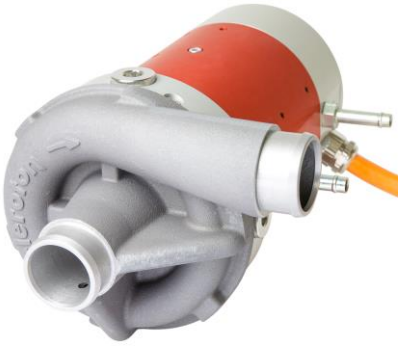

Figure 2: Product photo CT-25-10000.GB.

Figure 3 and Figure 4 show the load capacity, plotted versus the relative displacement of the bearing at various speeds, a bearing temperature of 20°C and a pressure of 1 bar(a). A direct comparison shows the almost linear behavior of the radial bearings, whereas the axial bearing behaves nonlinear.

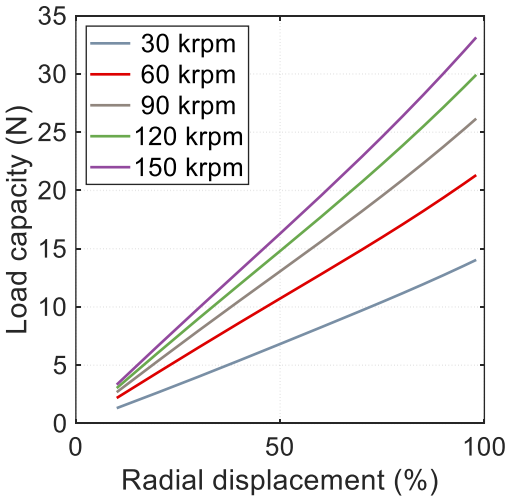

Figure 3: Radial bearing load capacity plotted versus relative radial displacement for CT-25-10000.GB at different rotational speeds at 20°C and 1 bar(a).

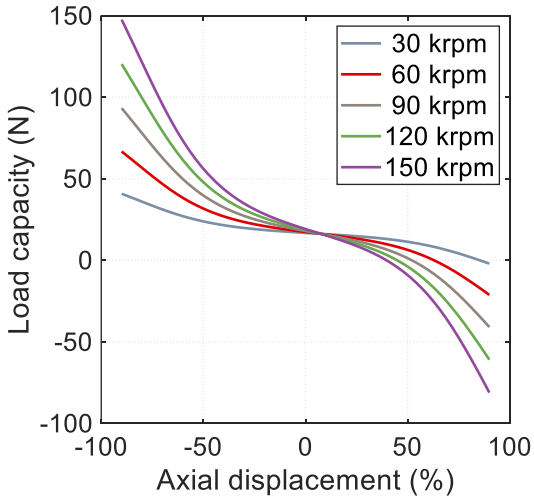

Figure 4: Axial bearing load capacity plotted versus relative axial displacement for CT-25-10000.GB at different rotational speeds at 20°C and 1 bar(a).

As described above, once the bearing dimensions are fixed, both the load capacity and the stability depend to a large extent on the operating conditions in the bearing, such as the prevailing temperature and pressure. An important parameter often referred to during air bearing design is the viscosity in the air bearing gap. It actually is dependent on the pressure and temperature in the air bearing. A critical operating condition is a cold start at high

altitude, with low temperatures e.g. in winter and thus low viscosity in combination with a low inlet pressure level. This leads to a low load capacity in the air bearing together with a relatively low margin on stability due to the low viscosity.

The above plots are valid for "static" loads such as steady state operation or a single shock acting on the air bearing. The complexity increases further when the frequency dependent response of the air bearing system is considered. This is relevant for vibrations at different frequencies. For illustrating the shock and vibration capability, the load capacity of the air bearing is divided by the (constant) rotor mass, resulting in the acceleration capability, which can be expressed in the commonly used unit of gravitational force equivalent (g). For the radial bearing, the acceleration capability is depicted in Figure 5 and Figure 6.

Figure 5 shows the shock capability of the compressor as a function of rotational speed. This figure represents the simplified case for a frequency of 0 Hz. It is assumed that a shock is an excitation of the system in the low frequency range. It can be observed that the shock capability of the radial bearing increases with speed.

Figure 6 shows the frequency dependent acceleration capability. In the low frequency range up to approximately 100 Hz, the acceleration capability is almost constant and largely independent of the excitation vibration frequency. Above 100 Hz, a reduction in the acceleration capability of the radial bearing can be observed until a minimum is reached at the eigenfrequency of the respective air bearing. At higher frequencies, the acceleration capability increases again.

The reduced acceleration capability in the medium frequency range can be mitigated, for example, by external suspension with dampers. The mounting and suspension point in the respective application is customer-specific and is therefore not further considered in this paper.

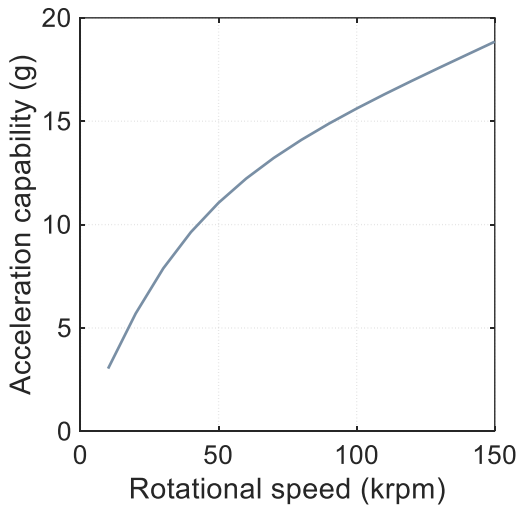

Figure 5: Acceleration capability of the radial bearing plotted versus rotational speed for a static load or single shock for the CT-25-10000.GB at 20°C and 1 bar(a).

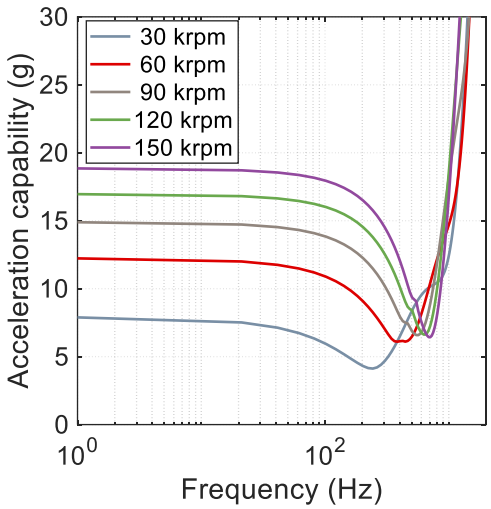

Figure 6: Acceleration capability of the radial bearing as a function of vibration frequency for the CT-25-10000.GB at 20°C and 1 bar(a).

3 Forces in an air bearing turbo compressor

Three main categories of forces can be distinguished which act on the air bearing system. These are:

- Externally imposed forces, such as shocks and vibrations
- Forces independent on the operating point, specifically the gravitational force of the rotor
- Forces that depend on the operating point or the operating conditions of the turbo compressor. Specifically these are the remaining imbalance and the axial thrust force. The force due to an imbalance changes with the speed. The axial thrust depends on the rotational speed, the inlet pressure and the inlet temperature as well as the operating point in the compressor map, i.e. the pressure ratio and mass flow.

The axial thrust force is composed of the force onto the back side of the impeller ($F_{Backdisk}$), the force due to the pressure difference over the suction (SS) and pressure (PS) sides of the blades ($F_{PS, SS}$), and the force on the hub and the area of the blade tip contour (F_{Hub}), see Figure 7. A detailed description of these axial thrust force components, the calculation of the components and the resulting axial thrust force can be found in [6].

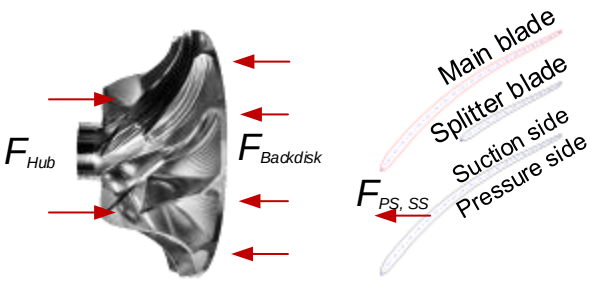

Figure 7: Composition of axial thrust force components on the impeller.

The pressure build-up around the impeller changes at different operating positions in the compressor map. This is because the outlet pressure and flow conditions over the impeller and therefore the pressure levels change around the impeller outlet. This is accompanied by a change in the force components listed above, and therefore the resulting axial thrust force, depending on the operating point within the compressor map.

The range of the axial thrust force is further depending on input conditions such as changed inlet pressure or temperature, as this also results in different compressor maps as depicted in Figure 8 and Figure 9. The operating behavior of the aerodynamics under different input conditions is described in [2]. This results in different axial thrust force ranges for each inlet condition. The resulting axial thrust force for inlet conditions of 20°C and 1 bar(a) are depicted the compressor map in Figure 10.

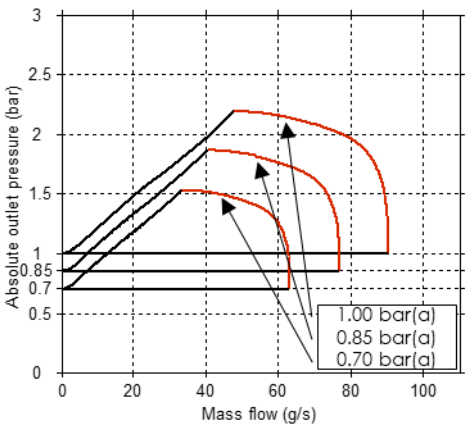

Figure 8: Compressor map of the CT-25-10000.GB with absolute outlet pressure at constant inlet temperature (20°C) and varying inlet pressures [2].

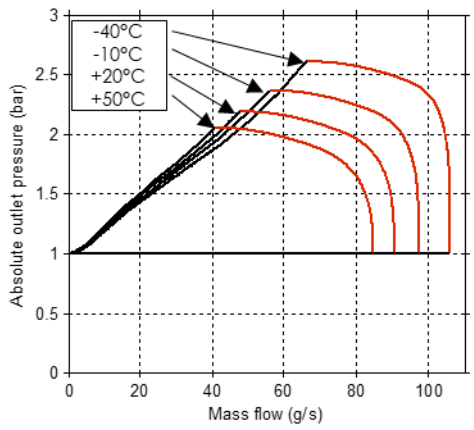

Figure 9: Compressor map of the CT-25-10000.GB with absolute outlet pressure at constant inlet pressure (1 bar(a)) and varying inlet temperatures [2].

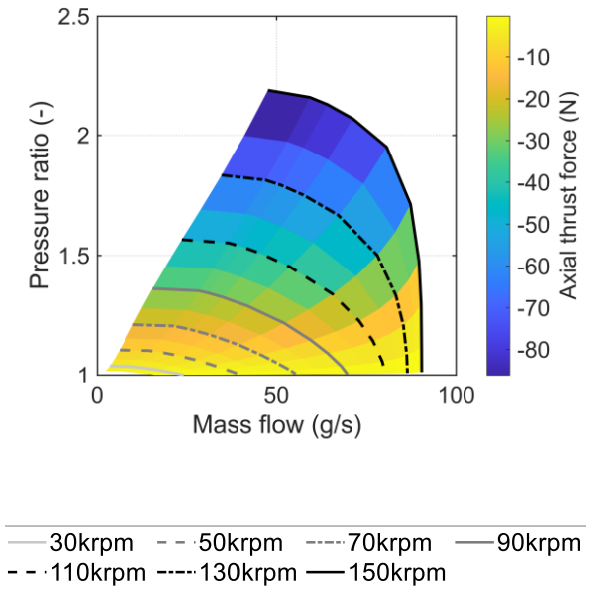

Figure 10: Axial thrust force of the CT-25-10000.GB for inlet conditions of 20°C and 1 bar(a).

4 Visualization of vibration resistance in compressor map

To calculate the performance of the air bearing for absorbing vibrations - the vibration resistance - of an air bearing turbo compressor, the sum of the forces resulting from the aerodynamic design as of chapter 3 and the acceleration capability resulting from the air bearing design tool according to chapter 2 are coupled. Critical operating points are, for example, situations with high axial thrust force and low load carrying capacity of the bearing,

which can occur during cold starts at high ambient pressures. The low temperature results in a low sonic velocity, which leads to a high pressure ratio of the turbomachine at a defined speed. The high pressure ratio also results in a high axial thrust force, with the maximum pressure ratio and therefore maximum thrust force at the operating point close to the surge line. This axial thrust force must be absorbed by the axial bearing, which has a low load capacity due to the low viscosity at cold start. The remaining margins for the absorption of additional forces from shock and vibrations are minimal at this therefore critical operating point.

For users of air bearing turbo compressors such as fuel cell integrators, a simple method of representing and visualizing the complex dependencies of the vibration resistance is required. The following diagrams (Figure 11 to Figure 28) show the compressor map for various operating conditions. For each operating condition, the vibration resistance is shown separately for the radial and axial bearings as well as combined by superposition of the bearings. Each compressor map is limited by the surge line on the left-hand side and by the maximum permissible speed, the permissible torque or by bearing instability on the right-hand side. Furthermore, lines of constant speed are included in the map. Within this commonly known compressor map the vibration resistance is indicated by color coding. The orange, dark and light green and blue areas symbolize the range at which the compressor system has a vibration resistance of 5, 10, 15 and 20 g, respectively. Except for Figure 17 to Figure 19, the temperature of the air in the air bearing corresponds to the temperature at the compressor inlet, which therefore reflects start-up operation of the compressor. After start-up, the air in the air bearing is heated due to losses, which improves the vibration resistance. This can be seen in Figure 17 to Figure 19, which reflect cases where the compressor is at operating temperature (steady state bearing temperature). It is assumed that the temperature in the thrust bearing is 150°C and 100°C in the axial bearings. The figures shown are limited to vibration resistance in the low frequency range where the load capacity of the bearings is still relatively constant, see Figure 6. For higher frequencies, different plots can be generated, or it is assumed that other mitigation measures such as dampers are employed.

These figures further allow to define a minimal speed required to achieve a certain vibration resistance and operating range. Comparing the performance of the axial and radial bearing, it is evident that the axial bearing has a higher vibration resistance. The radial bearing offers the lowest vibration resistance in cold operation at low inlet pressure. This is due to the fact, that the bearing load capacity is lowest because of the combination of low viscosity and low pressure. The axial bearing offers the lowest vibration resistance in cold operation at a high inlet pressure level. This is due to the fact that the axial thrust forces are maximal and the bearing load capacity is low. The most uncritical operating condition in terms of vibration resistance is therefore at high inlet temperature (hot operation). A further finding from the figures is that the vibration resistance increases after cold start when the air in the air bearing is heated. This allows for defining start-up procedures, e.g. preheating. Finally, the data obtained with this methodology can be implemented in a fuel cell system master controller and used in a manner analogous to the operating map of an internal combustion engine to keep the compressor operating in a safe operating condition at all times.

Radial bearing	Axial bearing	Superposition
1 bar(a), 20°C, reference case, start-up		
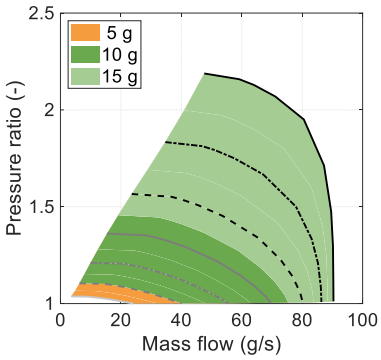	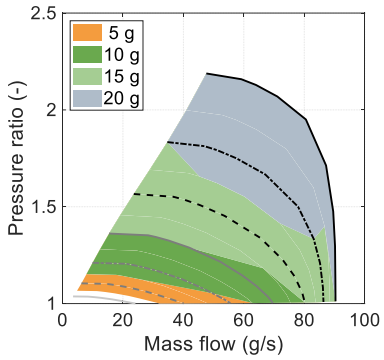	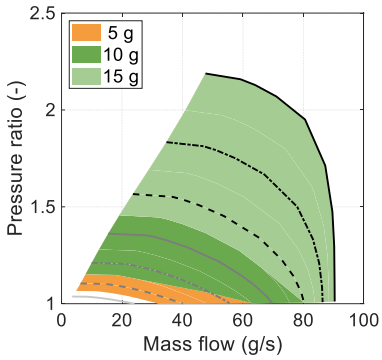
Figure 11	*Figure 12*	*Figure 13*
30krpm 50krpm 70krpm 90krpm 110krpm 130krpm 150krpm		
1 bar(a), -20°C, cold operation (subpolar countries, low altitude), start-up		
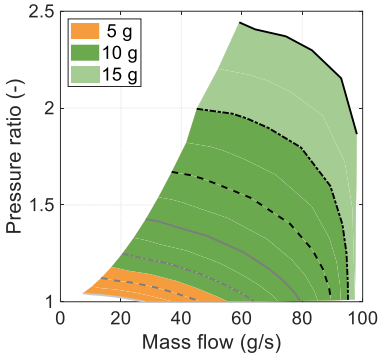	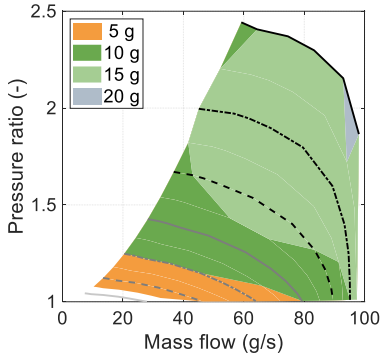	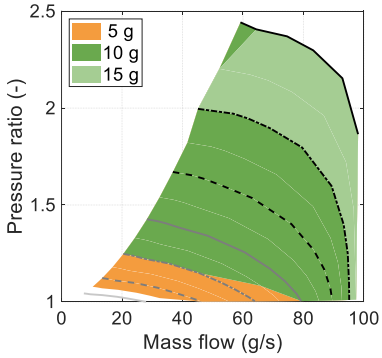
Figure 14	*Figure 15*	*Figure 16*
30krpm 50krpm 70krpm 90krpm 110krpm 130krpm 150krpm		
1 bar(a), -20°C, cold operation (subpolar countries, low altitude), operating temperature (150°C radial bearing air temperature, 100°C axial bearing air temperature)		
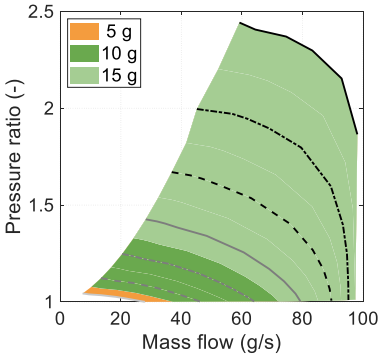	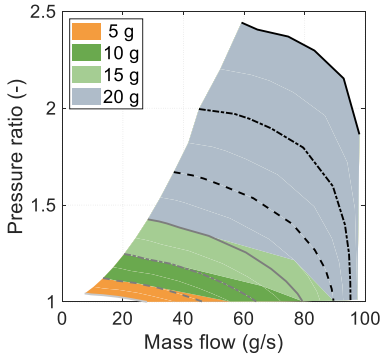	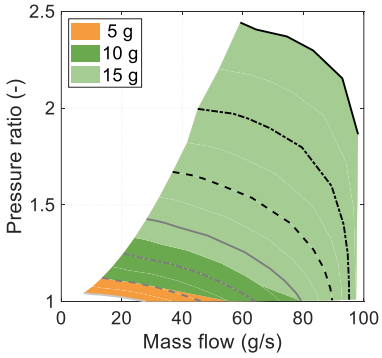
Figure 17	*Figure 18*	*Figure 19*
30krpm 50krpm 70krpm 90krpm 110krpm 130krpm 150krpm		

Radial bearing	Axial bearing	Superposition
0.7 bar(a), -20°C, cold operation (high altitude), start-up		
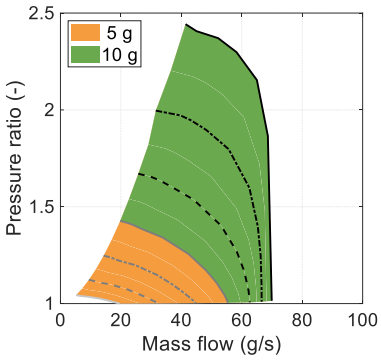	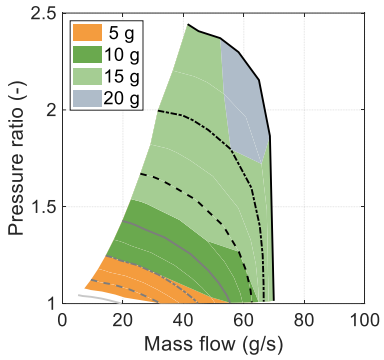	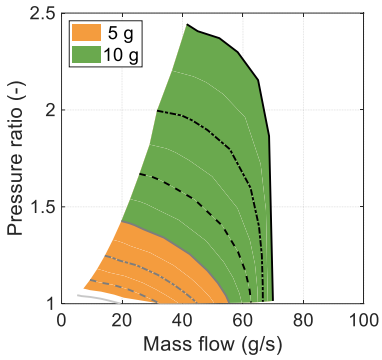
Figure 20	*Figure 21*	*Figure 22*
30krpm 50krpm 70krpm 90krpm 110krpm 130krpm 150krpm		
1 bar(a), 40°C, hot operation (desert), start-up		
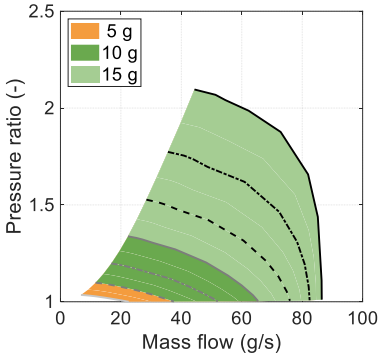	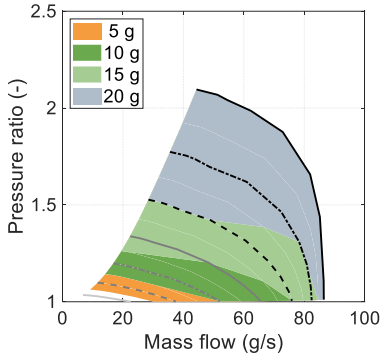	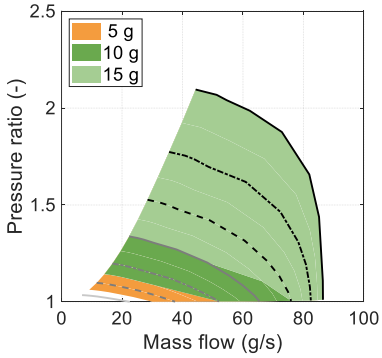
Figure 23	*Figure 24*	*Figure 25*
30krpm 50krpm 70krpm 90krpm 110krpm 130krpm 150krpm		
0.7 bar(a), 40°C, hot operation (high desert), start-up		
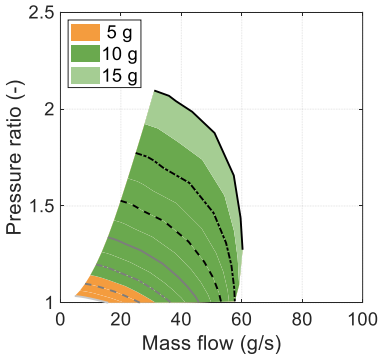	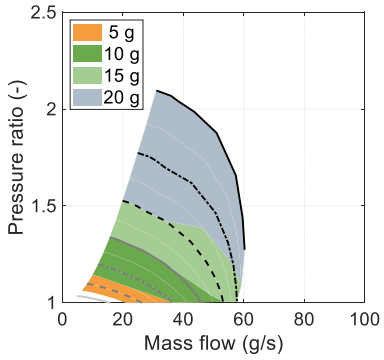	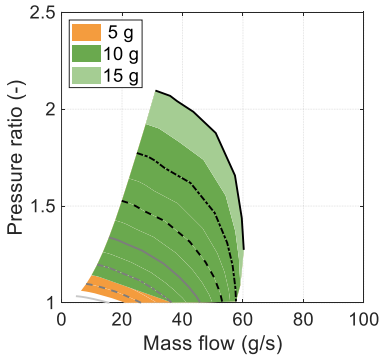
Figure 26	*Figure 27*	*Figure 28*
30krpm 50krpm 70krpm 90krpm 110krpm 130krpm 150krpm		

5 Conclusion

Vibration resistance in air bearing turbo compressors depends on inlet temperature and pressure, rotational speed, the operating point in the compressor map, and the compressor design itself. Therefore, the vibration requirement, beside all other requirements, has to be included in the air bearing design. This can be used as an advantage, as the vibration resistance of an air bearing turbo compressor can be predicted with the design tools without extensive vibration testing. In developing new compressors, the vibration resistance can be included and verified with design tools.

For existing air bearing turbo compressors, the vibration resistance can be analyzed for different operating conditions with the respective tools. For this, a visualization of vibration resistance in the commonly used compressor map is presented, allowing the fuel cell system integrator to take qualified decisions for the mechanical integration of the compressor concerning vibrations.

6 ACKNOWLEDGEMENTS

The work presented in this paper has been conducted within a project that was supported by the European Union's Horizon 2020 research and innovation program under grant agreement No. 858504.

7 References

[1] E. Commission, „Delivering the European Green Deal | European Commission," 13 09 2021. [Online]. Available: https://ec.europa.eu/info/strategy/priorities-2019-2024/european-green-deal/delivering-european-green-deal_en.

[2] P. Fröhlich, „Development of an oil free turbo compressor for mobile fuel cell applications – challenges and results," in *Konferenzband der ersten FC³ Fuel Cell Conference*, Chemnitz, 2019.

[3] B. Blunier and A. Miraoui, "Proton exchange membrane fuel cell air management in automotive applications," in *Journal of Fuel Cell Science and Technology*, 2010.

[4] J. H. Vohr and C. Y. Chow, "Characteristics of Herringbone-Grooved, Gas-Lubricated Journal Bearings.," *Journal of basic engineering,* p. 568–576, 1965.

[5] F. Dietmann, A. Looser and C. Zwyssig, "Vorteile und Herausforderungen von miniaturisierten, gasgelagerten, ölfreien Turbo Kompressoren für Wärme-& Kälteanlagen," in *DKV-Tagung*, Kassel, 2016.

[6] J. Schiffmann, „Integrated Design, Optimization and Experimental Investigation of a Direct Driven Turbocompressor for Domestic Heat Pumps," *École Polytechnique Fédérale de Lausanne,* 2008.