Structure Design and Simulation Research of Active Magnetic Bearing for Helium Centrifugal Cold Compressor

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TIPC, CAS.

Background

The many large devices of scientific experiment – CEBAF, LHC, CERN, EAST and so on, are equipped with helium refrigeration system. In the refrigeration system, cold compressors are used to pump gaseous helium from saturated liquid helium tank obtain super-fluid helium. As the rotation speed of the spindle for cold compressor, now being developed at TIPC, CAS, is as high as 60000 rpm, and active magnetic bearings are regarded as the best support assembly than other bearings. However, the structure and control strategy of AMBs applied in cold compressor need customize according to the work condition.

Objectives

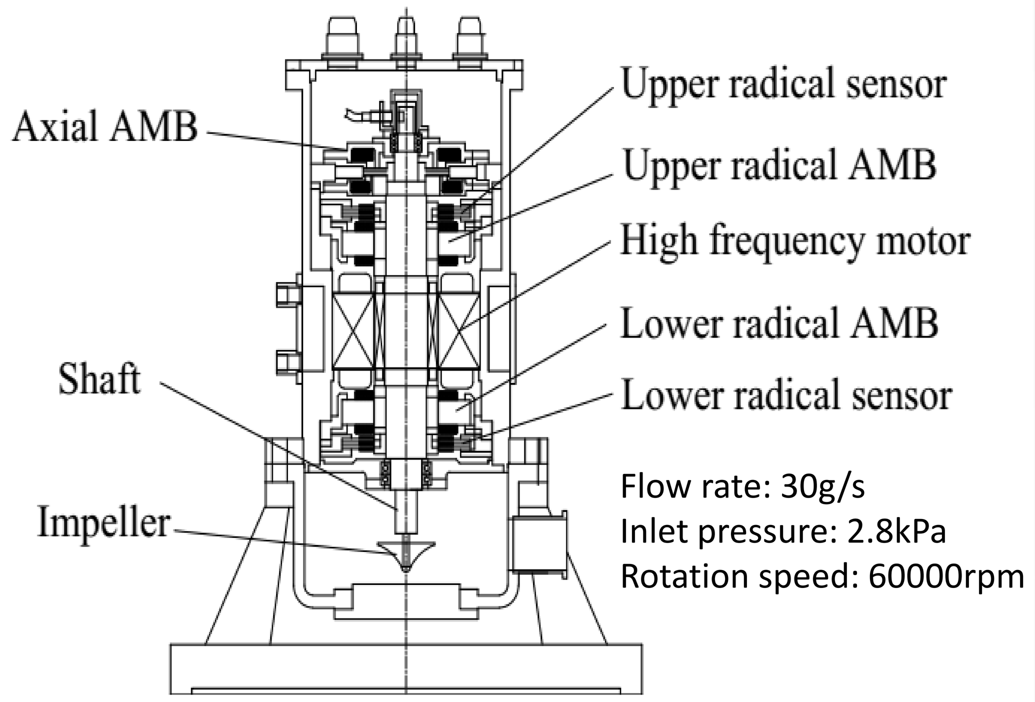
- Structure and hardware design of the whole five-degree-freedom AMB system for a cold centrifugal compressor with rotation speed of 60000rpm.
- Rotor dynamic analysis and controller design to ensure more than 20% separate margin and provide proper stiffness and damper rates at the critical speeds within the bandwidth.

Conclusion

- Structure parameters design methods of the two radial magnetic bearings and one axial magnetic bearing are discussed in detail.
- The choice of electronic control system hardware of AMBs -- controller, sensor and power amplifier and so on, is well explained.
- Rotor modal analysis results show there is 54% safe margin between rated speed (1000Hz) and vortex frequency (1540Hz) of the first order bending mode, verifying the rationality of structure design of the rigid rotor.
- The control strategy, combining PID arithmetic with other phase compensators, proposed by the paper, provides satisfying stiffness and damper for AMB system. Control simulation results demonstrate that the control method not only stables AMB system but also guarantees good performance of closed-loop behavior.
- The research work offers important base and experience for test and application of AMB experiment platform for system centrifugal cold compressor.

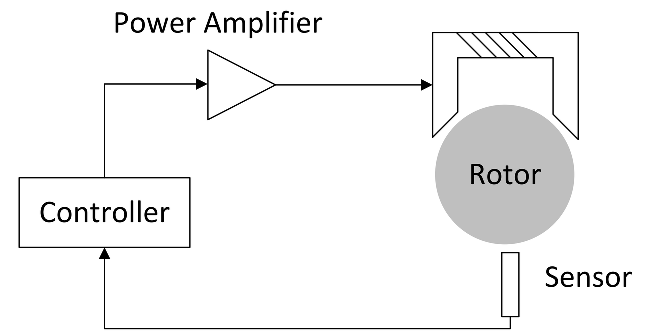
Structure

characteristics of cold compressors



Flow rate: 30g/s
Inlet pressure: 2.8kPa
Rotation speed: 60000rpm

Hardware design



- **Controllers**: DSP (TMS320C6713)
- **Monitoring system**: host computer and LABVIEW
- **inductive transducer**, **Switch amplifiers**
- **Auxiliary bearings**: ceramic bearings of SKF

Structure parameter

Parameter	Value
Radial AMB	
Number of magnetic poles	8
Load capacity	200
Diameter of stator	60/120
Axial length	20
Maximum current	6.5
Axial AMB	
Load capacity	500
Outer diameter of stator	80
Interior diameter of the stator	50/62.5
Maximum current	6.5

- **Pole sequence**: N-S-S-N-N-S-S-N
- **Radial load capacity** is suggested at 200N, to counteract vibration force caused by imbalanced mass
- **Axial load capacity** is 500N to balance three kinds of forces
- Upper axial bearing, is bigger than the lower to decrease heat-up of spindle

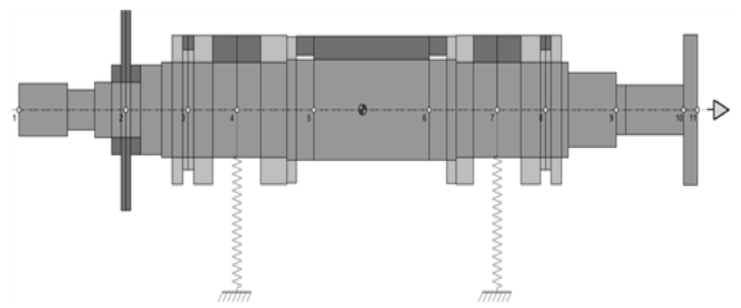
Controller

Model analysis

Assume:

- No axial stiffness
- Radial damping was set at 0, stiffness at $5*10^5$
- No decoupling between radial degrees of freedom

Software:
DyRoBes



Controller design

Take one of the radial magnetic bearings for an example.

Transfer function of magnetic bearing: $G = \frac{k_i}{ms^2+k_x} = \frac{28.4}{5.223s^2-264000}$

Time delay: $[\text{num}, \text{den}] = \text{padé}(1e^{-4}, 3)$

Transfer function of PID and other filters:

$$F_{PID} = 1.5e^6 \cdot \frac{\left(1+\frac{s}{2\pi\cdot 300}\right)\left(1+\frac{s}{2\pi\cdot 400}\right)}{\left(1+\frac{s}{2\pi\cdot 700}\right)\left(1+\frac{s}{2\pi\cdot 800}\right)} + \frac{1}{1+\frac{s}{2\pi}}$$

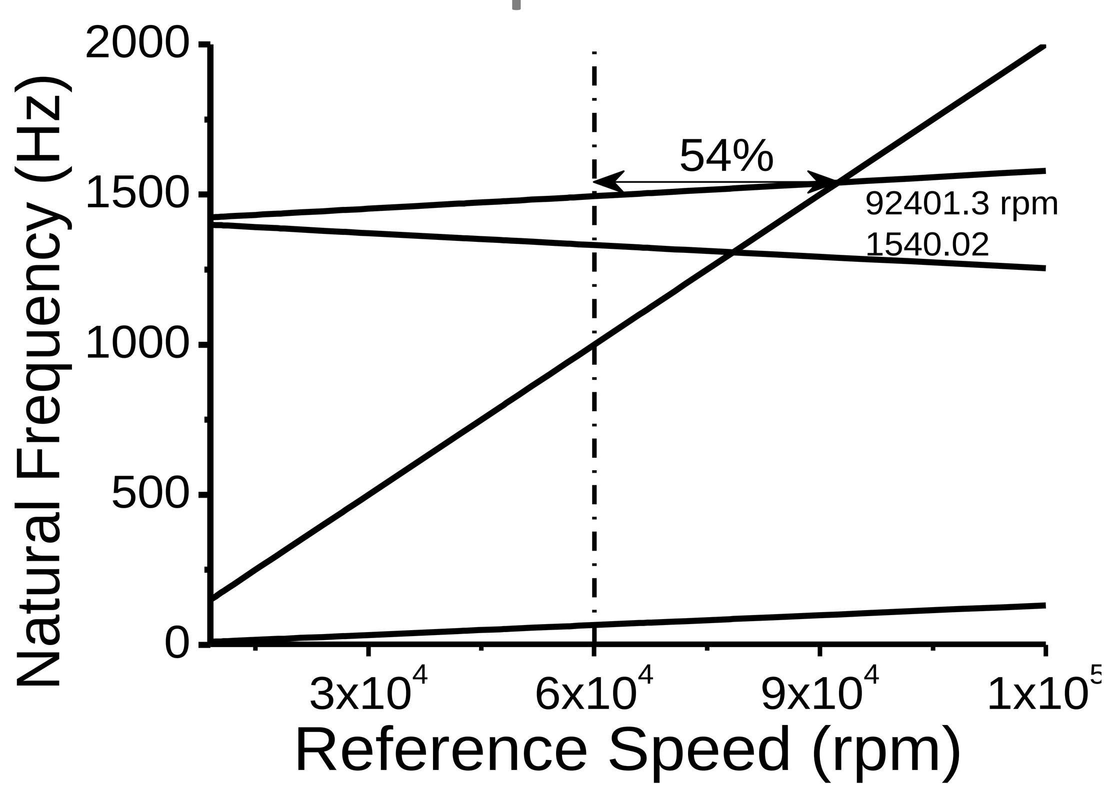
$$F_1 = \frac{1}{\left(1+\frac{0.2s}{2\pi\cdot 1900}\right)+\left(1+\frac{s^2}{2\pi\cdot 1900}\right)}$$

$$F_2 = \frac{1+\left(\frac{s}{2\pi\cdot 5100}\right)^2+\left(\frac{0.2s}{2\pi\cdot 500}\right)}{1+\left(\frac{s}{2\pi\cdot 4950}\right)^2 \ldots (2\pi\cdot 4950)}$$

[illegible] 截图(Alt + A)

Results

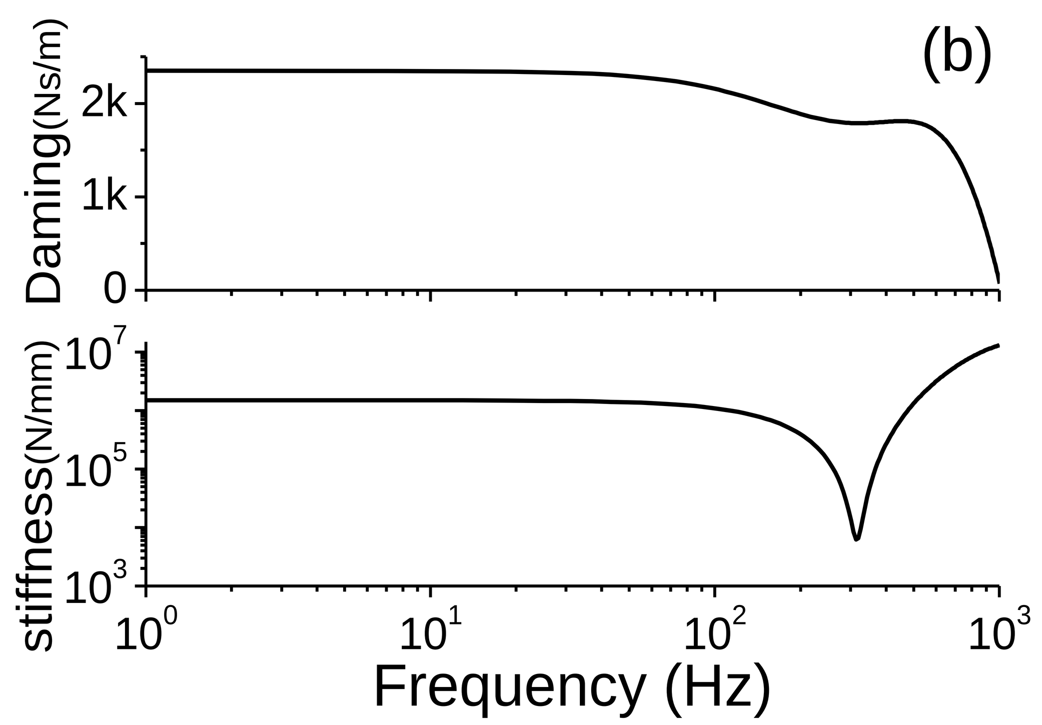
Campbell chart



- There is 54% safe margin between rated speed (1000Hz) and vortex frequency (1540Hz) of the first order bending mode
- Rotor can be seen as a rigid rotor, so conventional PID arithmetic, combining with phase compensation method, can control rotor well

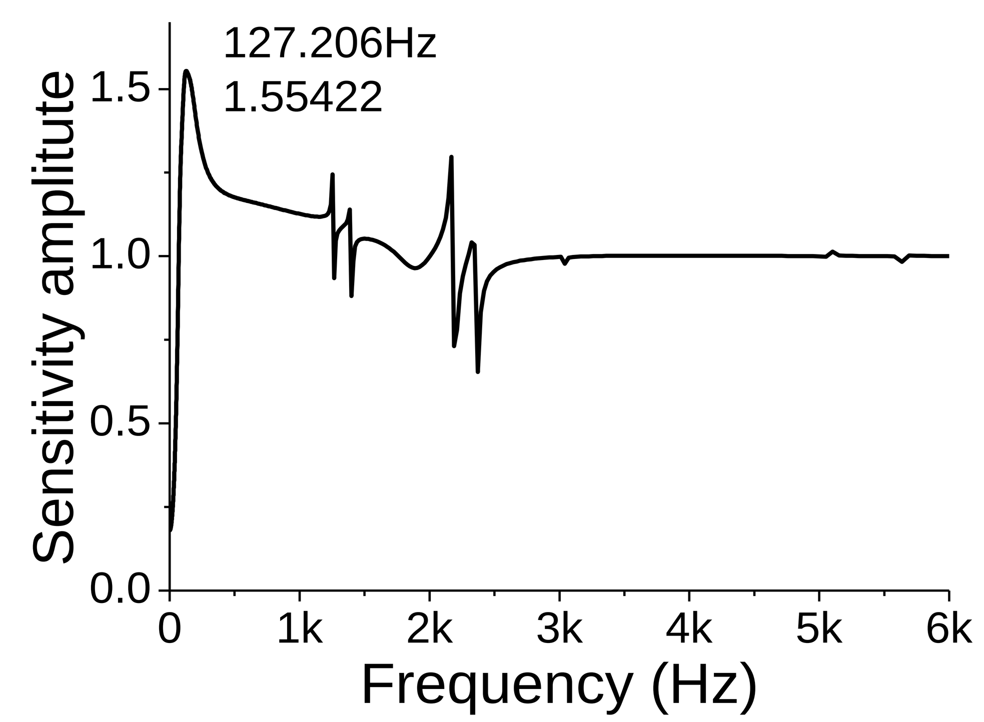
Stiffness and damping

Proposed controller can offer decent positive stiffness with the same order of magnitude as the displacement stiffness as well as proper damping.



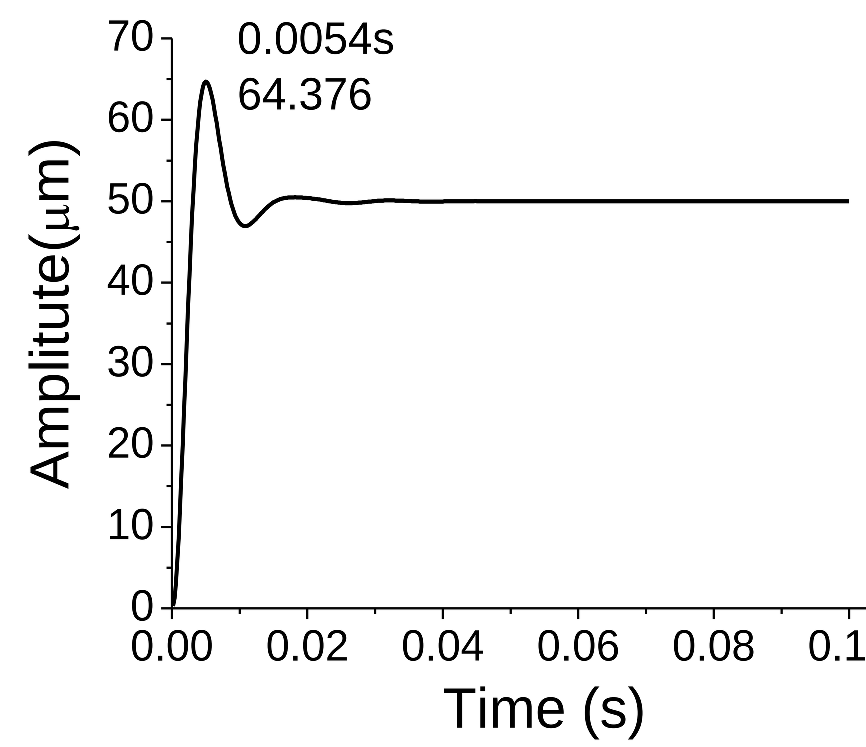
- Guarantees damping ratios of the parallel and conical mode with frequency of 104.41Hz and 144.90Hz at 0.34 and 0.32, within working rotation speed
- The first bending mode frequency is at 1399.42Hz, with 39.94% safe margin and its damping ratio value is at 0.21

Sensitivity functions



According to the ISO standard for the assessment of AMB system robustness, the sensitivity function peak values are no more than 3, the system with the control strategy described above can be considered a "Zone A" system, which is optimally robust to plant uncertainty

Step response



- The displacement in one of radial directions reached its peaks 0.064mm at time 0.0054s
- At last it kept in the central location at 0.05mm

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