Vibration System Analysis of Magnetic Journal Bearing for MMR Condition

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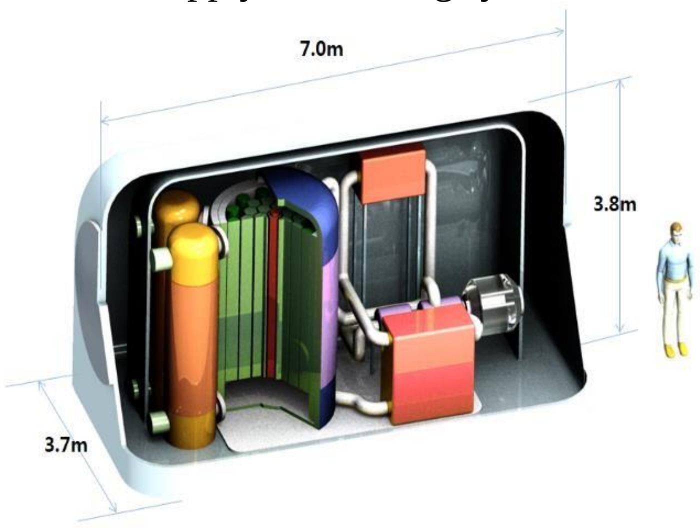
Introduction & Background

KAIST-MMR (MMR, Micro modular Reactor) 's Advantages

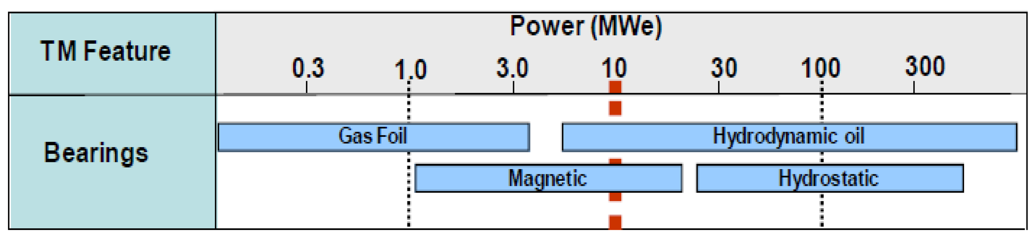
- MMR (fully modularized fast reactor with super critical CO_2) has high power density with moderate heat source temperature.
- MMR can replace the diesel engine to avoid violating the newly released IMO regulation.

Appropriate bearing selection

From the power scale of the MMR, magnetic bearing is well applicable. Oil lubricated bearing is excluded because oil supply and sealing system harms its compactness and independence.



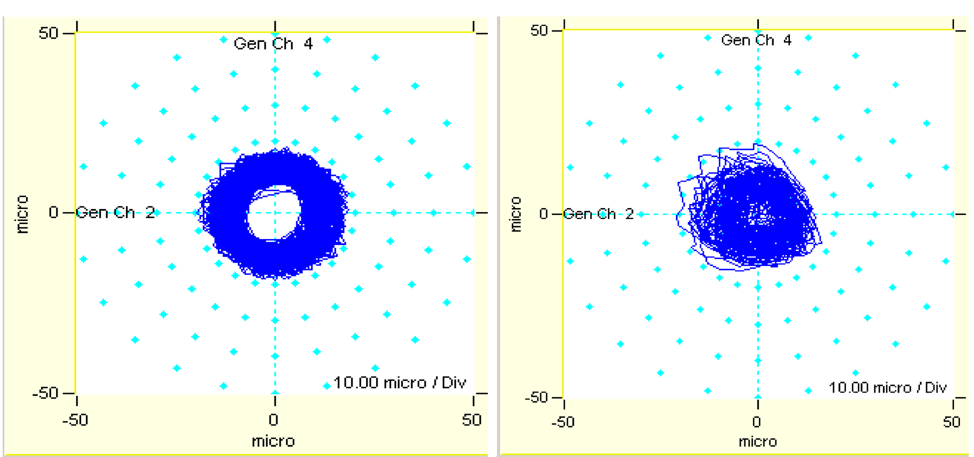
Configuration of MMR



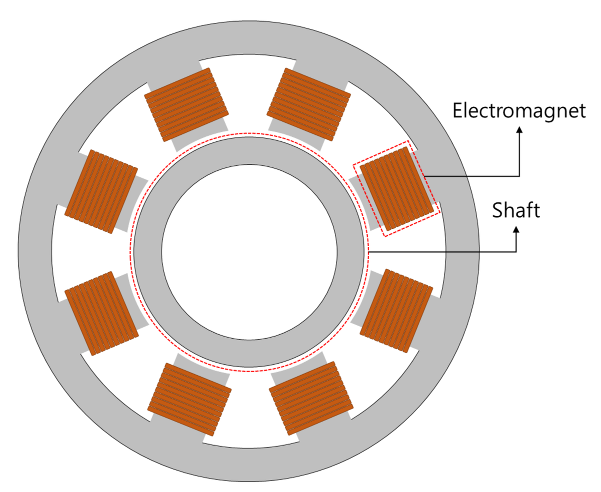
Bearing options for $S\text{-}CO_2$ Brayton cycles with various power scales

Magnetic bearing's radial instability issue

- Under high pressure & high speed operation
- Shaft breakaways from the revolution orbit
- Leaked working fluid cools the rotor
- No such phenomenon with low density fluid



Compressor shaft trajectory under air condition (left, 30000 RPM) and $S\text{-}CO_2$ condition (right, 14000 RPM))



Cross-section of radial magnetic bearing

In this poster, the modeled $S\text{-}CO_2$ lubrication pressure distribution in the magnetic journal bearing geometry with uniform circular motion is analyzed with its physical properties. To explain and verify the results, the experimental results with shaft position is substituted into the model for comparison. Also, the results are analyzed with Fast Fourier Transform (FFT) method to discuss lubrication instability.

Modified fluid force analysis model

Lubrication in magnetic bearing with inner coated geometry

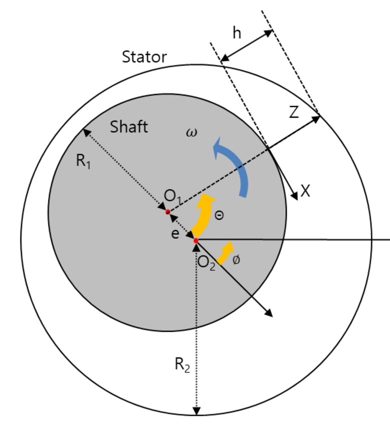
Magnetic bearing's electromagnet is exposed to the working fluid leaked through the labyrinth seal. Because the complex geometry is difficult to model, smooth geometry is analyzed with model at first.

Fluid force model with Reynolds equation

- Thin film fluid dynamics equation
- Velocity profile from Navier & Stokes equation
  → Substitute to the continuity equation
- Negligible axial direction & Quasi steady (perfect revolution)

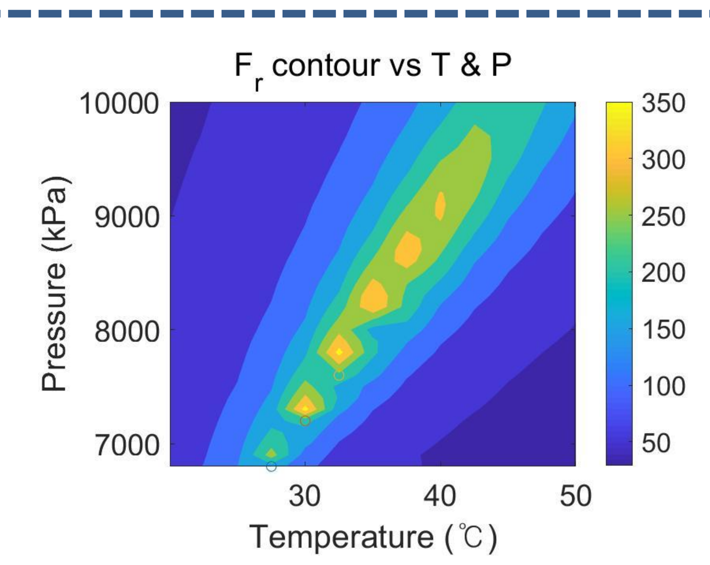
$$\frac{\partial}{\partial X}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial X}\right) = \frac{\partial(\rho h)}{\partial t} + \frac{hu}{2}\frac{\partial \rho}{\partial X} + \frac{u}{2}\frac{\partial h}{\partial X}\rho$$

- Purpose : Pressure distribution & force exerted to the shaft

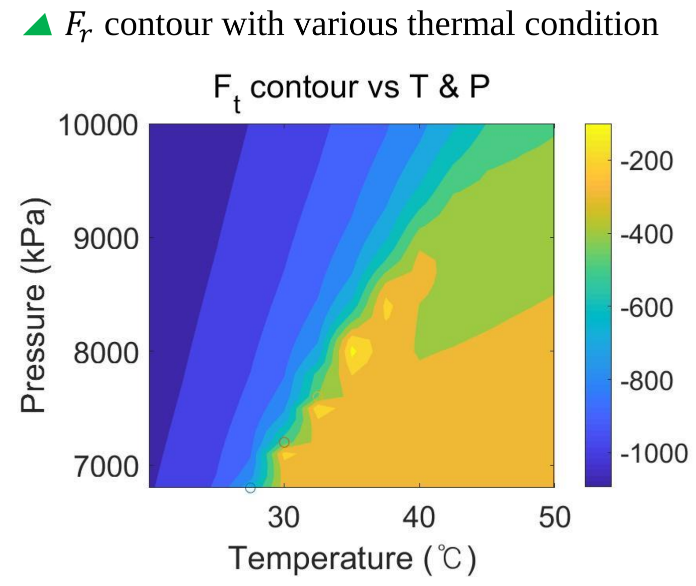


Bearing modeling coordinate description

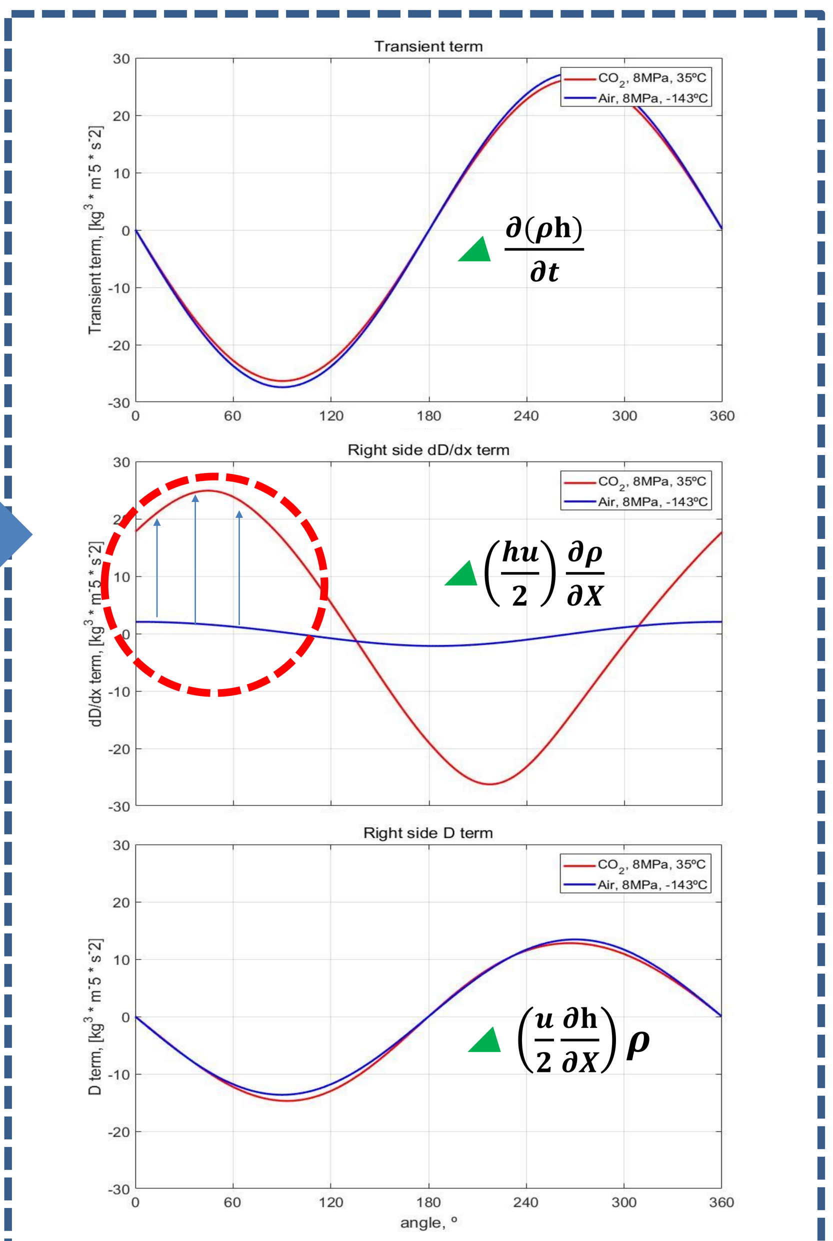
Fluid force model results for 30,000 RPM and, ε (Eccentricity ratio) =0.08



F_r contour with various thermal condition



F_t contour with various thermal condition

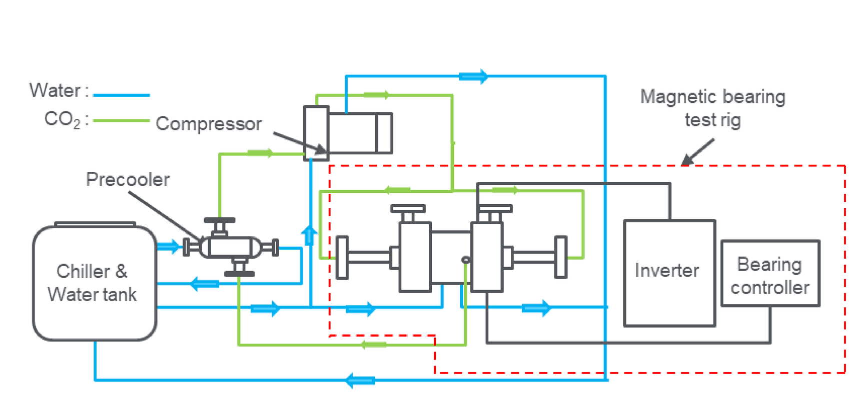


- High Density and its change is main reason of the fluid force gradient
- Control of thermal condition is required for experiment

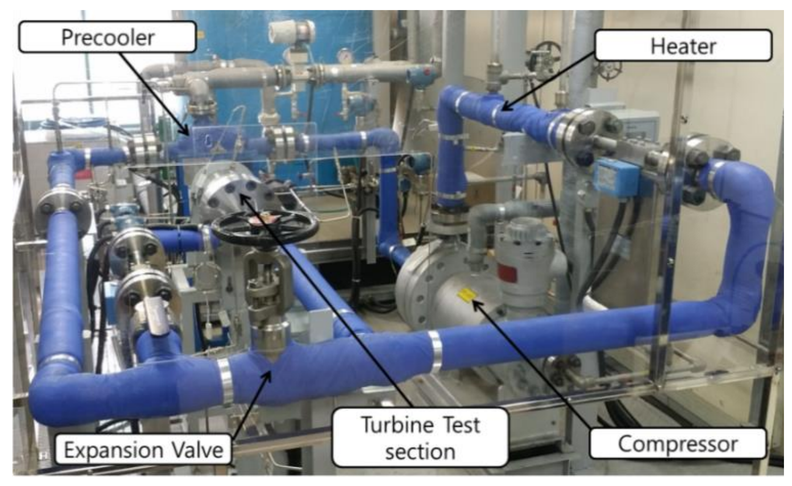
Experimental study of magnetic bearing instability

Layout of the experiment loop

The pump, chiller and heat exchanger are derived from the SCO_2PE which is $S\text{-}CO_2$ pressurizing loop constructed in KAIST to control the thermal condition



Layout of the Bearing Instability Experiment

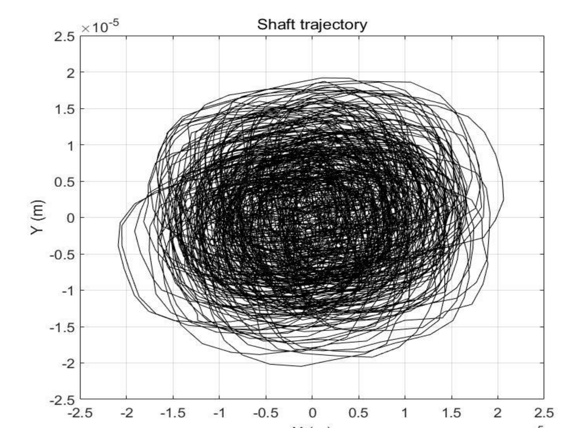


$S\text{-}CO_2$ power cycle demonstration facility ($S\text{-}CO_2PE$)

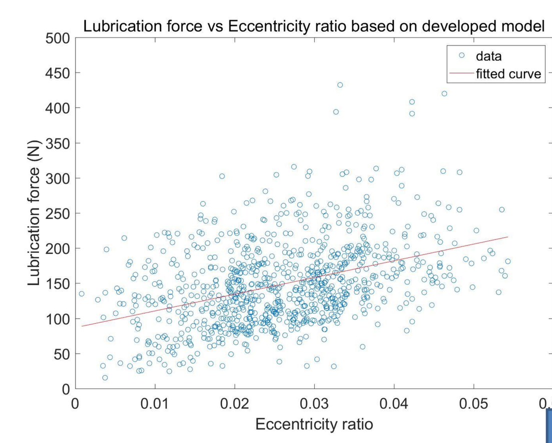
Force analysis

The shaft trajectory data is inserted to the developed fluid force analysis model. From this, the fluid force exerted on the shaft during the experiments are estimated. The calculated results are used to verify the model.

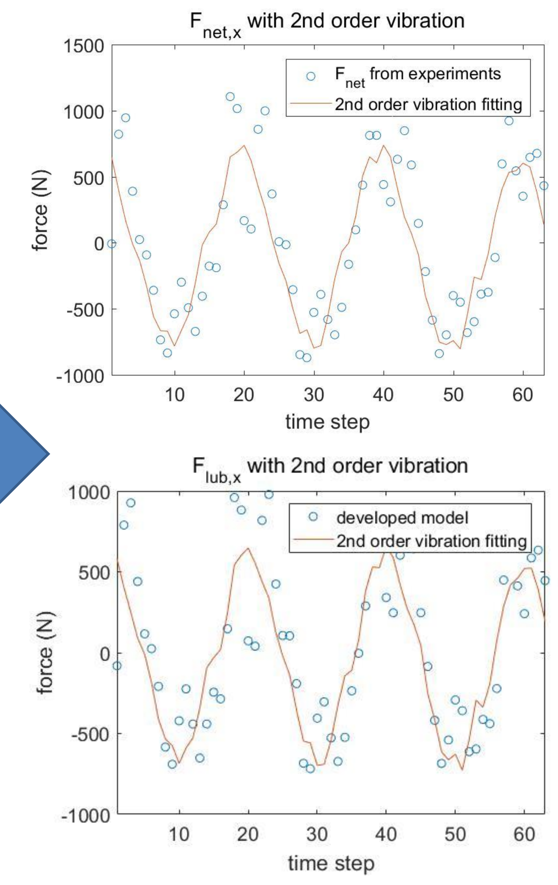
$$F_x = K_{xx}X + C_{xx}\dot{X} + K_{xy}Y + C_{xy}\dot{Y}$$
$$F_y = K_{yx}X + C_{yx}\dot{X} + K_{yy}Y + C_{yy}\dot{Y}$$



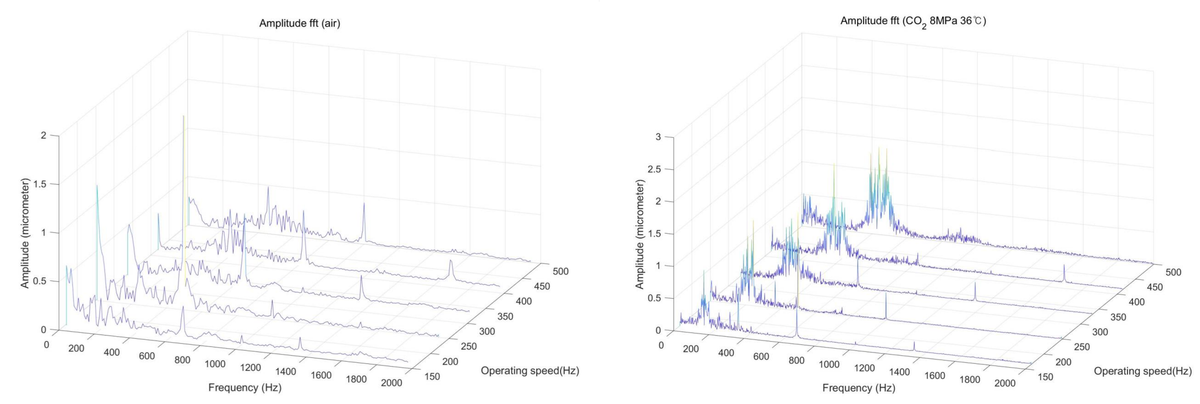
Shaft trajectory with 8MPa, 36℃ CO_2 and 30000 RPM condition



F_{lub} with 30,000 RPM



Forces in 2nd order vibration system



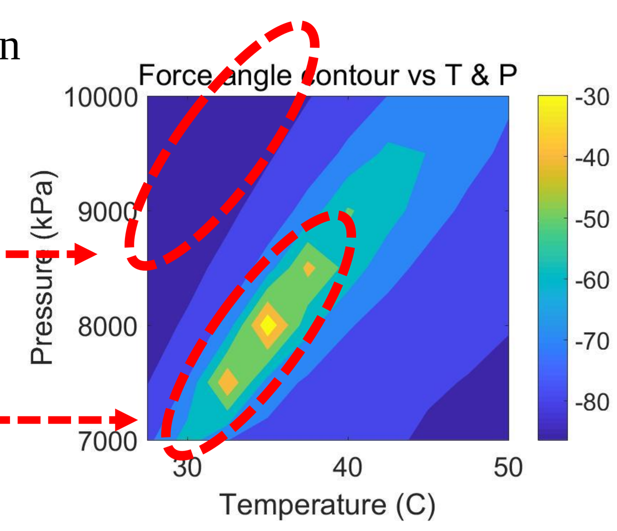
FFT of the shaft trajectory data from air (left) and $S\text{-}CO_2$ test (right)

Vibration parameters

$K = \begin{pmatrix} K_{xx} & K_{xy} \\ K_{yx} = -K_{xy} & K_{yy} \end{pmatrix}$ form is given. The F_t from the different-sign cross coupled stiffness (CCS, K_{xy} & K_{yx}) is collinear with the whirl velocity so destabilize the shaft control by growing energy of motion.

Conclusions & Future work

- The fluid force from CO_2 is sensitive to thermal condition because of the density change
- The fluid force is following the 2nd order vibration system
- The F_t can be the instability source because of the different-sign CCS
- High density of $S\text{-}CO_2$ can be the instability source of the magnetic bearing levitation
- This analysis cannot define the effect of the rapid angle change near pseudo-critical line



Frequency Comparison

- Low frequency noise can be mainly due to the lubrication instability from comparing $S\text{-}CO_2$ and vacuum condition
- The relation between the force angle change and the noise will be researched

State Space Analysis

- $\dot{X} = AX + Bu$, $X = \begin{pmatrix} \dot{x} \\ \dot{y} \\ x \\ y \end{pmatrix}$, $A = \begin{pmatrix} -\frac{C_{xx}}{m} & -\frac{C_{xy}}{m} & -\frac{K_{xx}}{m} & -\frac{K_{xy}}{m} \\ -\frac{C_{yx}}{m} & -\frac{C_{yy}}{m} & -\frac{K_{yx}}{m} & -\frac{K_{yy}}{m} \\ 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{pmatrix}$
- With A's eigenvalue, the vibration system's convergence can be predicted.
- AMB's control strategy can be designed with desired eigenvalue.
- The effectiveness of it is planned to be tested with several control strategy