Investigation of the active magnetic bearing capacity range for sCO_2 power cycle

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1. Introduction

The sCO_2 (supercritical carbon dioxide) power cycle has the advantage of higher thermal efficiency compared to other gas cycles, as it reduces the compression work of the compressor. Additionally, due to the high density of CO_2, the size of the system components is smaller, and fewer auxiliary systems are required, leading to a simplified system layout [1]. However, one of the challenges of using CO_2 as a working fluid is its high solubility with lubricating oil, which makes it difficult to use oil-lubricated bearings for supporting turbomachinery and driving shafts. By adopting active magnetic bearings, the dissolution of oil into the working fluid can be prevented, maintaining the purity of the working fluid. Furthermore, magnetic bearings have a significantly longer lifespan compared to conventional contact bearings, making them highly effective for unmanned systems [2].

Magnetic bearings generate a magnetic field using electromagnets, and this magnetic field acts between the rotor and the stator to levitate and support the rotating shaft. As the magnetic field lifts the rotor, the shaft of the bearing is kept from contacting the stator as shown in Figure 1 [3]. This results in no friction, and there is no need for wear or lubrication. Magnetic bearings typically use an electromagnetic control system to monitor the position of the rotor in real-time and adjust it to keep the rotor rotating stably. This system controls the magnetic field to manage the vibration or movement of the shaft.

One limitation of magnetic bearings is that when the weight of turbomachinery and driving shafts is large, the bearing's load capacity may become insufficient. This study aims to estimate the weight of turbine and driving shafts based on the output of the sCO_2 power cycle and, based on these estimates, propose a method for determining the maximum system output that magnetic bearings can support.

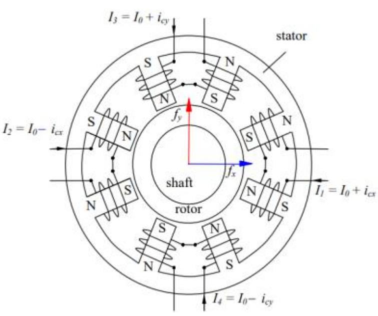

Fig. 1. Coil geometry of four pole pair radial magnetic bearing [3]

2. Rotor Assembly Weight Assumption

To estimate the weight of the turbine and driving shaft based on the output of the sCO_2 power cycle, it is necessary to determine the diameter and rotational speed of the rotating machinery for a given power output.

First, the KAIST-CCD, which is a Closed Cycle Design code for several working fluids and layouts is employed to calculate the mass flow rate of the sCO_2 cycle, as well as the turbine inlet and outlet pressure and temperature conditions [4]. The KAIST-TMS (Turbo Machinery Specific) code utilizes Balje's Ns-Ds diagram, where the specific speed (Ns) and specific diameter (Ds) values corresponding to a turbine efficiency of 90% are used to determine the rotational speed and diameter of the turbine [5]. Turbine type is divided into radial and axial type depending on the cycle net power, 10MWe. The dimensionless parameters specific speed (Ns) and specific diameter (Ds) are given in Equations 1 and 2, respectively. Once the turbine diameter is determined, its volume is assumed to be proportional to the cube of the diameter. The volume is then multiplied by the density of steel (7860 kg/m³) to estimate the turbine mass.

To estimate the driving shaft diameter, Equations 3 and 4 are used. First, Equation 3 calculates the shaft torque by dividing the turbine output by the angular velocity. Equation 4 then determines the shaft diameter based on the allowable torsional stress of steel and the calculated torque. By multiplying this diameter with an appropriate shaft length, the shaft volume and weight are determined. Finally, the total rotor weight is estimated by summing the turbine weight and the driving shaft weight. Flow algorithms of the above descriptions are arranged in Figure 2.

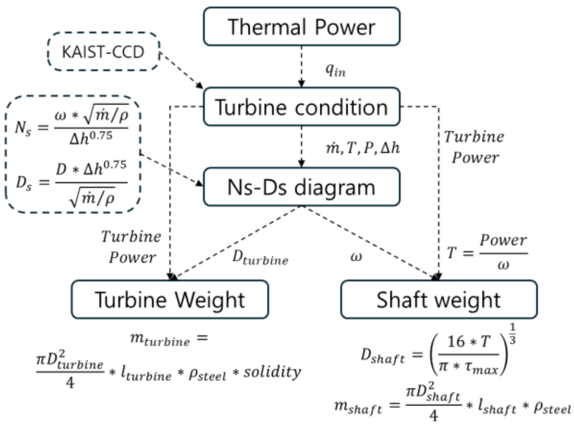

Fig. 2. Flow of the rotor assembly mass assumption

(Equation 1) $N_s = \frac{\omega * \sqrt{\dot{m}/\rho}}{\Delta h^{0.75}}$

(Equation 2) $D_s = \frac{D(\Delta h)^{0.75}}{\sqrt{\dot{m}/\rho}}$

(Equation 3) $T = \frac{Power}{\omega}$

(Equation 4) $D_{shaft} = \left(\frac{16 * T}{\pi * \tau_{max}}\right)^{\frac{1}{3}}$

3. Maximum Magnetic Force Assumption

The force exerted by a magnetic bearing on the shaft is given by Equation 5. This equation states that increasing the current intensity (i), the number of coil turns (N), or the area where the magnetic force is applied (A_g) is necessary to increase the bearing support force. Although it may seem that increasing the current intensity or coil turns can indefinitely enhance the support force, this is not true due to the phenomenon of magnetic saturation. Figure 3 shows the magnetic flux and its saturation depending on the magnetic field strength for several materials [6]. The saturation magnetization of steel is around 1.7T. In conclusion, as Equation 6 states, expanding the magnetic force area is a practical strategy for increasing the load capacity of magnetic bearings. However, this area cannot be increased indefinitely, as an increase in the magnetic force application area leads to an increase in the shaft length and load, which in turn increases the load that the magnetic bearing must support.

(Equation 5) $F = A_g \mu_0 N^2 i^2 / s^2$

(Equation 6) $F_{max} = A_g B_{sat}^2 / \mu_0$ where $B = \mu_0 N i / s$

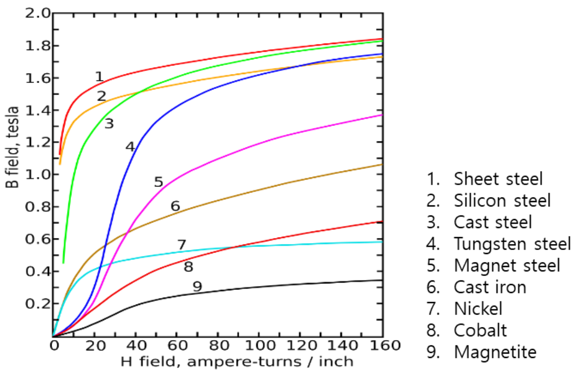

Fig. 3. Magnetization curves of 9 ferromagnetic steels [6]

As explained in Equation 6, the maximum force that can support rotor assembly by magnetic bearing is determined by the saturation of the magnetic flux density (B_{sat}) and the magnetic flux area (A_g). In this study, the design flux density of Silicon Steel that yields at 1.2 T, as presented in Figure 3, was set. For the sCO_2 turbomachinery, due to the high rotational speed, external disturbances can affect the rotor. Therefore, the margin should be kept from the flux density saturation to ensure that the system can respond to sudden external disturbances.

Estimating the area of the flux region is crucial for estimating the support load. In this study, the flux width is set to be 20% of the shaft diameter, and the length is set to be twice this width. Since a magnetic bearing set contains two radial bearings, the total magnetic flux area is four times the area of the width and length described above. This can be expressed mathematically as shown in Equation 7. These coefficients could be changed as the real system can contain.

(Equation 7) $A_g = (D_{saft} * 0.2)^2 * 2 * 4$

4. Results

The turbine weight, shaft weight, and the magnetic force that the bearing can support based on the shaft diameter for different net power outputs are presented in Table 1. In Figure 4, the intersection point between the rotor assembly load and the magnetic force that the bearing can support is shown, which occurs around a 25 MWe sCO_2 power system.

Table 1: Weight and Forces of Each Component as the sCO_2 Net Work Increases

Net Work (MWe)	Turbine weight (N)	Shaft weight (N)	Rotor Weight (N)	Magnetic force (N)	Magnetic area (mm^2)	Weight/ support ratio
0.3	5.348	3.098	8.445	18.76	32.74	0.45
1	24.09	10.32	34.41	62.52	109.1	0.55
3	95.10	30.98	126.1	187.6	327.4	0.67
10	428.3	103.3	531.6	625.2	1091	0.85
20	1019	206.5	1225	1250	2182	0.98
30	1691	309.8	2001	1876	3274	1.07
50	3202	516.3	3719	3126	5456	1.19
100	7617	1032	8649	6252	10912	1.38
200	18116	2065	20181	12504	21823	1.61
300	30072	3098	33170	18756	32735	1.77

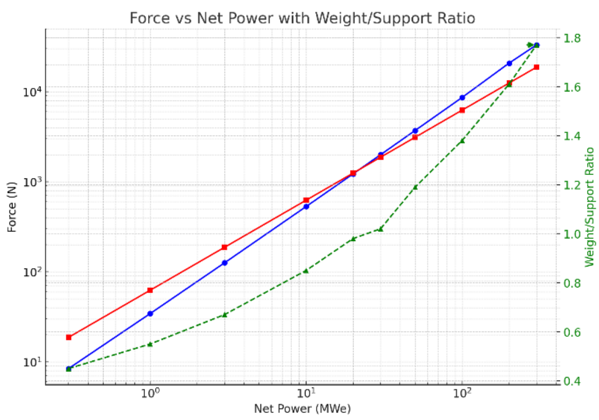

Fig. 4. Comparison of rotor weight and allowable magnetic force for different sCO_2 power levels

The turbine weight is proportional to the 0.75 power of the power output, while the shaft weight is proportional to the 0.5 power of the power output. The magnetic force that the bearing can support is proportional to the 0.5 power of the power output, meaning the exponent for the turbine weight is higher compared to the other weights and support forces. As a result, for outputs above 100 MWe, the turbine mass increasingly becomes dominant, leading to a situation where the weight grows to a level that becomes difficult for the magnetic bearing to support.

5. Summary and Future works

This study investigates the limit of active magnetic bearings for supporting the turbomachinery of an sCO_2 power cycle. The focus is on the comparison of rotor assembly weight and the supporting magnetic force. The results highlight the significance of considering the flux density saturation and flux area when designing magnetic bearings for higher power outputs. The intersection of the rotor weight and the magnetic bearing capacity is found to occur around a 25 MWe output. This research provides a foundation for likelihood of adopting magnetic bearing to sCO_2 power systems.

This study assumes simplified geometries and static loading conditions to estimate the rotor assembly weight and the supportable magnetic force by active magnetic bearings. However, actual turbine structures are considerably more complex, consisting of multiple stages, casings, and auxiliary components that can significantly increase the total mass.

Moreover, the current analysis only considers static gravitational loads. In realistic operating environments, additional dynamic effects such as rotor vibration, shock loads, gyroscopic forces, and thermal expansion must be considered. These effects can introduce transient overloads that exceed the designed bearing capacity if no additional design margin is considered. Thus, the current 25 MWe limit derived from the rotor weight and magnetic bearing force intersection should be seen as a theoretical threshold under ideal conditions.

Additionally, the adoption of hybrid bearing systems (e.g., magnetic bearings supplemented with auxiliary mechanical backup supports) may be explored to extend the applicability of active magnetic bearings to higher-power sCO_2 systems. Experimental validation or advanced multiphysics simulations (e.g., FEM, CFD) could also enhance the practical feasibility and help guide the design of robust bearing structures.

NOMENCLATURE

Ns : specific speed [-]
Ds : specific diameter [-]
ω : angular velocity $[rad/s]$
$\dot{m}$: mass flow rate of turbine $[kg/s]$
Δh : isentropic enthalpy difference $[J/kg]$
ρ : density $[kg/m^3]$
T : torque $[Nm]$
τ_{max} : allowable torsion stress (250MPa) $[N/m^2]$
A_g : Area of the magnetic flux $[m^2]$
s : gap between rotor and stator (magnetic bearing) $[m]$
μ_0 : permeability of free space ($4\pi \times 10^{-7}$) $[H/m]$
D : diameter $[m]$
i : electric current [A]
B : Magnetic flux $[kg * m^2/s^2/A]$
B_{sat} : Saturated magnetic flux $[kg * m^2/s^2/A]$

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