A Foil-Magnetic Thrust Bearing Using LCR Circuit

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***Abstract* – An ac magnet based on the LCR resonance circuit shares load with a foil thrust bearing with eight pads, which supports the weight and impeller thrust load of a vertically oriented gas compressor. For the stability of the combined bearing, the amplitude of excitation voltage to the LCR circuit is controlled as a function of rotor speed. To augment the low damping inherent in the ac suspension, a velocity feedback control is applied to modulate the excitation voltage and achieve electric damping.**

***Index Terms* – ac magnet, foil thrust bearing, foil-magnetic bearing, LCR circuit**

I. Introduction

A foil bearing is a type of gas-lubricated bearing with a flexible support structure. For example, a radial foil bearing typically consists of a smooth stainless steel foil about 0.1-mm-thick wrapped around a shaft bearing journal [1]. Between the smooth foil and cylindrical bearing housing are corrugated steel foils (bump foils; also about 0.1-mm thick) that provide the flexible support.

Similar to the design shown in Figure 1, a thrust foil bearing is comprised of several (typically eight) bearing pads mounted on a flat steel plate. Each pad consists of a segment of smooth top foil with an underlying segment of bump foil. The bumps at the pad leading edge are shorter in height, creating a taper that promotes gas film formation between the pad and a thrust disk for bearing action. Compared to conventional gas bearings with rigid support, foil bearings can better tolerate thermal distortion and gas debris and support higher load.

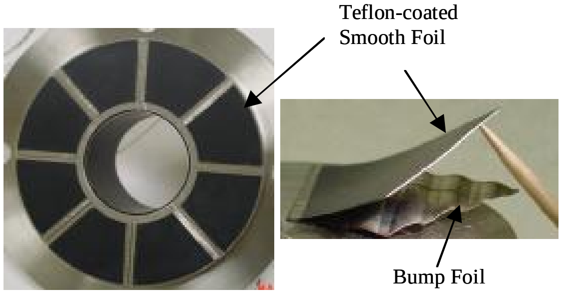

Fig. 1 Eight-pad foil bearing

Since the gas (e.g., air) viscosity is low, foil bearings are designed to operate at high speeds. It is advantageous to use foil bearings instead of rolling element bearings as backup or auxiliary bearings in high-speed active magnetic bearing (AMB) systems [2], because there is no inertia-related skidding/wear during the rapid backup bearing acceleration that results when AMBs lose power.

A foil-magnetic bearing [3,4] combines foil and magnetic bearings in a concept first developed in the 1990s to allow the high-speed gas turbine rotors of aircraft engines to operate in a high-temperature environment. This combined or so-called "hybrid" bearing can support higher load than either foil bearings or AMBs alone and can serve as an enabling technology for high-energy density machines. For radial support, the combined bearing can be configured to have the foil part side by side with the magnetic part. Or, it may have the foil part located within the magnetic gap of the AMB, because the foil layers can be made thin.

For thrust applications, the foil-magnetic bearing is a combination of foil "push" and magnetic "pull" that naturally adds their load capacities together. At high speeds, a foil bearing can support as much load as an AMB for the same projected bearing area. Properly implemented, the combined bearing can therefore carry more load per pound of weight than a conventional AMB or foil bearing.

With foil-magnetic bearings, there is no need for additional rolling-element bearings for backup, which makes extra weight and axial space savings possible. Most importantly, since the foil bearing inherently favors high-speed operation and the AMB behaves well at low frequencies, it is complementary to combine the two and form a better bearing in terms of dynamics, which is crucial for a reliable high-speed rotor-bearing system.

A foil-magnetic bearing can be challenging to implement in terms of the hardware required. In addition to the required conventional control electronics, it also needs a supervising digital controller plus rotor speed and coil current sensing to manage load sharing between the foil and magnetic bearing parts. One challenge here is to minimize the complexity of the bearing control so that it can become a practical stand-alone bearing product.

This paper presents a concept to simplify the design by replacing the conventional AMB with an ac magnetic suspension [5]. This suspension is, in essence, a tuned LCR circuit where L represents the inductance of magnetic cores (made of ferrite, powdered iron or laminated steel), magnetizing coil, and bearing air gap; C represents an added capacitance, which determines the circuit resonance frequency; and R is the resistance of the coil with or without additional external resistors, which dictates the Q or amplification factor of the resonance response. The advantage of this suspension approach is that no shaft position or coil current sensors are needed. This technique has been applied successfully to a magnetically suspended blood pump [6].

Through a practical example, we will demonstrate the process of designing a combined foil-magnetic thrust bearing using the LCR circuit. The process involves calculation of the foil bearing stiffness and damping properties, sizing of the ac magnet, and analysis of system stability and load sharing. The example bearing is a foil thrust bearing used to support the weight and impeller thrust load of a vertically oriented gas compressor. An ac magnetic bearing will be configured to reduce the weight on the foil bearing during starts and stops (and thus reduce the foil wear), and share the total load at high operating speeds. The combined thrust bearing can also be used in small gas turbine engines and other applications where high thrust load is required.

To augment the low damping inherent in the ac suspension [7], a velocity feedback method using an existing condition-monitoring sensor will be presented to show how to modulate the excitation voltage to the LCR circuit and achieve electric damping.

II. FOIL-MAGNETIC THRUST BEARING

The vertical compressor rotor is supported by two radial foil bearings and a foil thrust bearing as shown in Figure 2. The rotor weighs 44.5 N. The compressor generates a downward thrust load proportional the rotor speed squared, which can be represented by:

$$F_t = (89)(rpm/50,000)^2 \text{ N} \qquad (1)$$

In words, the hydrodynamic thrust load is 89 N or a total of 133.5 N including the rotor weight to act on the thrust bearing at 50,000 rpm.

As shown in Figure 1, the foil thrust bearing has eight pads equally spaced on a flat steel plate between an ID of 38.1 mm and an OD of 76.2 mm. Each pad has a smooth top foil (0.1-mm thick) supported by a corrugated foil (0.076-mm thick). The latter is called the “bump” foil, and the different bump heights at the leading edge of this foil form a wedge to enhance hydrodynamic bearing action. The gas lubrication theory of the thrust bearing with a compliant support has been detailed in [8]. A computer code that implements this theory is used to estimate the thrust load, gas film thickness, power loss, and so forth, for each pad [9].

In this example, the foil thrust bearing sees one atmosphere air pressure. Its load versus the air film thickness was calculated for five different speeds and plotted in Figure 3. At a film thickness of 0.03 mm, the bearing can barely support any load as the load capacity is a very nonlinear function of film thickness. In other words, the gradient of load versus film, i.e., the thrust bearing stiffness, changes sharply in a small range, i.e., 0.03 mm. This introduces some difficulty in the load sharing with a generally softer magnetic bearing. It should be emphasized that although the foil bearing has the capacity to take high loads, the associated thin film does place a burden on reliability. Therefore, it is preferable for the foil bearing to share the load with a magnetic bearing, allowing it to work with a thicker film.

The combined bearing concept in Figure 4 shows an ac magnetic bearing on top that pulls the rotor upward while the foil bearing pushes at the bottom. The load sharing can be represented the two-spring model in Figure 4. The two springs are in parallel to share the load of weight and axial

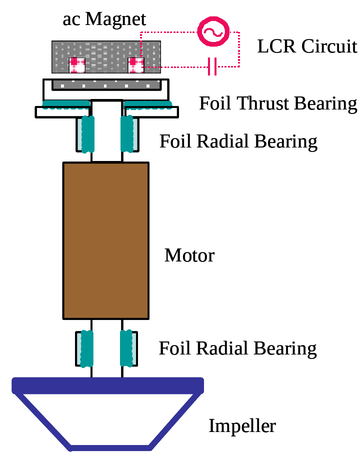

Fig. 2 Vertical compressor rotor supported by combined foil-magnetic thrust bearing

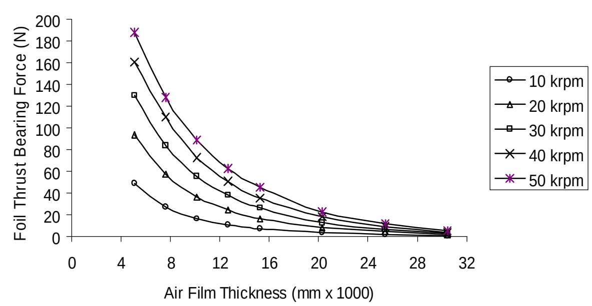

Fig. 3 Foil thrust bearing force versus film thickness

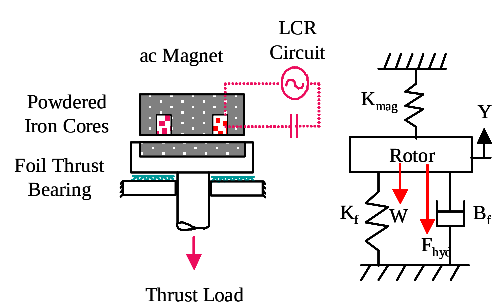

Fig. 4 Combined foil-magnetic thrust bearing and its equivalent mathematical model

hydrodynamic force of the compressor impeller and should have comparable stiffness. The ac magnetic bearing should not totally unload the foil bearing, because it does not have damping of its own and the combined bearing can become unstable.

III. AC MAGNET AND LCR CIRCUIT

The ac magnet cores are made of powdered iron with a saturation flux density of about 1.2 Tesla. Its stator has two standard circular poles with equal area. A stainless steel ring is shrunk on the outside diameter of the rotor core to protect it from bursting under centrifugal load. The design parameters of the magnet are listed in Table I.

TABLE I
ac MAGNET PARAMETERS

Outside diameter	67.3 mm
Pole area	0.161×10^{-3} m^2/pole
Nominal gap	0.254 mm
# of coil turns	175
Wire size (bare Cu)	0.508 mm
Coil resistance	2.5 Ω
Inductance	14.4 mH
Load capacity	134 N

To form the LCR resonance circuit, we add a capacitor of 5 x 10^{-6} farad and a resistor of 2.5 Ω in series with the ac magnet. At the nominal gap, the resonance frequency is

$$f_n = \frac{1}{2\pi}\sqrt{\frac{1}{LC}} = 595\ Hz \qquad (2)$$

We set the excitation frequency at 675 Hz. When the gap increases, the magnet inductance decreases. The resonance frequency increases and becomes closer to 675 Hz. Thus, the coil current and the ac magnetic force increase, and the stator will pull the rotor back to its nominal gap size. When the gap decreases, the reverse is true.

The reason that we add 2.5 Ω to make the total resistance of 5 Ω is to achieve a resonance Q-factor of about 10. Without the additional resistor, the Q is too high and the ac magnet force can be too sensitive to gap variation. Figure 5 shows how the ac magnet force can vary by changing excitation voltage or frequency. Figure 6 presents the ac peak current and copper power loss versus magnetic gap with the excitation at 675 Hz and 15 V.

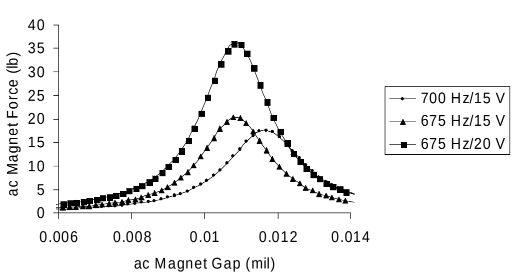

Fig. 5 ac magnet force as a function of magnetic gap, excitation frequency, and voltage

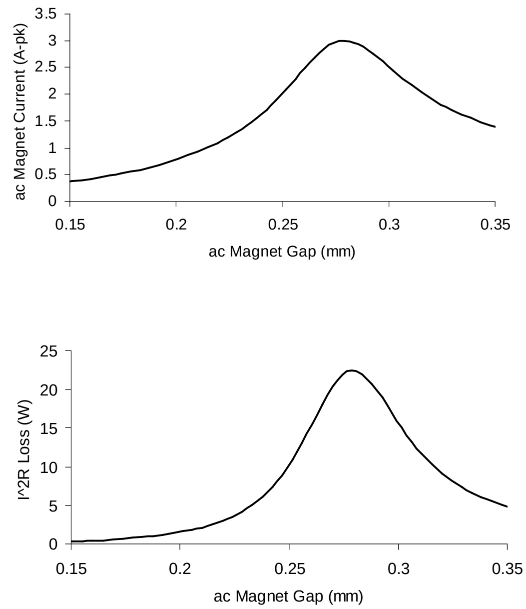

Fig. 6 ac magnet current and copper loss versus magnetic gap

IV. LOAD SHARING AND STABILITY

As previously mentioned, the foil thrust bearing is equivalent to a non-linear spring. To ensure a proper amount of load sharing between the foil and magnetic parts and system dynamic stability, a non-linear transient analysis has been performed. The system dynamics is represented by the following two equations.

$$\frac{d(LI)}{dt} + RI + \frac{1}{C}\int I dt = E_o SIN(2\pi f t) \qquad (3)$$

$$M\frac{d^2Y}{dt^2} = -W - F_{hyd} + F_f - B_f\frac{dY}{dt} + F_m \qquad (4)$$

where:

L	=	$(4\pi x 10^{-7})N^2A/(2Gap)$
N	=	coil turns
A	=	ac magnet ring pole area
Gap	=	Gapo – Y
Gapo	=	nominal magnetic gap
I	=	coil current
R	=	total resistance
C	=	capacitance
E_0	=	excitation voltage
f	=	excitation frequency
t	=	time
M	=	rotor mass
Y	=	rotor displacement
W	=	rotor weight
F_{hyd}	=	hydrodynamic thrust load
F_f	=	foil bearing force
B_f	=	foil bearing damping

and

$$F_m = ac\,magnetic\,force = -\frac{1}{2}\frac{dL}{d(Gap)}I^2 \qquad (5)$$

The instantaneous foil bearing force F_f is calculated by interpolation of the data presented in Figure 3 with the film thickness equal to 0.03+Y. This calculation also yields an instantaneous foil bearing stiffness K_f. The foil bearing damping coefficient is calculated as

$$B_f = 0.1\ K_f/\omega_n \qquad (6)$$

Note that ω_n equals 2π times the system natural frequency, which is about 150 Hz. The factor of 0.1 is a conservative estimate based on experience.

Equations (3) and (4) are solved by integration in time with zero initial conditions. The results at 50,000 rpm and 15 V (0-peak) excitation are presented in Figure 7. The displacement Y settles at –0.015 mm, which means the foil is compressed by 0.015 mm and the foil takes a load of 46.7 N. The ac magnet takes the rest, i.e., 86.8 N, as shown by the average magnetic force curve. The steady-state current in the coil is ±2.7 A, which implies an I^2R loss of about 18 W. The corresponding foil bearing power loss was calculated as 72 W, much higher than the ac magnetic loss.

If we use the same 15 V for excitation at some lower speeds, the foil would be totally unloaded and the system becomes unstable. We may elect to have the hydrodynamic load taken by the ac magnet. Since the magnetic force (thus the hydrodynamic force) is proportional to current squared and the current is proportional to the voltage, we may set the linear relationship between the excitation voltage and speed as:

$$E_0 = (15)(rpm/50{,}000)\ V \qquad (7)$$

The lower speed results are summarized in Table II.

TABLE II
SUMMARY OF LOAD-SHARING RESULTS

rpm	Y (mm)	W (N)	F_{hyd} (N)	F_f (N)	F_m (N)
50000	-0.015	44.5	89	46.7	86.8
30000	-0.019	44.5	32	44.5	32
10000	-0.025	44.5	3.6	44.5	3.6
0	-0.003	44.5	0	16.5	28

Note that at zero speed, there is no hydrodynamic gas film to separate the thrust disk from the foil. Although there is a layer of Teflon coating on the foil, it is still advisable to minimize the load so as to minimize the foil wear during starts and stops. Here, we have set the excitation to 10 V at zero to a lift-off speed. The latter was estimated to be 4000 rpm. The required excitation voltage versus speed as plotted in Figure 8 and must implemented with additional electronics.

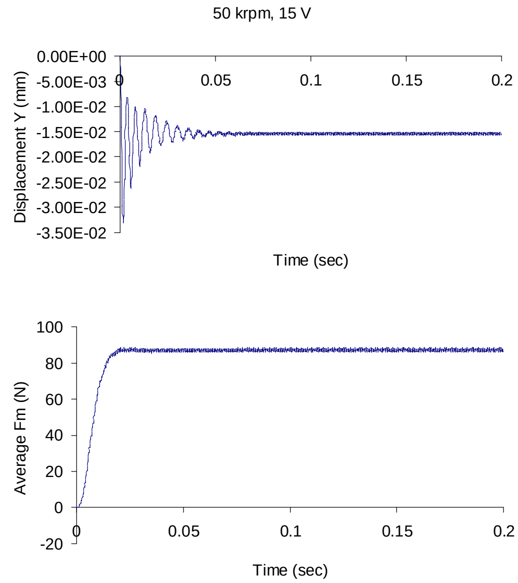

Fig. 7 Rotor transient responses in combined bearing/axial displacement and ac magnet share of load at 50,000 rpm and 15 V excitation

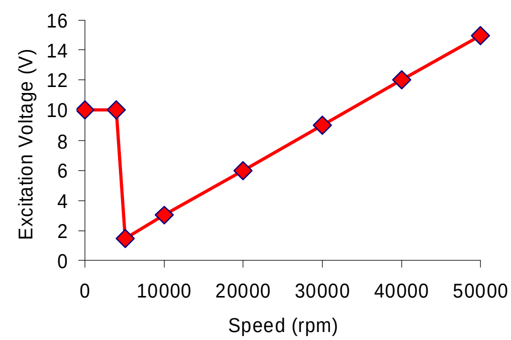

Fig. 8 Stability-required LCR excitation voltage as a function of rotor speed

V. ELECTRIC DAMPING

One of the drawbacks of the ac magnetic bearing is its lack of damping. The sensorless bearing does have its displacement information imbedded in the current signal. In other words, when the rotor vibrates at a frequency, one observes that the current signal amplitude modulates at that frequency, and the vibration amplitude is proportional to modulating amplitude. Therefore, one may obtain the displacement information by recreating the modulating envelope of the current signal. The envelope signal is then fed back to stabilize bearing function [10]. However, this boot-strapping approach is electronically difficult and may not be worth the effort. On the other hand, displacement sensors often exist for condition-monitoring purposes in turbomachinery. In this situation, we may take advantage of this arrangement and generate a velocity feedback mechanism as shown in Figure 9.

We can modulate the excitation voltage by a feedback, which is defined as:

$$\Delta E = -C_d \frac{dY}{dt} \qquad (8)$$

where C_d is a constant. In words, we will replace the excitation voltage E_o of equation (3) by $E_o + \Delta E$. The advantage effect of the electric damping is demonstrated in the simulation results presented in Figure 10.

At zero speed, if the excitation voltage is increased from 10 to 13 V, the foil bearing will be totally unloaded. Since the ac magnet does not have its own damping, the system goes unstable as shown by the displacement and force plots of Figure 10 with Cd = 0. The plots corresponding to the feedback case with Cd = 30 show that the rotor is lifted off the foil by 0.0025 mm, and the ac magnetic bearing is stable and takes the rotor weight of 44.5 N all by itself.

VI. SUMMARY AND CONCLUSIONS

We have analytically shown the feasibility to use an ac magnet and share load with a foil thrust bearing. The addition of the ac magnet enables the thrust bearing to take higher load, consume less power and extend the service life of the foil bearing.

The load sharing between the foil and ac magnetic parts is complicated by two aspects: 1) the foil thrust bearing is equivalent to a highly nonlinear spring, and 2) the ac magnetic bearing has no damping of its own.

For dynamic stability, the thrust disk must be kept in contact with the foil. The ac magnet must be stiff enough to take part of the load, but not to unload the foil at all operating speeds. To satisfy this requirement, a simple "supervising controller" (in additional to the LCR circuit electronics) is needed to change the ac magnet excitation voltage as a function of speed.

When an axial displacement sensor is available, one may create electric damping for the ac magnet by using velocity feedback to modulate the excitation voltage. The electric damping enhances the system dynamic stability and the flexibility of load sharing.

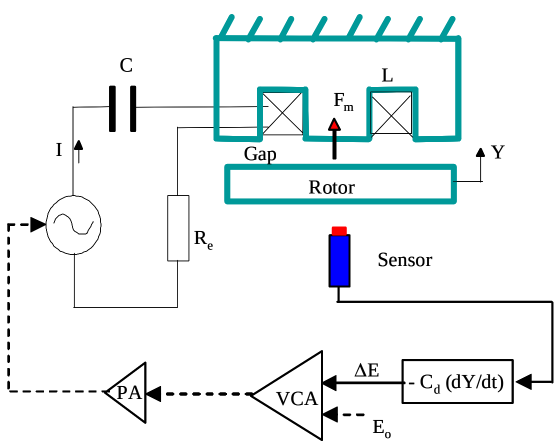

Fig. 9 Velocity feedback for electric damping.

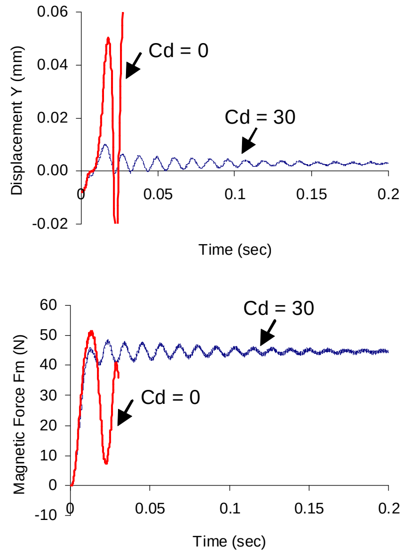

Fig. 10 Stability effect of electric damping (zero speed, 13 V)

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