Influence of different operating conditions on centrifugal compressor surge control with active magnetic bearings

Xudong Guan, Jin Zhou, Chaowu Jin, Yuanping Xu and Hengbin Cui

College of Mechanical and Electrical Engineering, Nanjing University of Aeronautics and Astronautics, People’s Republic of China

ABSTRACT
The purpose of this paper is to study the surge control effects of centrifugal compressors with active magnetic bearings (AMBs) under several operating conditions. First, the working principle of the magnetic thrust bearing is introduced, and the tracking performance of the rotor under the low frequency is verified by experiments. Then, the Greitzer model of centrifugal compressor is established. The relationship between tip clearance and the performance of compressor is derived, and the proportional-integral-derivative surge controller is designed based on the mass flow feedback. Finally, the surge control platform of the centrifugal compressor with active magnetic bearings is set up, and the surge control experiment of the magnetic bearing system is carried out. The influence of different speeds and different throttle valve openings on surge control performance is analyzed. The experimental results illustrate that speed has little effect on surge control, and the different throttle valve opening sizes have a certain influence on surge. The surge controller designed in this paper has a good control effect on different throttle valve opening sizes. Thus, the magnetic thrust bearing is effective in surge control by adjusting the tip clearance of the compressor.

ARTICLE HISTORY
Received 7 October 2018
Accepted 30 June 2019

KEYWORDS
Centrifugal compressor; magnetic bearing; speed; throttle valve opening; surge control

1. Introduction
In recent years, the magnetic levitation technique has been increasingly applied in the field of fluid machinery (Li, Wang, Weaver, & Shao, 2018). Fluid machinery with AMBs installs the rotating part of the fluid machinery on the extension of the magnetic bearings, and it is driven directly by a high speed motor. Fluid machinery with AMBs is characterized by having no mechanical contact, no friction, no wear, a long expected life, no lubrication, high efficiency, low noise, active control, and so on (Xu, Di, Zhou, Jin, & Guo, 2016). However, the surge problems of fluid machinery can cause serious damage, such as irreversible damage to blades and bearings (Torrisi et al., 2017). Common anti-surge control methods include surge avoidance and surge control (Gravdahl & Egeland, 1999). Surge avoidance includes a fixed-limit flow method and variable-limit flow method (Kurz & White, 2004). A surge margin needs to be considered for such methods, which not only reduce the compressor’s working area, but also results in the bad performance of a compressor because it cannot run at the best working point. Surge control can be divided into passive control and active control. We focus on the study of active surge control in this paper, due to the advantages of active control, such as active regulation (Gravdahl & Egeland, 1999).

Active surge control is to control the unstable air flow which causes surge so as to prevent the occurrence of surge (Gu, Sparks, & Banda, 1999). However, there are still some problems in the practical application of active surge control methods, including the difficulty of actuator installation, high cost, complicated algorithms, and power loss. Therefore, it is necessary to propose a more convenient, more reliable surge control mode that has a faster response speed and can reduce power loss.

For a centrifugal compressor with AMBs, magnetic thrust bearings can be applied to adjust the tip clearance between the impeller tip and the static shroud of a compressor to achieve surge control. The theoretical basis is given in some literature (Sanadgol & Maslen, 2004; Sanadgol & Maslen, 2005a; Yoon, 2011). No additional hardware is needed in this method, but the control strategy of a magnetic thrust bearing needs to be modified without sacrificing efficiency, and the response speed is fast (Sanadgol, 2006). However, the relevant research is still scanty. In the process of surge control by adjusting tip clearance, the effect of different working conditions on the surge control of a magnetic suspension compressor
is less studied. Because rotational speed affects the characteristic curve of the compressor, the load on the compressor impeller increases with the increase in rotational speed. Therefore, the stability and axial tracking performance of the magnetic bearing system under different rotational speeds deserve further analysis. The influence of mass flow and pressure increases on surge control performance is very important. The mass flow and pressure rise are directly related to the compressor characteristic curve, which will also affect the compressor working equilibrium position. The gas pressure of fluid machinery is a key index directly related to the operation of the equipment. Much literature (Akbarian et al., 2018; Ardabili et al., 2018; Baghban, Jalali, Shafiee, Ahmadi, & Chau, 2018; Mou, He, Zhao, & Chau, 2017) has carried out some simulation and experimental research. In this paper, the change of the throttle curve opening shape is used to simulate the change of the mass flow of a compressor, and the corresponding experimental study is carried out. In summary, the active surge control research into a centrifugal compressor with AMBs is presented in this article under the two different kinds of operating conditions presented earlier.

The magnetic thrust bearing is used as an actuator for the active surge control of the compressor. Firstly, the rotor position control performance of the magnetic thrust bearing system is theoretically studied and verified by experiments. Then the surge control model of the compressor is established and an effective surge control strategy is applied. The effectiveness of the surge controller is verified by simulation. Finally, the surge control performance under different speeds and different throttle curve opening is analyzed by experiments.

2. Control performance of a magnetic thrust bearing system

In the process of surge control, in order to change the tip clearance, applying a magnetic thrust bearing to drive the rotor to change the axial position in real time is necessary. And in the working state of the compressor, the axial pressure difference will lead to the axial load and other interference on the rotor, so it is particularly crucial to ensure the position control performance of the rotor.

The research on the position tracking of the magnetic bearing-rotor system is more common in the field of machining, where the magnetic bearings are applied to change the position of the cutting tool. Some scholars (Pesch, Smirnov, Pyrhönen, & Sawicki, 2015; Smirnov, Pesch, Pyrhönen, & Sawicki, 2015) have studied the position tracking performance of the rotor controlled by radial magnetic bearings, and studied proportional-integral-derivative (PID) control and advanced robust control such as $\mu$-Synthesis. Minihan et al. (2003) investigated the position tracking of large axial gaps using a nonlinear control algorithm. In this paper, the classical algorithm, PID control, is applied to designing the magnetic bearing controller, and the position control performance of the magnetic thrust bearing is carried out. Figure 1 is a schematic diagram of the magnetic thrust bearing. In this diagram, a different suspension position corresponds to different current, air gap, and so on.

According to Maxwell's principle, the electromagnetic force of the rotor is obtained as follows.

$$ F_x = \frac{\mu_0 A N^2}{4} \left[ \left( \frac{i_0 + i_s + i_x}{x_0 + \delta + x} \right)^2 - \left( \frac{i_0 - i_s - i_x}{x_0 - \delta - x} \right)^2 \right] $$

(1)

where $i_0$ is the bias current, $i_s$ is the control current corresponding to the suspension position, $\delta$ is the variation of the suspension position relative to the middle position, $i_x$ is the current disturbance of the suspension position, and $x$ is the displacement disturbance near the suspension position. $\mu_0$ is the vacuum permeability, $A$ is the magnetic pole area, and $N$ is the number of coil turns. $x_0$ is a unilateral air gap suspended in the middle position. In this experiment, $x_0 = 0.45$ mm, that is, the displacement range of the rotor shaft is 0–0.9 mm. The displacement signal is converted into a voltage signal of 0–5 V by a displacement sensor. Therefore, the displacement sensor has a resolution of 0.18 mm/V. Therefore, when the rotor is suspended in the middle position, the corresponding displacement signal is 2.5 V.

From Equation (1), we can see that the electromagnetic force of the rotor changes nonlinearly with the change of the suspension position, thus it is necessary to experimentally study whether the control can meet the position tracking requirement under enough displacement strokes. A position control experiment is carried out on a five-degree-of-freedom magnetic bearing-rotor test rig; the principle of the experiment is illustrated in Figure 2.
Compressor surge instability initiates a low frequency and large amplitude oscillations in the observed pressure rise and mass flow. Thus, only the low frequency is taken into consideration in research on the position tracking performance of magnetic thrust bearing. In this paper, the square wave signals and the sinusoidal signals are taken as examples to test the position tracking performance of magnetic thrust bearing. The experimental results are illustrated in Figure 3.

In Figure 3, the displacement signals are converted to voltage signals by displacement sensors. According to the position tracking results of square wave signals and the sinusoidal signals, the sinusoidal signals’ error fluctuation is less than 0.1 V. The square wave signal is a mutational signal, and the error is a little large. Overall, the tracking performance of both signals is not significantly delayed at low frequencies.

3. Surge control model with variable tip clearance

When the gas in the compressor system is limited and the pressure increment in the plenum reaches the critical point, the compressor cannot generate enough gas pressure to resist the gas pressure in the plenum. In this case, the gas will be forced to flow towards the compressor inlet, interfering with the gas flow throughout the system, and the gas inside the compressor system begins to oscillate. This phenomenon is called surge. The pressure critical point of surge is called the surge point. The surge point is close to the peak value of the compressor characteristic curve, and the surge point separates the compressor stable operation zone from the compressor unstable region.

3.1. Effect of tip clearance on the compressor performance

Figure 4 is the classical compressor system structure diagram, including the compressor, the pipeline, the plenum volume and the throttle valve. Both the pressure rise and the mass flow are expressed as dimensionless. \( \Psi_c \) is the compressor pressure rise, \( \Phi_c \) is compressor mass flow, \( \Psi_p \) is the pressure rise of the plenum, and \( \Phi_{th} \) is the mass flow of the throttle valve.

The dimensionless pressure rise \( \Psi \) and dimensionless mass flow \( \Phi \) are represented as (Greitzer, 1976):

\[
\Psi = \frac{\Delta p}{\frac{1}{2} \rho_o U^2}
\]  

Figure 4. Compressor system structure diagram.
\[ \Phi = \frac{m}{\rho_0 U A_c} \]  

(3)

where \( \Delta p \) is the pressure increment, \( m \) is mass flow, \( U \) is the linear velocity of the compressor impeller tip contour, \( \rho_0 \) is the air density, and \( A_c \) is cross section area of the compressor pipeline.

We select the parameters of a single stage centrifugal compressor, GT70, from the American Honeywell Company. The compressor characteristic curve is obtained (Tangjun, 2011) as follows.

\[ \Psi_c(\Phi_c) = 1.656 \times \left\{ 2.22 + 0.2 \left[ 1 + \frac{3}{2} \left( \frac{\Phi_c \times 4.91}{0.75} - 1 \right) \right] \right\}, \Phi_c \geq 0 \]  

(4)

\[ \Phi_c < 0 \]  

(5)

Under the subsonic flow condition, the dimensionless mass flow of throttle \( \Phi_{th} \) is the function of throttle curve opening \( u_{th} \) and dimensionless pressure rise \( \Psi p \).

\[ \Phi_{th} = c_{th} u_{th} \sqrt{\Psi p} \]  

(6)

\( c_{th} \) is throttle valve constant which is chosen as 1.2. The compressor characteristic curve and throttle curve (load curve) can be obtained as presented in Figure 5. The intersection point of these two curves is the stable operating point of the compressor.

It can be seen from Figure 5 that with a decrease in the throttle valve opening, the throttle curve moves up. It can be concluded that if the throttle valve opening \( u_{th} \) reaches 15\%, the operating state would be out of the characteristic curve and then surge occurs.

In the centrifugal compressor, the influence of tip clearance is usually expressed by changes in the compressor efficiency. A simple model deduced by Senoo and Ishida (1987) to show the relationship between compressor efficiency and the change of tip clearance is represented as:

\[ \eta = \frac{\eta_n}{1 - k_0 \frac{\delta_{cl}}{b_2}} \]  

(7)

where \( \delta_{cl} \) is the change of tip clearance, and \( \delta_{cl} = c_{ln} - cl. k_0 \) is chosen to express the tip clearance, and \( k_0 = \frac{0.25}{1 + 0.25 \gamma} \) (Senoo & Ishida, 1987).

Ideally, the influence function of the tip clearance on compressor pressure rise could be derived as:

\[ \varphi = \frac{p_c}{\rho_0} = \left( 1 + \frac{\gamma^2 \varphi c_{ss} - 1}{1 - k_0 \frac{\delta_{cl}}{b_2}} \right)^\gamma \]  

(8)

where \( \varphi_c \) is the pressure ratio of the compressor, \( \varphi_c = \frac{p_c}{\rho_0} \), \( \varphi_c_{ss} \) is the pressure ratio of the compressor when the tip clearance is \( cl \). The relationship of the dimensionless pressure rise and the pressure ratio could be derived according to Equation (2). Then, with Equation (8), the influence equation of tip clearance on the compressor dynamics is obtained as follows.

\[ \Psi_c = \frac{p_0}{\frac{1}{2} \rho_0 U^2} \left( 1 + \frac{1}{b_2} \frac{\gamma - 1}{\gamma} \right) \left( \Psi - 1 \right) - 1 \right \} \]  

(9)

where \( U \) is linear velocity of the compressor impeller.

In order to facilitate the design of the linear surge controller, the model is linearized by neglecting the non-linearity of the system. When \( \delta_{cl} = 0 \), through the Taylor expansion of Equation (9), it is obtained that:

\[ \Psi_c = \Psi_{c_{ss}} + \frac{p_0}{\frac{1}{2} \rho_0 U^2} \left( - \frac{\gamma - 1}{\gamma - 1} \frac{1}{b_2} \left( \frac{\gamma - 1}{\gamma} \Psi_{c_{ss}} - 1 \right) \right) \delta_{cl} \]  

(10)
The classical Greitzer model is introduced into the linearized tip clearance equation:

\[ \dot{\Phi}_c = B\omega_H \left( \Psi_{c,ss} + \frac{\rho_1}{2\rho_0} k_{cl} \delta_{cl} - \Psi_p \right) \]  

where \( \omega_H \) is Helmholtz frequency, and \( B \) is the stable constant.

From Equation (11), we can draw the graph of the characteristic curve affected by the tip clearance, as shown in Figure 6. It can be clearly seen that the pressure rise and mass flow will change as the tip clearance changes. When the mass flow rate is constant, the pressure rise of the compressor slowly decreases with the increase in the tip clearance, that is, the output pressure decreases. The reason is that as the tip clearance increases, the compressor efficiency decreases. On the contrary, the pressure rise increases as the tip clearance is reduced. It can also be seen that the compressor characteristic curve changes linearly with the tip clearance.

### 3.2. Surge control strategy

First, the appropriate surge control algorithm should be chosen. The literature (Sanadgol, 2006) proves that the control effect of mass flow feedback control is better than that of output pressure feedback control. In this paper, the mass flow feedback control is applied to simulate the surge control system. And this control strategy is simple in structure, easy to implement, and does not require a precise system model.

\[ \dot{\Psi}_p = \Psi_p - \Psi_{eq} \]  
\[ \dot{\Phi}_c = \Phi_c - \Phi_{eq} \]

\( \Psi_{eq} \) and \( \Phi_{eq} \) are pressure rise and mass flow of equilibrium working point respectively. Mass flow feedback control strategy is defined as:

\[ \delta_{cl} = -k_p \dot{\Phi}_c \]  

The stability of the system can be judged by the Lyapunov stability principle (Sanadgol & Maslen, 2005b). The surge control system of a centrifugal compressor with AMBs includes the magnetic thrust bearing and its controller, compressor model, and surge controller. The block diagram is shown in Figure 7.

The actual magnetic thrust bearing not only has the bandwidth limitation, but also has some other characteristics. For example, the resonance peak value in the frequency domain response may affect the surge control. Thus, it is necessary to study the influence of the accurate model of the magnetic thrust bearing system on surge control. First, the rotor test rig is controlled by the PID control method. Then, according to the system closed-loop frequency sweep experiment, the closed-loop transfer function of the system from the position reference signal to the actual rotor displacement is obtained. Finally, a mathematical model is established. And the corresponding simulation verification and experimental study can be carried out in combination with the PID surge controller.

The structures of the AMB controller and surge controller are presented in Equation (15). And in the surge controller, \( K_p = 15 \), \( K_i = 0.5 \), \( K_d = 0.1 \), \( f = 1000 \).

\[ C(s) = \frac{K_p}{s} + \frac{K_i}{s} + \frac{K_d}{2\pi f s + 1} \]  

Figures 8–9 display the simulation results without and with surge control respectively. Their throttle curve opening size is changed from 18% to 12%. It can be observed from Figure 8 that when surge control is not applied, the pressure rise and mass flow begin to oscillate gradually with the decrease of the throttle curve opening, and it appears as a surge ring on the characteristic curve. When surge control is applied to the system, the pressure rise and mass flow of the compressor system are always forced to move around the equilibrium position, as shown in the enlarged characteristic curve graph in

---

**Figure 6.** The relationship between tip clearance and the characteristic curve.

**Figure 7.** Surge control block diagram.
4. Experimental investigation

In this section, we use the model established in the third section. Specifically, according to the axial position adjustment calculated by the surge controller, the corresponding position adjustment is carried out so as to avoid the occurrence of surge and realize active control. In the process of the experiment, the rotational speed signal detected by the speed sensor is introduced into the surge control model through dSPACE 1202 in real time, and the amount of displacement that is needed to be adjusted is calculated. Then the displacement is introduced into the magnetic bearing controller as the reference signal to adjust the axial displacement in real time, and the corresponding experiments are carried out in different speeds and different throttle opening sizes. The principle of the experiment is presented in Figure 10 (a). Figure 10 (b) shows the test rig. Major parameters of the magnetic thrust bearing are shown in Table 1. The length of the rotor is 1.038 m. And the mass of the rotor is 14.56 kg.

The control test process is as follows. Firstly, the surge suppression simulation is realized by adjusting the tip clearance of the compressor to suppress the fluctuation of pressure rise and mass flow in the compressor model. Then, the reference signal of variable tip clearance is introduced into the actual magnetic thrust bearing system. Finally, the axial position adjustment of the rotor, that is, adjustment of tip clearance, is studied to observe its tracking performance by considering different operating conditions.

4.1. Surge control with different speeds

From the theoretical analysis, it can be seen that the control of the axial position of the compressor is related to the rotational speed. Therefore, it is crucial to verify the stability of the axial control process of the magnetic thrust bearing at different rotor speeds.

Figure 11 indicates the relationship between different speeds and compressor characteristic curves. It can be concluded from this figure that the speed has little effect on characteristic curves. However, attention should be paid to whether the magnetic bearing system can
operate stably at different speeds. For this reason, it is still necessary to verify the tracking stability at different speeds.

The surge control performance of the rotor at three rotational speeds of 3000, 6000, and 9000 r/min is verified respectively. Under the PID control and the gradual throttle valve opening, the performance of the magnetic compressor surge control with different rotational speeds is shown in Figure 12.

As can be seen from Figure 12, at different speeds, the axial position adjustment of the rotor is stable, and has a
good tracking performance. It only shows a slight lag. The adjustment of axial displacement is smaller in the case of gradual throttle valve opening, fluctuating in the range of 2.4 V to 2.6 V. The tracking error is also small, within 0.05 V, which is easier to control to some extent.

Comparing the three different speeds, it can be concluded that the pressure rise and mass flow rate of the compressor system under the same throttle valve opening are almost the same, and the axial position adjustment signals calculated by the surge controller are almost the same.

4.2. Surge control with different throttle opening sizes

The control of the axial position of the compressor is related to the throttle opening. Therefore, it is vital to verify the stability of the axial adjustment process under different throttle opening sizes. The change of throttle valve opening essentially changes the pressure rise and mass flow of the compressor model. For this reason, the throttle valve opening is mainly divided into two kinds: gradual opening and mutational opening, and then the research is carried out from these two aspects. Figure 13(a) and Figure 13(b) depict the surge control performance under the conditions of gradual change and mutational change, respectively.

Comparing Figure 13(a) and Figure 13(b), it is evident that the adjustment of axial displacement is smaller and the tracking error is also smaller under the condition of gradual change. This situation is easier to control. The reason is that the sudden change in the throttle valve opening will lead to the sudden change in pressure rise and mass flow, thereby deteriorating the compressor flow field distribution. And sudden change in the reference position will also directly affect the rotor stability, so it is not conducive to control.

5. Conclusion

The axial position of the rotors of a centrifugal compressor with AMBs is changed by using magnetic thrust bearings, thus the tip clearance of the compressor is changed, which finally makes an active control to surge. The method has the advantages of no additional hardware, no efficiency loss, and so on.

In this paper, the position control performance of the magnetic thrust bearing-rotor system in surge control which uses the variable tip clearance method is presented. The surge control model of the compressor is established, and the feasibility of the control strategy proposed in this paper is verified by theoretical analysis.

The experimental results illustrate that the magnetic bearing system can adjust the axial position according to the axial adjustment displacement calculated by the surge control model under different speeds and different throttle valve openings. By comparing the effects of different rotational speeds on surge control, we can see that the effect of rotational speed on the surge control is small. And the comparison of the influence of different throttle valve openings on surge control draws the conclusion that the gradual throttle opening is easier to control.

The control strategy in this paper can be applied to control the severe working conditions such as a surge in magnetic fluid machinery in an early and effective manner, which has a strong engineering significance. In the future, it is necessary to build a magnetic suspension centrifugal compressor test rig with impeller, volute, throttle valve, and its pipeline, as well as to study the surge control performance of a magnetic suspension compressor with disturbance.

![Figure 13. Surge control performance with different throttle opening sizes](image-url)
Disclosure statement
No potential conflict of interest was reported by the authors.

Funding
This work has been supported by National Natural Science Foundation of China (51675261), Jiangsu Province key R & D programs (BE2016180), Postgraduate Research & Practice Innovation Program of Jiangsu Province (KYCX17_0244).

ORCID
Xudong Guan http://orcid.org/0000-0001-7453-5540
Chaowu Jin http://orcid.org/0000-0002-2136-8125

References
Akbarian, E., Bahman, N., Mohsen, J., Sina, F. A., Shahaboddin, S., & Kwok-Wing, C. (2018). Experimental and computational fluid dynamics-based numerical simulation of using natural gas in a dual-fueled diesel engine. Engineering Applications of Computational Fluid Mechanics, 12(1), 517–534.

Ardabili, S. F., Najafi, B., Shamshirband, S., Bidgoli, B. M., Deo, R. C., & Chau, K. W. (2018). Computational intelligence approach for modeling hydrogen production: A review. Engineering Applications of Computational Fluid Mechanics, 12(1), 438–458.

Baghban, A., Jalali, A., Shafiee, M., Ahmadi, M. H., & Chau, K. W. (2018). Developing an ANFIS based swarm concept model for estimating relative viscosity of nanofluids. Engineering Applications of Computational Fluid Mechanics, 13(1), 26–39.

Gravdahl, J. T., & Egeland, O. (1999). Compressor surge and rotating Stall: Modeling and control. Springer Publishing Company, Incorporated.

Greitzer, E. M. (1976). Surge and rotating stall in axial flow compressors, part I: Theoretical compression system model. Journal of Engineering for Gas Turbines & Power, 98(2), 190–196.

Gu, G., Sparks, A., & Banda, S. S. (1999). An overview of rotating stall and surge control for axial flow compressors. IEEE Conference on Decision & Control, 3, 2786–2791.

Kurz, R., & White, R. C. (2004). Surge avoidance in gas compression systems. Journal of Turbomachinery, 126(4), 501–506.

Li, Q., Wang, W., Weaver, B., & Shao, X. (2018). Active rotodynamic stability control by use of a combined active magnetic bearing and hole pattern seal component for back-to-back centrifugal compressors. Mechanism & Machine Theory, 127, 1–12.

Minihan, T. P., Lei, S., Sun, G., Palazzolo, A., Kascak, A. F., & Calvert, T. (2003). Large motion tracking control for thrust magnetic bearings with fuzzy logic, sliding mode, and direct linearization. Journal of Sound & Vibration, 263(3), 549–567.

Mou, B., He, B. J., Zhao, D. X., & Chau, K. W. (2017). Numerical simulation of the effects of building dimensional variation on wind pressure distribution. Engineering Applications of Computational Fluid Mechanics, 11(1), 293–309.

Pesch, A. H., Smirnov, A., Pyrhönen, O., & Sawicki, J. T. (2015). Magnetic bearing spindle tool tracking through μ-synthesis robust control. IEEE/ASME Transactions on Mechatronics, 20(3), 1448–1457.

Sanadgol, D. (2006). Active control of surge in centrifugal compressors using magnetic thrust bearing actuation. Dissertations & Theses.

Sanadgol, D., & Maslen, E. (2004). Sliding mode controller for active control of surge in centrifugal Compressors with magnetic thrust bearing Actuation. Proceedings of the Ninth International Symposium on magnetic bearings, Lexington, USA.

Sanadgol, D., & Maslen, E. (2005a). Effects of actuator dynamics in active control of surge with magnetic thrust bearing actuation. IEEE/ASME International Conference on advanced Intelligent Mechatronics Proceedings, 1091–1096.

Sanadgol, D., & Maslen, E. (2005b). Backstepping for active control of surge in Unshrouded centrifugal Compressors with magnetic thrust bearing Actuation. ASME Turbo Expo: Power for Land, Sea, and Air, 883–889.

Senoo, Y., & Ishida, M. (1987). Deterioration of compressor performance due to tip clearance of centrifugal impellers. Transactions of the Japan Society of Mechanical Engineers, 52(473), 386–392.

Smirnov, A., Pesch, A. H., Pyrhönen, O., & Sawicki, J. T. (2015). High-precision cutting tool tracking with a magnetic bearing spindle. Journal of Dynamic Systems Measurement & Control, 137(5), 051017-1–051017-8.

Tanguiu, L. (2011). Surge model and stability analysis of centrifugal compressor (Master dissertation), Shanghai Jiao Tong University.

Torrisi, G., Grammatico, S., Cordinovis, A., Mercangöz, M., Morari, M., & Smith, R. S. (2017). Model predictive approaches for active surge control in centrifugal compressors. IEEE Transactions on Control Systems Technology, 25(6), 1947–1960.

Xu, Y., Di, L., Zhou, J., Jin, C., & Guo, Q. (2016). Active magnetic bearings used as excitors for rolling element bearing outer race defect diagnosis. Isa Transactions, 61, 221–228.

Yoon, S. Y. (2011). Surge control of active magnetic bearing suspended centrifugal compressors (Dissertations & Theses).