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Clearance Control of a Mechanical Face Seal©

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Seal clearance control is an advancement in noncontacting mechanical face seal operation because seal clearance variation caused by process disturbances may cause either severe face contact or excessive leakage, each of which is regarded as seal failure. The objective of this research is to control the seal clearance at a desired value overcoming operation disturbances, including variations in shaft speed and sealed fluid pressure. A flexibly mounted rotor (FMR) noncontacting mechanical face seal test rig is used in this study. The clearance control concept is to adjust the closing force that acts upon the flexibly mounted rotor. The seal clearance is measured by an eddy current proximity probe. The seal axial dynamic model is experimentally determined for the design of a proportional-integral (PI) controller with anti-windup. The controller is then applied to the test seal. Results show that the controlled seal maintains or follows set-point clearance changes with and without disturbances in sealed water pressure and shaft speed. The controlled seal is shown to respond quickly (having a small time constant) with a small control effort.

KEY WORDS

Face Seals

INTRODUCTION

The technology of mechanical face seals has considerably advanced to where they are widely used in pumps, compressors, and powered vessels. Many design variations have evolved to handle various applications and operating conditions. The very basic objective in mechanical seal design, however, still remains to maintain a full and stable fluid film at the sealing dam. Two requirements that often oppose each other are minimum leakage and minimum wear, i.e., long life.

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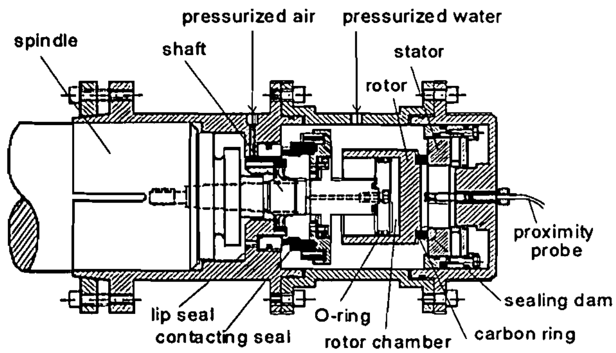
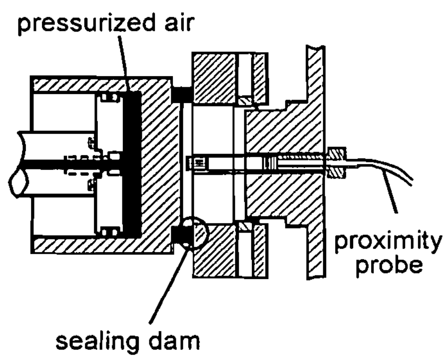
Trends and the state of art of mechanical face seal design are presented by Etsion *(1)*. It is stated that the seal clearance is a very important parameter in mechanical seal design because it strongly affects the leakage. In addition, seal clearance is important because it determines the nominal separation between the rotor and the stator, and has a direct influence upon the rotordynamic coefficients and the seal dynamic response *(2)*, *(3)* that determines the conditions under which rubbing contact might occur *(4)*. Seal failure is often characterized by excessive wear of the seal faces when the fluid film thickness is too small, or by excessive leakage when the fluid film thickness is too large. In critical applications, such as nuclear reactor cooling pumps or liquid oxygen (LOX) turbopumps where seal failure may have severe implications, a key factor in seal design is to maintain a predetermined clearance between the rotor and the stator. A seal with actively controlled seal clearance is one of the most promising ideas in future seal development *(1)*.

Several research efforts have been undertaken to develop controlled mechanical seals *(5)-(8)*. Either the opening force or the closing force have been chosen as the means of controlling the seal clearance. Salant et al. *(5)* developed an electrically controlled flexibly mounted stator (FMS) mechanical seal. In this seal, an electromechanical actuator was used to change the seal coning and, therefore, the opening force, to control the thickness of the lubricating film between the faces. The control system used information received from thermocouples that measured temperatures in the film and on the seal face. Heilala and Kangasneimi *(6)* also designed a seal with a controllable opening force. Instead of controlling the seal coning, they used compressed air supplied to the seal interface to change the opening force. The input to the control system was the seal face temperature, measured by a thermocouple. Etsion et al. *(7)* performed a feasibility study for a controlled FMS mechanical seal. Their control method adjusted the closing force acting on the flexibly mounted stator in order to maintain a desired face separation. The feedback to the control system was the face temperature that is indicative of face separation but is easier to measure than face separation. Wolff *(8)* developed an actively controlled FMS mechanical seal. The active control of the film thickness is established by controlling the radial convergence of the seal interface with a piezoelectric actuator. The seal was operated with both a manual control and a closed-loop control system which used either the leakage rate or the face temperature as the feedback.



(a) Cross section of the test rig details



(b) Blow-up of seal detail

Fig. 1—Schematic of the FMR noncontacting mechanical seal test rig. (a) cross section of the test rig details (b) blowup of seal detail

All previous research used face temperature as feedback to the control system. A deficiency with face temperature measurement is that thermocouples measure a local temperature in the sealing dam. Clearly, this is not a direct measurement of the fluid film thickness. Consequently, the feedback to the control system may not be sufficiently fast for active and quick control to take place due to large heat transfer thermal inertia. This measurement approach may induce large time constants and lags between event occurrence and control action. Therefore, face temperature may not be the most effective feedback signal for seal clearance control.

The high voltage supply needed by the actuator used in *(5)* and *(8)* may present another safety concern in sealing volatile fluids. The control scheme used in *(5)* was to increase or decrease the control input in steps as needed. Commercial PID controllers were used and tuned in *(6)* and *(8)*. Linear and nonlinear controllers were used in *(7)* but no details were given. In this work a PI controller is chosen. PI controllers are relatively easy to implement and are effective for processes whose dynamics might be quite complex, but as a first approximation can be represented as first order systems.

Closing force control has an advantage over opening force control in that it is easier to implement and it is not as sensitive to various parameters of the seal system *(7)*. A noncontacting mechanical face seal test rig with a pneumatic closing force adjusting mechanism is used in this research. Seal clearance is directly measured by an eddy current proximity probe. The experimental setup, controller design, and results of the controlled system performance are presented.

EXPERIMENTAL SETUP

The Mechanical Seal Test Rig

A noncontacting FMR mechanical face seal test rig was used in this research to study and implement the seal clearance control technique. The schematic of the test rig is shown in Fig. 1.

A 416 stainless steel rotor is flexibly mounted on a rotating shaft through an elastomer O-ring. This allows the rotor to track the stator misalignment and move axially. A carbon graphite ring is mounted on the rotor through an elastomer O-ring. The carbon ring end face is the actual rotor face. The stator is fixed in the housing. The stator and the carbon graphite ring form the sealing dam. The shaft is screwed into a spindle that is connected through two wafer-spring couplings to a motor. The motor speed is controlled by a motor speed controller.

Pressurized water is supplied from a water supply line into the housing. The water pressure is manually controlled by a pressure regulator. A water pressure gauge in the water supply line measures the water pressure.

Pressurized air is supplied from the main air supply line to the rotor chamber through holes in the housing and shaft. It is sealed by a lip seal at one end and separated from the water by a contacting seal at the other end. The air pressure is controlled by a computer through an electropneumatic transducer.

The seal operates at an equilibrium position with a certain clearance between the rotor and the stator, where the opening force and closing force are balanced. The opening force is contributed by the hydrostatic and hydrodynamic pressure components acting in the sealing dam. This force is affected by the seal geometry, coning angle, the sealed pressure, the ambient pressure, the fluid viscosity and the seal clearance. The closing force is composed of the support load contributed by the secondary seal (elastomeric O-ring), the hydraulic force acting on the rotor back side, and the air pressure in the rotor chamber. The test seal has a geometrical balance ratio of 0.5, and is balanced at a clearance of 6.5 μm, with a stator coning of 1.6 mrad, and at an air pressure of 0.018 MPa. Various seal clearances can be obtained by varying the air pressure acting on the rotor.

Control System Block Diagram

The block diagram of the seal clearance control system is shown in Fig. 2. An eddy current proximity probe mounted at the end of the housing was used to measure the dynamic response of the rotor along the shaft axis (Fig. 1). The proximity probe has a bandwidth of about 10 kHz. It can measure the static and dynamic distance between its tip and the rotor end surface. The smallest dynamic displacement it can measure is 0.1 μm. The clearance of the seal is obtained as follows: when the motor is stationary, high air pressure is applied in the rotor chamber to ensure that the rotor is pressed against the stator, at which a probe reading is taken.

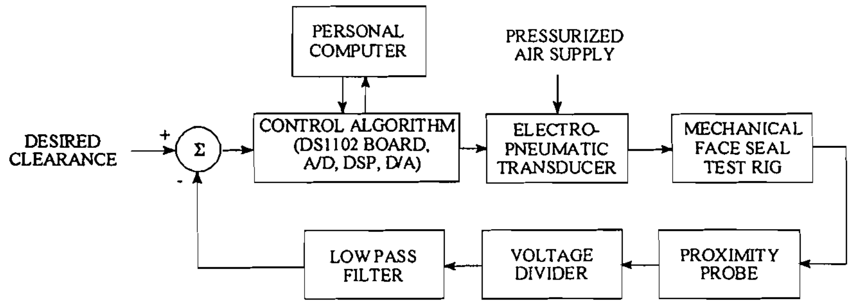


Fig. 2—Block diagram of the seal clearance control system.

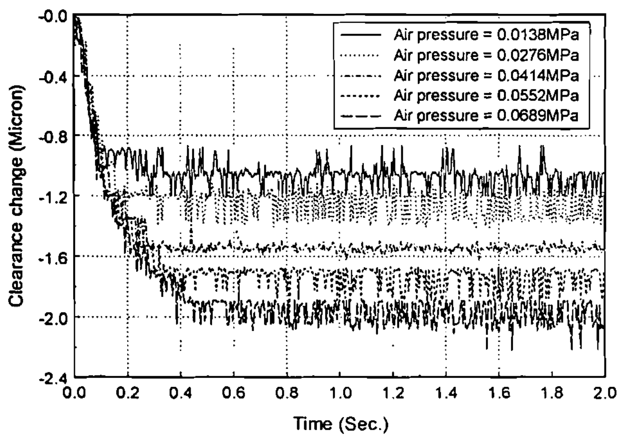


Fig. 3—Step responses of the test seal for various air pressure step inputs.

This reading represents the reference for zero clearance. The clearance of the seal at any time is the difference between the instantaneous reading and the zero reference. The output range of the proximity probe is from zero to negative 22 volts.

In order to interface with the DS1102 board, which can only take signals from -10 volts to +10 volts, a voltage divider is used to drop the maximum amplified voltage of the proximity probes outputs from -22 volts to -8.4 volts. The reduced voltage is then sent into an antialiasing low pass filter with a cut-off frequency of 1000 Hz. After filtering, the signal is sent to the DS1102 board that resides in a personal computer. It has onboard a floating-point digital signal processor (DSP), an analog to digital (A/D) converter and a digital to analog (D/A) converter. The proximity probe signal is obtained through the A/D converter of the board. The seal clearance is then calculated and compared with the desired seal clearance. Based on the error signal between the measured clearance and the desired clearance a control signal is calculated according to a PI control algorithm. The controlled signal is then sent to the electropneumatic transducer through the D/A converter. The electropneumatic transducer provides air pressure that is proportional to the voltage signal obtained from the D/A converter.

CONTROLLER DESIGN

Seal Axial Dynamic Model

The dynamic model of the FMR noncontacting mechanical face seal has been established by Green *(2)*. The rotor has three degrees of freedom: one axial translation along the shaft axis and two angular rotations about its two inertial axes in a plane that is perpendicular to the shaft axis. Because the axial mode is theoretically shown to be decoupled from the angular mode *(2)*, in this work, only the axial motion of the rotor is considered.

The entire seal system includes the seal itself, the electropneumatic transducer, the eddy current proximity probe, the voltage divider and the low pass filter. Instead of relying upon a composed theoretical model, system identification is being performed to provide the actual dynamic model of motion in the axial mode. The experimental model is obtained from the response of the seal clearance to the voltage applied to the electropneumatic transducer. Experiments are conducted at 0.207 MPa sealed water pressure, zero shaft rotating speed and 1.6 mrad stator coning angle.

Since the air pressure output of the electropneumatic transducer is proportional to the voltage applied to it, the applied voltage was used to represent the control effort throughout this paper. However, air pressure will be shown in the figures that follow to represent the physical meaning of the control effort. An equilibrium voltage 1.3 V, i.e., air pressure 0.018 MPa, is first chosen. Then several different step changes in voltage 1V, 2V, 3V, 4V, and 5V (corresponding to pressure steps of 0.0138 MPa, 0.0276 MPa, 0.0414 MPa, 0.0552 MPa, and 0.0689 MPa) are applied to the seal system from equilibrium.

The step responses of the seal clearance, shown in Fig. 3, indicate that a larger input step size gives a smaller gain and larger time constant. Therefore, it is concluded that the seal system is nonlinear, as found also in *(7)*. However, the air pressure steps applied are very large compared with the pressure variations needed later in the control scheme. Hence, even if nonlinearity exists in the seal system, it is possible to use linear approximation when small motions occur about an equilibrium point *(2)*. Considering this, for system identification, a step of 1V (0.0138 MPa air pressure) is applied from the equilibrium voltage 1.3V (0.018 MPa air pressure). The measured step response is shown in Fig. 4 along with a first order system approximation.

Theoretical seal axial dynamic model calculated according to *(2)* can also be approximated by a first-order system. The gain of

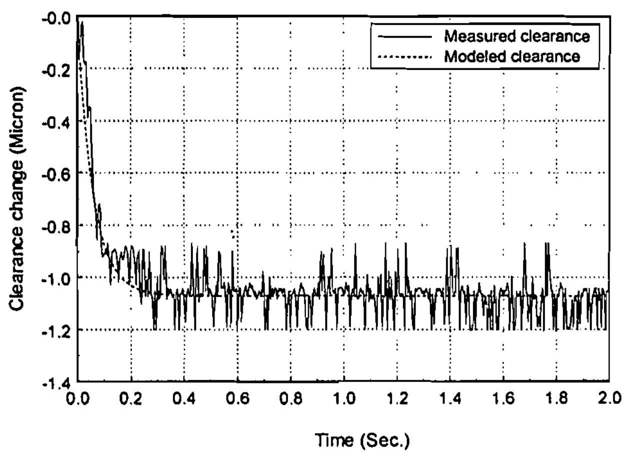


Fig. 4—Step responses of the test seal for air pressure step input of 0.0138 MPa.

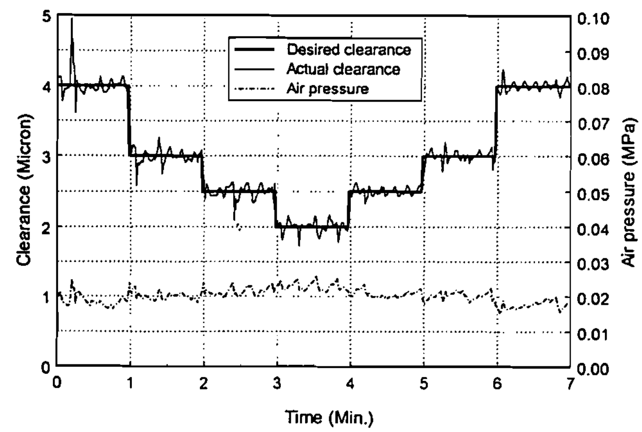


Fig. 5—Set-point change results for small clearance steps and range.

the system is obtained by dividing the maximum change of the seal clearance by the step input voltage. The time constant is the time it takes for the seal clearance to reach 63 percent of its maximum value. Therefore, a first order seal axial dynamic model is determined from Fig. 4 to be:

$$G_p(S) = \frac{-1.07x10^{-6}}{1+0.06S}$$

The step response of this model is also plotted in Fig. 4. It matches the measured step response very well. This model was used to design the seal clearance controller.

Controller Design

The control objective for this research is to make the seal follow desired clearance set-point changes and to make it immune to operation disturbance, such as variations in shaft speed and sealed fluid pressure during certain set-point operation. A proportional-integral (PI) controller is designed to accomplish this.

Two issues need to be considered when choosing the design specifications for the controlled closed-loop system. First, a large overshoot should be avoided because large overshoot means a greater possibility of face contact. Second, the closed-loop system should have small oscillations because large oscillations may cause the fluid film between the seal faces to collapse and cause seal failure. Considering this, the closed loop system should have large damping and a small natural frequency. The damping ratio is chosen to be 13.36 and frequency is chosen to be 0.96 rad/sec. The corresponding PI controller is determined to be:

$$G_c(S) = 5x10^5\left(1+\frac{1}{S}\right)$$

A practical issue needs to be tackled in implementing the PI controller. The present actuator (the electropneumatic transducer) has certain limitations where it can only output air pressure between zero and 0.138 MPa. When the control system operates over a wide range of operating conditions, it may happen that the required control variable reaches the actuator limits. It is commonly known that integral windup will happen when integration is used in the regulator when the actuator saturates. The consequence is that the controlled system may have large overshoot when the actuator saturates. Therefore, a controller with anti-windup was designed and implemented to avoid integral windup.

RESULTS AND DISCUSSION

Several experiments were conducted using the seal test rig to test the performance of the clearance controller. All experiments were conducted at nominal operation condition of 0.207 MPa sealed water pressure, 15 Hz shaft rotating speed, and 1.6 mrad stator coning angle.

The ability of the controlled system to follow the set-point changes in seal clearance was tested first. The results are shown in Figs. 5 to 7. The desired seal clearance (set-point changes), the actual measured seal clearance, and air pressure required to maintain the set-point seal clearance are plotted. Figure 5 shows that the controlled seal accurately followed small changes in clearance set-point, with steps of 0.5 μm ~ 1 μm and range of 2 μm. The big spike in the beginning of the plot indicates that the operating clearance is far away from the desired set-point just before the controller is applied to the seal system.

Figure 6 shows results for larger clearance set-point changes in step (1 μm ~ 3 μm) and range (6 μm). The controlled system followed the set-point changes very well. Figure 7 shows results for even larger clearance set-point changes in range (8 μm). Again, good controlled system performance was obtained as in Fig. 5. The large oscillations of the actual clearance at the beginning of the plot are because of the low fluid film stiffness and damping due to large seal clearance (10 μm) at the time the controller is applied to the seal. While the clearance responds very quickly and accurately to set-point changes, the control effort as indicted by the air pressure in Figs. 5 to 7 is very small (less than 0.035 MPa).

Experiments were also conducted in an eight-minute duration to check whether the controlled system can reject process disturbances during operation. In Fig. 8, the desired seal clearance was set to 4 μm. The following disturbances are introduced at one-minute intervals: after running the seal at a nominal speed of 15 Hz for one minute, the speed was increased to 20 Hz, then reduced to 15 Hz, and to 10 Hz, after which it was increased back to 15 Hz where it was kept constant. At that instant the sealed water pressure was increased from the nominal value of 0.207 MPa to 0.241

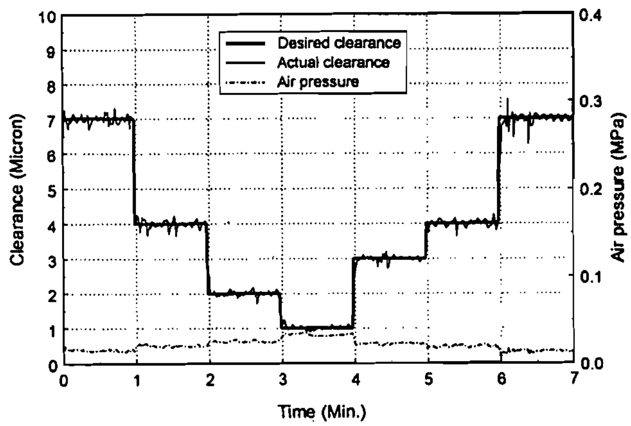


Fig. 6—Set-point change results for larger clearance steps and range.

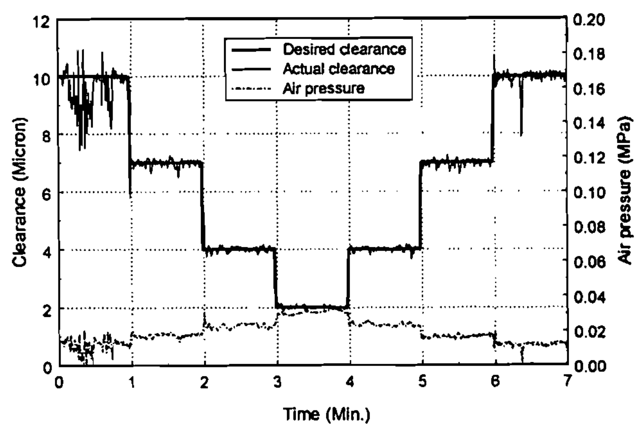


Fig. 7—Set-point change results for larger clearance steps and largest range.

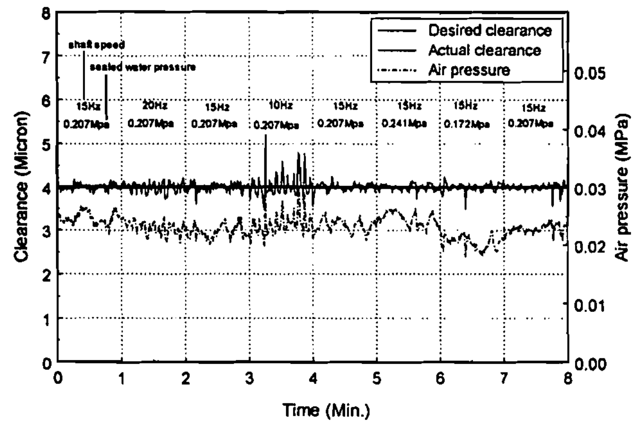


Fig. 8—Transient results without set-point change.

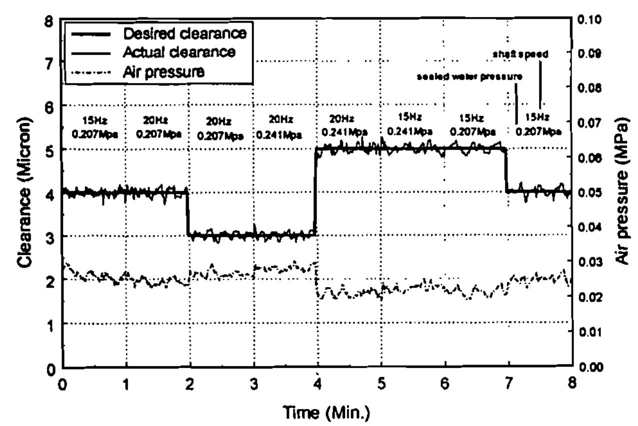


Fig. 9—Transient results with set-point changes.

MPa, then decreased to 0.172 MPa, after which it was increased back to 0.207 MPa.

As shown in Fig. 8, the controlled system remained at the desired (single set-point) clearance without being disturbed by the changes in shaft speeds and sealed water pressures. The larger oscillation when the shaft speed was reduced to 10 Hz is because of lower gyroscopic and hydrodynamic effects at lower speeds which tend to align the rotor and maintain stable operation of the seal. In Fig. 9, clearance set-point changes are introduced along with other disturbances. The following disturbances were introduced at one-minute intervals: after running the seal at 4 μm for one minute, the shaft speed was increased to 20 Hz, then the set-point clearance was reduced to 3 μm, after which the sealed water pressure was increased to 0.241 MPa, followed by an increase of the set-point clearance to 5 μm, reduced shaft speed to 15 Hz, then the sealed water pressure was reduced to 0.207 MPa, and finally the set-point clearance was decreased to 4 μm. It is shown in Fig. 9 that the controlled system can follow set-point changes in the presence of shaft speed and seal water pressure disturbances. Leakage was not measured for the experiments in this research, however, it was observed that when clearance was large, leakage was large, and vice versa.

CONCLUSIONS

Clearance control of a mechanical face seal is achieved by using a PI controller with anti-windup in a FMR noncontacting mechanical seal test rig. The controlled seal can follow set-point changes of seal clearance spontaneously and can operate at a pre-selected seal clearance without being affected by disturbances in shaft speed or sealed water pressure. The controller is designed solely based on an experimentally determined seal model. A theoretical model of the seal system is not required. Instead of using thermocouples to measure temperatures in the seal system and assuming qualitatively that temperature has correlation with seal clearance *(5)-(8)*, an eddy current proximity probe is used to directly measure the seal clearance. The controller is capable of controlling the seal clearance at a wide range of values, while keeping the control effort (air pressure required) fairly small.

Although the clearance controller proved to be effective in maintaining seal clearance set-points, it is only a first step in health maintenance. When large relative misalignments exist between the rotor and the stator, controlling just the seal clearance only may be insufficient to ensure noncontacting operation. A controller that can reduce (and ideally eliminate) the relative misalignment is necessary.

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