An Experimental Study on the Self-recovery Regulation System of Centrifugal Compressor Axial Displacement Fault

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Abstract: The axial displacement fault has very little indication before taking place and the fracture of oil film between thrust disk and tilt pads mainly contributes to this breakdown. Usually this fracture was caused by the contamination of oil or decrease of oil supply pressure which result in losing load bearing capacity of oil film. Also the increase of axial force caused by fluctuation of operation condition can result in the oil film fracture. Both of the above reasons can lead to tilt pads abrasion and emergency shutdown of machine. The method presented in this paper monitors the stiffness of oil film and identifies the reason of axial displacement increasing. By introducing the mechanism of axial displacement fault self-recovering and taking the axial displacement as the control objectives, the axial forces on rotor was controlled on-line and the minimum oil film thickness was ensured in real time. The experiment indicated that the mechanism can realize the axial fault self-recovery effectively, and with this system, the centrifugal compressor doesn't need to be shut down immediately when the axial displacement fault happens.

Keywords: centrifugal compressor, axial displacement, thrust bearing, fault self-recovery, PID controller, axial stiffness

1 Introduction

Large-scale centrifugal compressor is the central equipment of process industry such as petrochemical industry and metallurgical industry etc. It is closely related to the production process and contributed to be a huge system. The fault of this type of system may cause casualty and brings great pecuniary loss. International engineers and scientists have developed monitoring equipment and fault diagnostic technique from 1960's. Predictive maintenance and intelligent maintenance have been promoted into industrial enterprises, and the Emergency Shutdown (ESD) have been adopted extensively as well. The diagnostic technique has played an important role in safe production and obtained significant economical benefit.

However, the solution of equipment faults depends on ESD or disassembling maintenance after manual shutdown. This decreases the production time, increases the maintenance cost, and causes new faults or even accidents during startup and shutdown process of machines. The Fault Self-recovering(FSR) technique is the effective way to improve the reliability of equipments, maintain long period operation and reduce the production cost.[1]

This paper studies the essential reasons for the typical high-pressure centrifugal compressor faults caused by over standard of axial displacement in domestic factories recent years; and then establishes the fault diagnosis technique of axial displacement which makes oil-film stiffness of bearing as the parameter; based on these studies, establishing an on-line monitoring, diagnosis and self-recovering regulation system of axial displacement is highly significant to ensure the long period running of machines without faults; finally, an experimental study on the axial displacement fault self-recovering system was conducted. The results shows that the axial displacement fault of centrifugal compressor can be online elimination by this system.

2 Diagnose the axial displacement fault

2.1 Summary: The axial displacement fault

Axial displacement is an important shutdown indicator of centrifugal compressor ESD, and the major factor of why centrifugal compressor can't be operated for long period is because of axial displacement fault, especially the high-pressure CO_2 compressor. This fault doesn't have any obvious symptom before the axial displacement suddenly increases, with the tilt pads worn-out heavily after shutdown. There are two factors causing this fault, one is the difficulty to decide the axial force of rotor during design. In engineering design, The fall of pressure caused by air flow in clearance between the bladed disk and bladed lid was ignored when calculating the axial force, which leads to a big deviation between the calculated axial displace value and the actual, even the coefficient of safety is 1.7-2 in design, it can't guarantee the axial force not exceeding the carrying capacity of bearings. On the other hand, the fall of pressure can be calculated with the development of fluid mechanics computing technology, but the fall of pressure varies with the change of sealing leakage when compressor is running, at the same time, the conditions of compressor is fluctuant, and the fluctuation can't be predicted, so the residual axial force may exceed the carrying capacity of bearings and the bush-burning takes place[2]. A thrust bearing and tilt pads with high load perennial is presented in Fig.1. It is clearly shown that there is a lot of carbon deposition on the surface of the tilt pads. Because the axial displacement is beyond standard, the reliability of compressor is already too low, therefore even if the fault doesn't happen, the interlock shutdown of centrifugal compressor caused by axial displacement will happen in most cases once the condition changes.

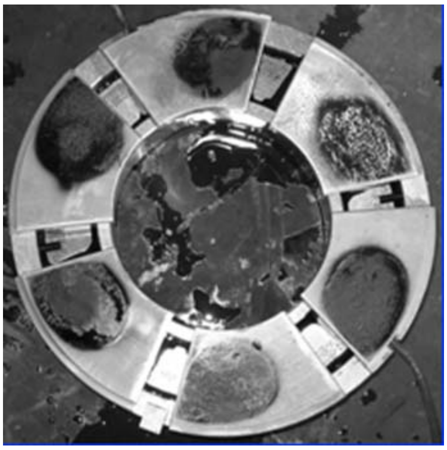
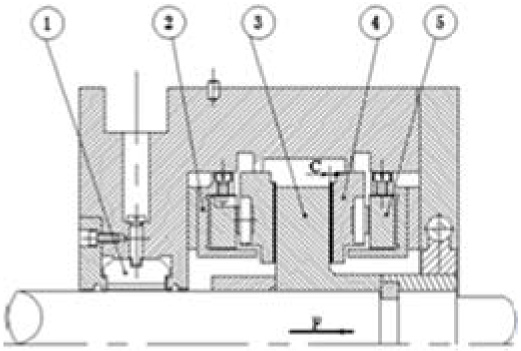


Fig.1 the thrust bearing pad with coking

The CO_2 compressor of urea plant installed in a branch of Sinopec is a high-pressure centrifugal compressor made by Shenyang Blower Factory, consisted of a 2MCL607 low pressure cylinder and a 2BCL306A high pressure cylinder. The maintenance about the thrust-bearing pads of the high pressure cylinder has never stopped from commissioning in 1984, Especially in the later half of 2002, with a number of faults happening continually because of the changing of operational conditions. For example, the axial displacement of this high pressure cylinder up to -395 μm suddenly from -230 μm during the adjustment of the operating conditions at 8:00 on the August 28, 2002, which could severely impair the safe running of compressor. It was found that the main thrust bearing pads was over-burnt and should be replaced during the maintenance on September 16[3]. The similar faults happened on another CO_2 compressor of urea plant in a branch of China National Petroleum Corporation. The interlock shutdown fault caused by axial displacement occurred many times since production commencement in 1988, especially in July of 2002, this kind of faults took place five times in one month.

2.2 The mechanical property of bearing

The axial displacement fault usually refers to the alarming or emergency shutdown caused by the axial displacement beyond standard level. The axial displacement is mostly consisted of bearing clearance, attenuation of the oil-film thickness, elastic deflection of bearing support, errors of measurement system, wear of the tilt pads surface and other factors[4]. The essential reasons of axial displacement fault are various as there are many factors bringing about the axial displacement . And the axial displacement caused by changing of the oil-film thickness have the most important effect on the bearing safety Other factors such as the temperature of lubricating oil, frequency of the rotor and the load of bearing varying the axial displacement also have certain impact[5].



1-journal bearing, 2-base ring, 3-thrust disc, 4-thrust bearing bush, 5-level block

Fig.2 the structure of thrust bearing

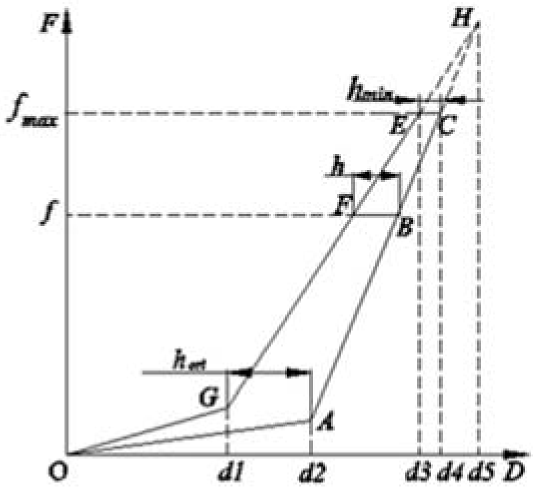


Fig.3 load-axial displacement of thrust bearing

Fig.2 presents the basic strcture of tilting pad thrust bearing. Look, there is a clearance C between the thrust disc (3) and the thrust bearing bush (4) when rotor stops. The clearance C will be filled with oil-film by the hydrodynamic slurry pressure when rotor runs, and the action of axial force will cause elastic deflection on the bearing. The stiffness of thrust bearing depends on the stiffness of oil-film, thrust bearing bush base ring and level block. The axial displacement of rotor is showed in fig.3. The vertical axis presents axial force and the horizontal axis presents axial displacement.

After the rotor is installed, The bearing should be adjusted at “zero” point of axial displacement, namely, the thrust disk at the middle of thrust bearing. And the axial force is zero too, right at the ‘O’ point of fig.3. The axial displacement arrives at ‘A’ point easily with a little axial force because of the existence of clearance C when rotor stops. The value of axial displacement here marked as d_2, the axial displacement arrives at ‘C’ point along the ‘AC’ line with the increment of axial force, and marked as d_4, the non-linear factors of the bearing structure were ignored here. A oil-film forms between the thrust disc and the thrust bearing bush because of the fluid’s viscidity when rotor is running. The axial displacement arrives at ‘G’ point with the affection of a little axial force, and marked as d_1. The axial displacement will arrive at ‘E’ point with the increment of axial force along the ‘GE’ line, axial displacement maked as d_3 here. The slope of ‘AC’ line just lie on the stifness of bearing structure, and the slope of ‘GE’ line depends on the stifness of oil-film and beraring structure in series. These two lines will cross at ‘H’ point with the difference of slopes if there is no control, and marked as d_5.

The thickness of oil-film h is the horizontal distance of ‘F’ point and ‘B’ point with the axial force f, and the distance between ‘G’ and ‘A’ is the original oil-film thickness, marked as h_{ori}. The minimum oil-film thickness will be thicker than h_{min} to ensure the bearing safety, and the axial load here marked as f_{max}. The alarm point and emergency shutdown point should be set at d_3 and d_4 separately when rotor is running.

2.3 Diagnosis of axial displacement fault

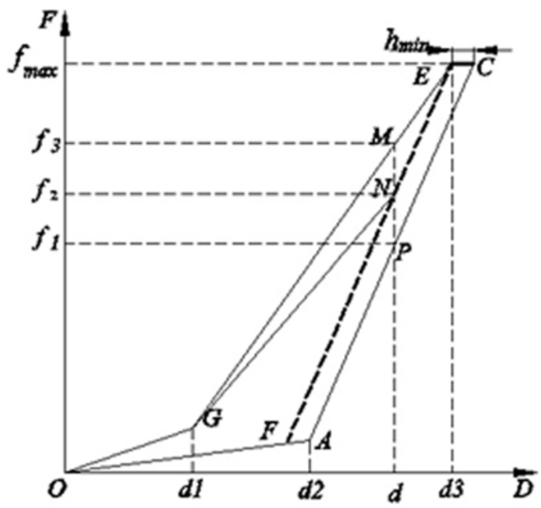


Fig.4 diagnosis theory diagram of axial displacement fault

According to the reasons leading to emergency shutdown by axial displacement beyond standard, we can divide the above into the axial force factors and not axial force factors in engineer practice. As it's difficult to measure the axial force of compressor rotor on line, it's not easy to diagnose the reasons leading to the axial displacement increment. For example, the axial displacement can increment suddenly without symptoms due to the increment of axial force or inferior quality of lubricating oil[6], all of these will lead to emergency shutdown finally. The diagnosis way of bearing fault by monitor the axial stiffness of bearing is introduced and presented in fig.4.

Given a bearing, we measure the bearing clearance and plot line OA. The load-displacement curve AC can be plotted by the ways of finite element analysis or add static load by materials universal test machine. To ensure the safety of bearing, the minimum thickness oil-film should be determined. If we move line AC h_{min} to left and get line FE, we will get the line of safeguard of bearing. To the operating machine, we only measure the axial displacement, and there may be three different axial force corresponding with the measurement value of axial displacement. Fig.4 shows three typical axial force f_1, f_2 and f_3. If the axial force is f_1, the thickness of oil-film will be zero, and the bearing pad will be damaged seriously. If the axial force is f_2, here the thickness of oil-film is h_{min}, we should do something to avoid the oil-film thickness being thin. And if the axial force is f_3, the bearing is safe.

The real-time oil-film stiffness is defined as K_i, and the oil-film stiffness of f_1, f_2 and f_3 are defined as K_{FP}, K_{FN} and K_{FM}, It will alarm when $K_i \leq K_{FN}$, and check up the quality of lubricating oil. The machine will interlock shutdown if $K_i \leq K_{FP}$.

2.4 Determine the axial stiffness of thrust bearing

The axial stiffness of thrust bearing can be defined as the axial force that is needed to move unit axial displacement. It's related with the structure of bearing, viscosity of oil-film and the speed of rotor. The relationship can be defined as:

$$F = f(u,r)D \qquad (1)$$

u is the viscosity of oil-film and r is rotor revolving speed. The axial stiffness f is presented as:

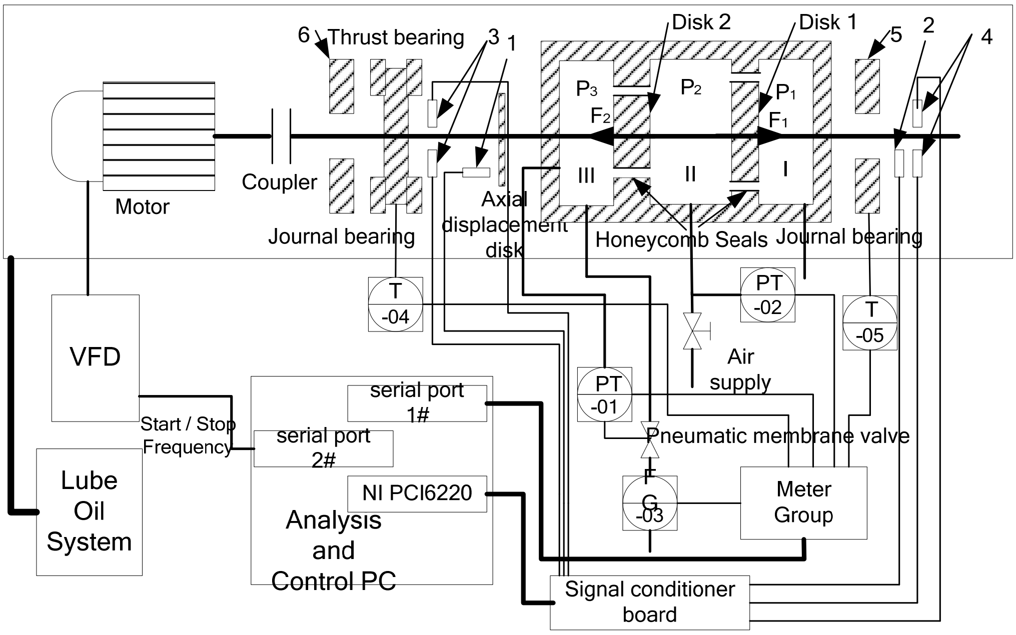
$$f(u,r) = \frac{dF}{dD} \qquad (2)$$

Therefore if we measure the axial displacement, given a little disturbance to axial force when the compressor is running, then the axial stiffness of thrust bearing can be calculated. On a real compressor, we can change the pressure of balance dick low pressure side by adjusting the control valve, and record the change of pressure as ΔP, so the change of axial force ΔF is:

$$\Delta F = \Delta P \times A \qquad (3)$$

Where A is the area of balance disk.

Referring to the fig.4, by Measuring the stiffness of thrust bearing on line and combining with the axial displacement, the health condition of compressor bearing can be identified. Based on this, with the ways stated in ref.[7], the axial displacement fault can be eliminated online and axial displacement fault self-recovering will be achieved accordingly.



PT-Pressure transmitter, FG-Flow transmitter, T-Temperature, 1-Axial displacement sensor, 2-Key sensor, 3&4-Vibration sensor

Fig.5 axial displacement fault self-recovering system

3 design of the experimental device

3.1 Introduction of the axial displacement fault experiment device

To verify the validity of the axial displacement fault self-recovering system, we design a simulate experiment device in Diagnosis and Self-recovering Research Center of Beijing university of Chemical Technology. The structure of experiment device is presented in fig.5.

The motor is connected with the rotor via the coupler, and the speed of motor can be adjusted by VFD which is controlled by the computer. The bearing (5) of compressor is five-pad tilting pads journal bearing, and the bearing (6) is five-pad tilting pads thrust journal bearing. The temperature sensors of bearing and load sensors of tilt pads are embedded in it. The sensors (3,4) are used to measure the vibration of rotor, sensor (1) is used to measure the axial displacement, and the data of sensor (2) is used to calculate the speed of rotor. The disk 1 and disk 2 simulate the impeller and balance disk of turbo machine respectively, which divide the cavity into three parts, the pressure of the three parts are P_1, P_2 and P_3 respectively. There is an axial force F_2 because of the pressure difference between the two sides of disk 2. F_2 equals to the balance force generated by balance disk of a real compressor. Similarly, F_1, generated by the pressure difference between the two sides of disk 1, equals to the thrust force of real machine. The combination of F_1 and F_2 is the axial force of the machine, supported by the thrust bearing.

3.2 the mechanical property of experimental device

The area of disk 1 and disk 2 are defined S_1 and S_2 respectively, hence, F_1 and F_2 can be calculated as:

$$\begin{aligned} F_1 &= (P_2 - P_1)\times S_1 \\ F_2 &= (P_2 - P_3)\times S_2 \end{aligned} \qquad (4)$$

So the total axial force of the experimental device can be calculated by equations of (4):

$$F = F_1 - F_2 \qquad (5)$$

The designed diameters of disk 1, disk 2 and the axis are 210mm, 150mm and 85mm respectively, so $S_1 = 0.029m^2$, $S_2 = 0.012m^2$.

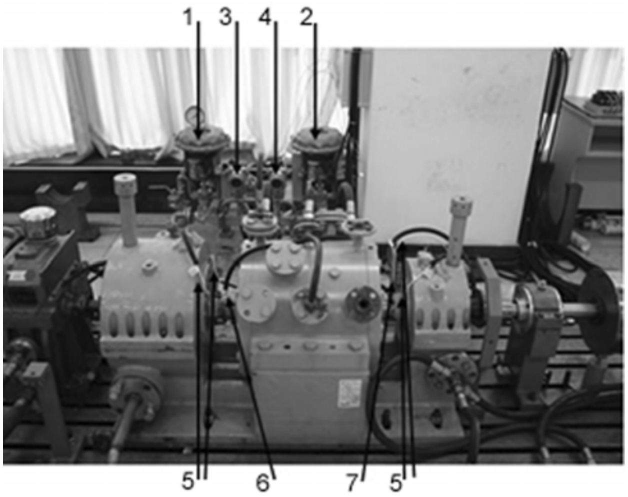
The measured value of P_1 is 0.003MPa in experimental study, therefore:

$$\begin{aligned} F &= (P_2 - 0.003)\times 0.029 - (P_2 - P_3)\times 0.012 \\ &= (0.017P2 + 0.012P_3 - 0.000087)\times 10^6\, N \end{aligned} \quad (6)$$

The axial force F is related to P_2 and P_3 seen from equation (6). If keep the pressure of air supply, the axial force can be controlled by changing P_3, the direction of axial force is correspond with F_1 which presented in fig.5. And control the pressure of P_3, as well as control the axial force F, just like keep the axial displacement not over standard.

3.3 The implementation of axial displacement fault self-recovering system

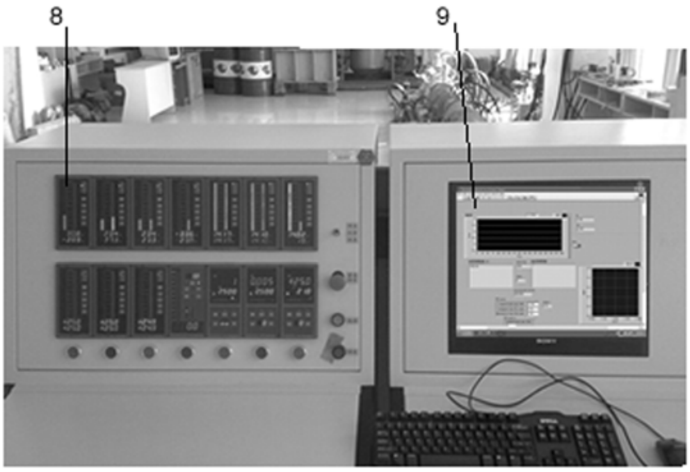
The system structure is presented in fig.5. The system collects, processes and analyze the real-time data of temperature of journal bearing and tilt pads, the axial displacement, the vibration of rotor, and the pressure of P_2 and P_3, and other data and transfer the control signal via a second meter to pressure adjusting valve. After this, the axial displacement is solved through self-recovering adjustment.



1&2- Pneumatic membrane valve, 3&4-Pressure transmitter, 5-Vibration sensor, 6-Axial displacement sensor, 7-Key sensor

Fig.6(a) the system experimental device

Fig.6(a) and 6(b) presents the axial displacement fault self-recovering system experimental device. The air comes out from the air supply and goes into the cavity II through a valve, and leaks to the cavity I and III through seals. The pressure cavity II and III can be measured by pressure transmitter (4, 3-Rosemount 3051), and the data are read by secondary meters. The pressure of cavity I and III can be controlled by pneumatic membrane valve (2,1-Arca 101-P1), and the system can control the opening of these valves by XSTS single return circuit PID adjusting meter. The meters are made by Beijing Kunlun TianChen Instrument Science Technology Company. The system is completed in Labview7.1. It reads the pressure data from secondary meters by serial port communication, and reads the data of axial displacement and vibration by NI PCI6220 data collection board, and control signal output to the valve by secondary meters too.



8-Secondary meters, 9-Self-recovering regulation system

Fig.6(b) the control platform device of the system

4 Experimental study

4.1 Determine the zero point of axial displacement

The sensitivity of used eddy current sensor is $8\, mv/\mu m$, and output signal is -24V to 0V. Turn on the valve of cavity I, then the pressure of cavity I is close to the zero. At the same time, turn off the valve of cavity III, so the pressure here we get will equal to the cavity II's pressure. Hence, $F_2=0$, $F=F_1$. Then change F=0 little by little by adjust the air supply, and the axial displacement sensor data is -8.07V when F=0. Then turn off the valve of cavity I, turn on the valve of cavity III, keep $F_1=0$, $F=F_2$. Adjust the air supply to make F=0 gradually, and the data of sensor is -10.99V, so the middle point of balance disk is $\frac{-10.99+8.07}{2}-8.07=-9.53V$, this point is the zero point of axial displacement, the sensor data of other measure points is defined as D_i, and corresponding axial displacement is defined as d_i:

$$d_i = \frac{D_i + 9.53}{8} \times 1000\mu m \qquad (7)$$

4.2 Study the mechanical property of experimental device journal bearing

The experimental procedure of analyzing the mechanical property of the experimental device as follows:

(1) Turn on the valve of cavity I, make the pressure of this cavity close to the atmospheric pressure. Turn off the valve of cavity III, make the pressure of this cavity close to the pressure of cavity II. The air supply is a screw compressor. Turn on the valve when the pressure of air supply arrives at 1MPa, then stop the screw compressor. Record the pressure data of cavity II and III and axial displacement data, plot the 'load-displacement' curve.

(2) Star up the motor and adjust the revolving speed of rotor to 8000r/min, and do the procedure (1) again, plot the curve of this speed.

(3) Adjust the speed to 7000r/min, 6000r/min, 5000r/min, 4000r/min, 3000r/min and 2000r/min respectively, and do the procedure (2) and plot the curve of different speed.

4.3 Identify the stiffness of bearing

The procedure of identifying the stiffness of bearing as follows:

(1) Star up the motor and adjust the revolving speed of rotor to 8000r/min, turn on the screw compressor and keep the pressure of cavity II stable. Turn on the valve of cavity I, adjust the opening of cavity III's valve to 20% and stay there, so the axial displacement and axial force will be stable after a few minutes.

(2) Record the temperature of lube oil, and adjust the opening of cavity III's valve to 30% and keep three seconds then adjust to 20% again. In the experimental process, record the data of pressure of cavity II, III and corresponding axial displacement. The stiffness of bearing can be calculated by equation (2).

(3) The temperature of lube oil will rise when the machine is running, so the stiffness of different temperature at 8000r/min can be achieved by repeat the step (2).

(4) Adjust the rotor revolving speed to 7000r/min, 6000r/min, 5000r/min, 4000r/min, 3000r/min and 2000r/min respectively, repeat step (2) and (3), and study the bearing stiffness of different speed of rotor revolving.

4.4 Axial displacement fault self-recovering control

The experiment aims to achieve the axial displacement fault self-recovering, and verify the real-time capability and stability of the fault self-recovering system. The process as follows:

(1) Star up the motor and adjust the revolving speed of rotor to 8000r/min, and turn on the screw compressor at the same time. Adjust the valve opening of cavity II to 10%.

(2) The safe set-point of axial displacement is 235μm. Run the fault self-recovering system, the real-time data of pressure of cavity II, III, the axial displacement and the valve opening of cavity III can be recorded within this system. Turn on the valve of cavity II completely within a short time manually after the pressure of air supply goes up to 1MPa. And the axial force will increase suddenly due to the

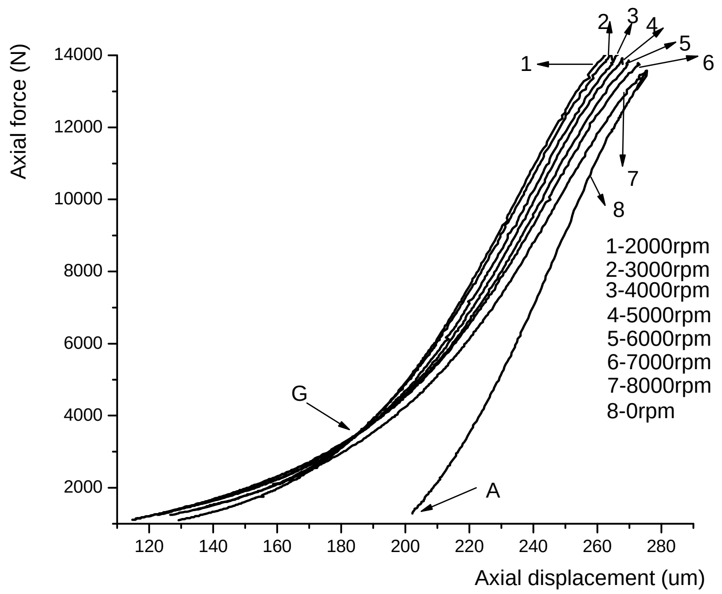


Fig.8. the experimental device's load-axial displacement curves of different rotor revolving speed

sudden increase of the cavity II's pressure. The axial displacement increases suddenly, and the control results shall be recorded.

(3) Select other controllers of the system, and repeat (2) again.

5 The results of experiment

5.1 The 'load-axial displacement' curve of bearing

The 'load-axial displacement' curve is plotted by the procedure of 4.2 and presented in fig.8. The conclusions as follows:

(1) The 'load-axial displacement' curve is similar to the curve of fig.3, the bearing clearance is 200 μm according to the position of 'A' point. The axial displacement is increased with the increase of load, and the axial displacement is up to 276 μm when the load is 13800N.

(2) Comparing the curves we will find that the oil-film thickness of low speed is thinner than the one at high speed when the load is at low lever. The oil-film thickness will be thinner and thinner with the increase of load and speed. The changing of oil-film thickness made these curves has a clear intersection point 'G', to the point, we can understand that the axial displacement is mainly controlled by stiffness of oil-film before the 'G' point, and after this, the axial displacement is mainly controlled by axial stiffness of tilt pads journal bearing. The axial stiffness of tilt pads journal bearing will turn lower with the rotor revolving speed getting higher. The more power will be achieved in unit time at a higher rotor revolving speed, so the temperature of lube oil will rise fast and make the viscosity of lube oil smaller.

(3) The no-linear phenomenon is clear in these curves, namely, the stiffness of thrust bearing varies with the change of load. But after the 'G' point, the curves are close to linear, the stiffness changes to stable, the slopes of these curves turn close to the slope of 'load-axial displacement' curve where the rotor stopped.

(4) The near polynomial can be achieved by the method of least squares, as listed in table 1. These polynomials of normal operation conditions provide reference for diagnosis the fault., and determined the axial displacement fault is caused by exceeding axial force or not.

Table 1, The polynomial fit of axial force of axial displacement at different revolving speed

revolving speed（r/min）	polynomials fit
2000	$F = 0.73083d^2 - 185.0143d + 12825.12303$
3000	$F = 0.73014d^2 - 188.4900d + 13503.80559$
4000	$F = 0.70117d^2 - 181.7717d + 13188.48948$
5000	$F = 0.66574d^2 - 171.4285d + 12472.14904$
6000	$F = 0.65224d^2 - 170.3231d + 12687.15597$
7000	$F = 0.62964d^2 - 164.8987d + 12434.87057$
8000	$F = 0.62251d^2 - 165.8128d + 12598.76896$

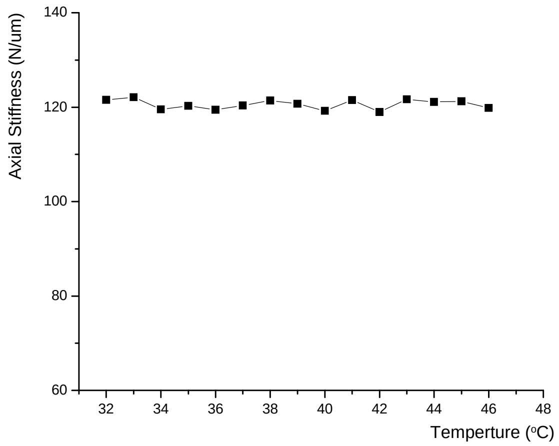


Fig.9(a) the varying curve of axial stiffness with the temperature

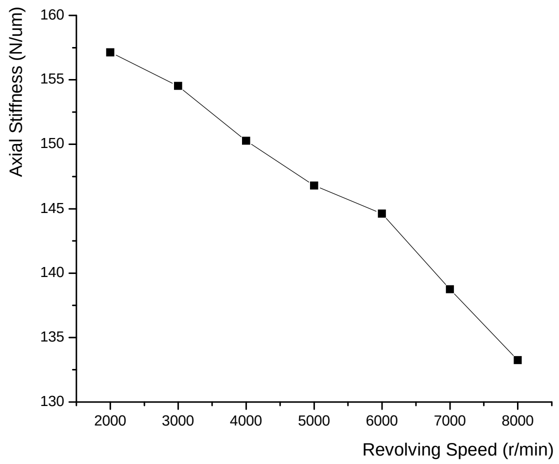


Fig.9(b) The varying curve of axial stiffness with revolving speed

5.2 Identify the stiffness of bearing

A axial force disturbance made with purpose is added the running rotor in order to measure the axial displacement, and obtain the stiffness of bearing in this way. The reason for the fault can be determined by comparing the current stiffness with the regular one. Fig.9(a) is the axial stiffness of different temperatures, plotted at the revolving speed of 8000r/min and the axial force is 7242N by the ways of 4.3. We can see that the axial stiffness changes little by little with the raising of the temperature. Fig.9(b) presents the change of stiffness with the rotor's revolving speed, keeping the temperature same. It's clear the stiffness is lower when revolving speed gets higher.

4.3 Results of several different controllers of the axial displacement fault self-recovering system

4.3.1 Result of Boolean controller

Determine the safe threshold of axial displacement according to the design of compressor, the max value means the self-recovering system should start working when axial displacement goes over this value , and means if the measured axial displacement is , the conditions of machine is safe. And the Boolean controller will output 100% opening signal to the controlled valve if and turn off the valve if . The controlled axial displacement curve is presented in fig.10(a). It is clear that the axial displacement is axial float, and this condition is a new incertitude factor to the production plant. Hence Boolean controller isn't reliable to this fault although it's responsive.

4.3.2 The results of PID controller and fuzzy PID controller

The PID controller is one of the most widely used controllers in process control of industries. The PID curve of fig.10(b) is the result of the controller. The PID controller will only have better result in single operation condition. If the operation condition changes, the controller will lose its advantage. The short response time and trim control can't be achieved at same time.

The fuzzy controller is adopted, its curve is presented in fig.10(b). This controller can conduct the varying operations. When the error is big, the controller can make the variable reach set-point in a short time and then adjust its output to get trim control[9][10]. The control result is better than PID controller and can be shown in fig.10(b).

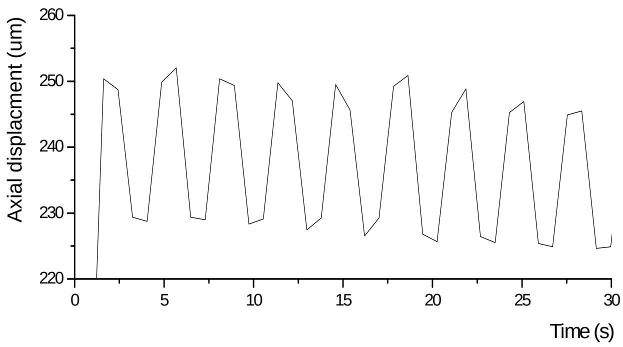


Fig.10(a) The controlled axial displacement curve

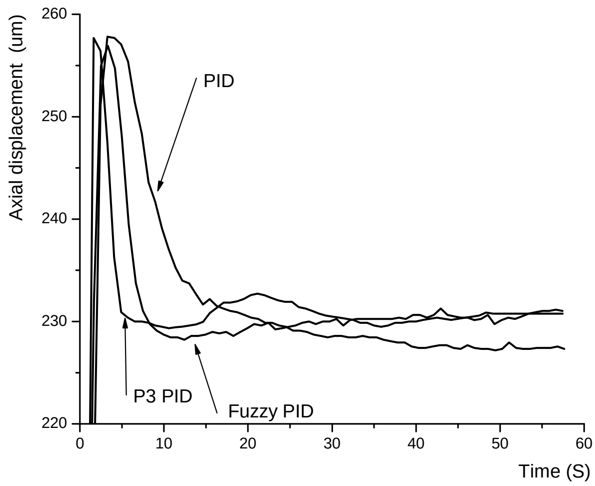


Fig.10(b) The axial displacement curves of different controllers

5.3.3 Make the pressure of cavity III as control variable

The former controllers control variable is axial displacement, and the axial force is related with the cavity II and III's pressure according to the equation (7). The experimental result shows that its near linear between the axial force and axial displacement when axial force is bigger than 4000N. Take 8000r/min as an example:

$$d_i = \frac{F}{120.404} = \frac{(0.017P_2 + 0.012P_3 - 0.000087)\times 10^6}{120.404}\ \mu m \quad (8)$$

Here 120.404 is axial stiffness of bearing, P_2 can be measured, so the threshold value of P_3 with corresponding P_2 can be calculated by:

$$P_{3sp} = \frac{120.404d_i + 87 + 17000P_2}{12000}\ MPa \quad (9)$$

The control curve is presented in fig.10(b). Its response time is the shortest of all these controllers. But the premise is that the axial stiffness must be reliable. And the controller's parameter should reset at different revolving speeds.

Based on the overall consideration of various factors, the fuzzy PID controller is the better choice to this fault self-recovering system.

6 Conclusions

The below conclusions are obtained through the experimental study on axial displacement fault self-recovering system:

(1) The system can control the axial displacement safely by controlling the pressure of balance disk lower pressure side, and avoid the emergency shutdown.

(2) The axial displacement fault can be diagnosed by the online identification of axial stiffness.

(3) The temperature of lube oil has little influence on the stiffness of bearing, but the stiffness varies with the revolving speed. The system adapts better for the high revolving speed compressor.

To conclude, the axial displacement fault self-recovering system has important significance on extending the running period of process production in petrochemical enterprises.

Acknowledgements

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