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THEORETICAL AND EXPERIMENTAL INVESTIGATION OF COMPRESSION LOADS IN TWIN SCREW COMPRESSOR

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ABSTRACT

The compression mechanism in twin screw compressor consists of two helical rotors. An important consideration in the design and analysis of twin screw compressor is the evaluation of the compression loads, which are the forces and moments induced on each rotor due to the gas compression. In this paper, the finite element method (FEM) is presented for computing the compression loads of the compressor. Each rotor is mapped to 3D elements. The elements correspond to rotor surface are loaded by the chamber pressure. The compression loads are computed by integrating the force and moment at each node in every element. To verify the FEM, A test rig is set up, and the axial forces of each rotor are measured by the sensor located between the bearings. The results of theoretical calculation by the FEM are in good agreement with the measured data.
Keywords: twin screw compressor, load, FEM, experimental

1. INTRODUCTION

The twin screw air compressor is a positive displacement compressor and has been widely used in gas industries. One of the major advantages of the twin screw compressor is its flexibility under various operation conditions. It utilizes the continual variations of the space formed between rotor grooves and case of the compressor to compress the gas, so the compression loads occur in the compressor. Thus, a numerical method for computing the compression loads will be useful for designer analyzes, for example determining the bearing loads.

In recent years, the researches on compression loads or bearing loads in twin screw compressor have been presented in the literature. In 1990, Zhou et al. [1] presented the simplest method to compute the compression loads, which simplified the complex rotor profile geometry and affected the magnitudes of compression loads. In order to improve the computation result, G. P. Adams et al. [2][3][4][5] reported a method for compression load calculation based on the rotor profile geometry and pressures at the suction and discharge ends by mapping the 3D rotor surface into a 2D region. The computation accuracy will be enhanced by this solution way at a certain extent. In 1999, W. S. Lee [6] proposed a mathematical model to simulate the behavior of the twin screw air compressor and compute the compression loads. Xing[7] provided an analytical model for computing the compression loads in twin screw compressors.

In this paper, the finite element method (FEM) is used for computing the compression loads of twin screw compressor. The male and female rotors are mapped to 3D elements. The elements correspond to rotor surface are loaded by the chamber pressure, which is measured by the pressure sensor. The compression loads are computed by integrating the force and moment at each node in every element. To obtain the high accurate computation result, the interlobe seal curve is applied in the rotor surface. Also the test rig is set up for measuring the axial forces of each rotor by the force sensor located between two bearings.

2. COMPRESSION LOADS COMPUTATION

2.1 Rotor Meshing

When the compressor is operating, the contact between the male rotor and female rotor forms an interlobe seal curve along the male and female rotors. This interlobe seal curve, i.e. contact line, divides the helical sections of the male rotor or female rotor into separate chambers. A suction chamber is formed between the suction end and the contact line and a compression chamber is formed between the contact line and discharge end. So the contact line is very important for compression loads computation in twin screw compressor, since it is the borderline between the compression pressure and suction pressure chambers. The contact line on the surface of male rotor or female rotor is shown in Fig.1.

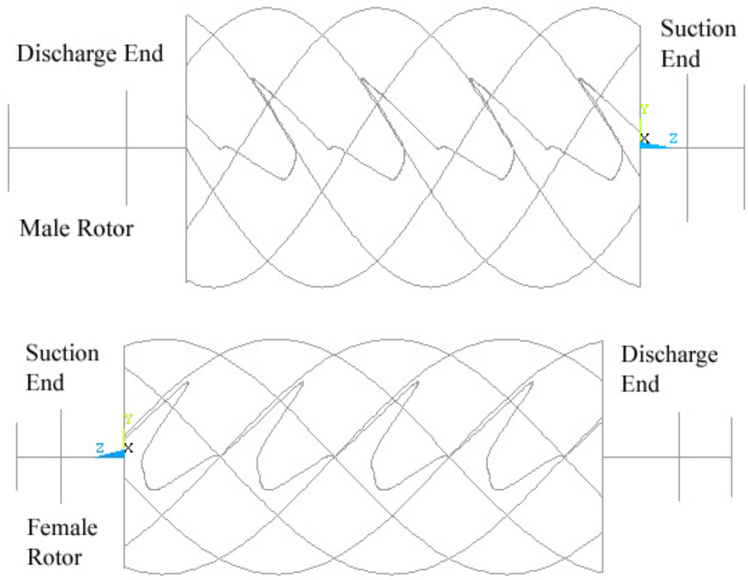

Fig.1 The contact line on the rotor surface

In order to make the pressure acting on the rotor surface loaded on the surface further reasonably, the rotor body should be meshed by the 3D grid elements. In the meshing process, it should be achieved the high grid density without the large deformation that would result from the rotor surface. The rotors meshed successfully are shown in Fig.2. The grids used for the present calculation in male rotor and female rotor are 500,000 and 250,000 volume elements respectively.

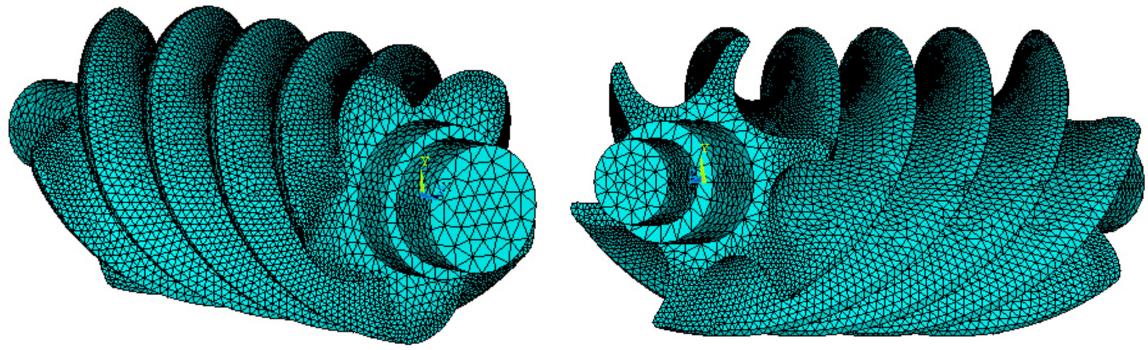

Fig.2 The meshed rotors

2.2 Load Pressure and Computation

The compression loads varies with the pressure change in the compressor lobe, as a function of the rotation angle of the male rotor. The value of the pressure loaded on the rotor surface is important to this computation. In order to obtain the actual pressure in the compressor, the p-V indicator diagram is recorded by the pressure sensor. The indicator diagram recorded is shown in Fig.5. The rotors that loaded the measured pressure are shown in Fig.3.

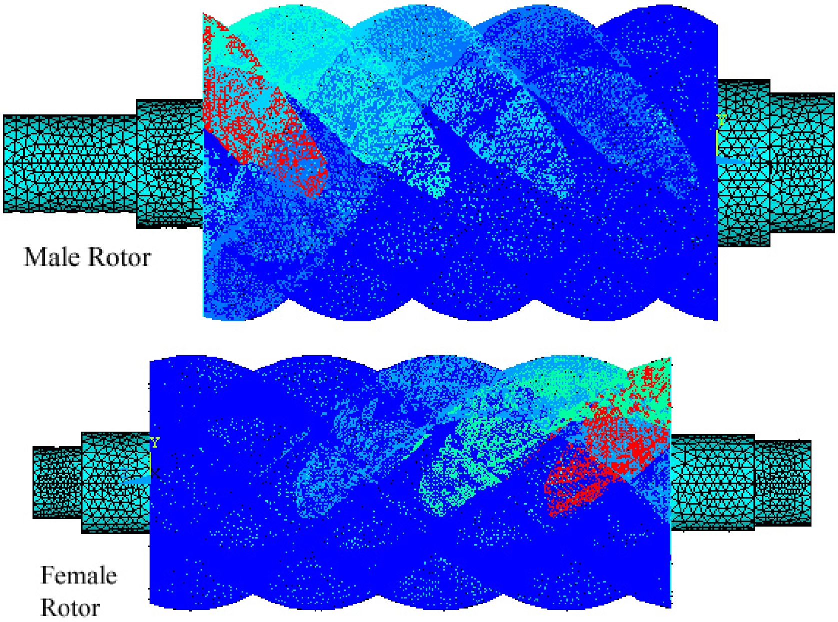

Fig.3 The rotors loaded pressure

The forces of the nodes can be computed by the rotor surface load and the node matrix. So the compression loads are computed by integrating the force and moment at each node in every volume element.

3. COMPRESSION LOADS MEASURED

3.1 Pressure Measured

The pressure in the compressor is measured by a pressure sensor, which installed in the shell of the twin screw compressor, as shown in Fig.4. The pressure in one section of the compression process and discharge process can be recorded, as shown in Fig.5. The micro-sensor is XT-190 pressure sensor, which produced by Kulite Group. Not only this kind of sensor's response frequency is high, but also it has temperature-compensating circuit.

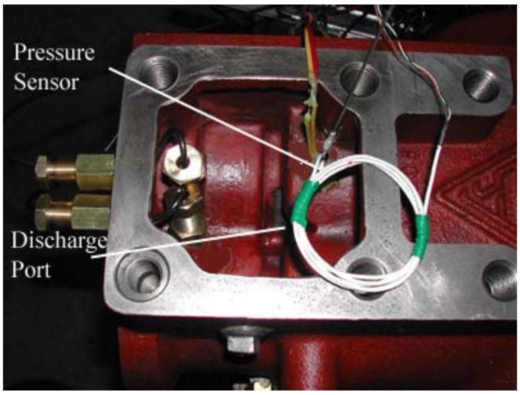

Fig.4 The sensor in the compressor

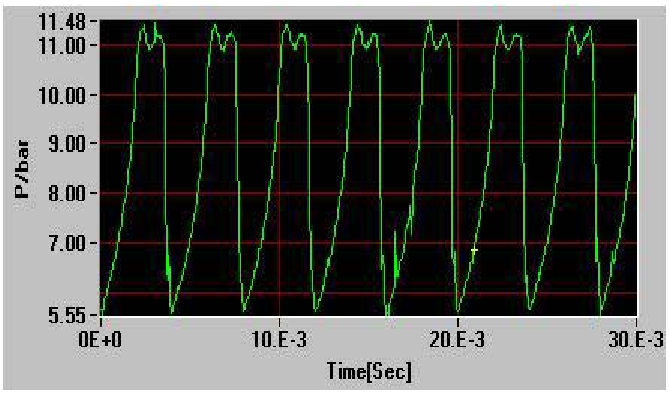

Fig.5 Pressure measured

3.2 Force Measured

The schematic diagram for measuring axial force of the bearing received is shown in Fig.6. The force sensors are installed between the two bearings. To prevent the sensor from rotating with the shaft and sliding random, a tie strap is used to protect the sensor. So the axial force of the bearing received by the compression gas can be measured with the force sensor. The force sensors that mounted in the shaft chamber between the two bearings at the discharge end are shown in Fig.7. This sensor's response frequency can up to 20Hz and be competent for measuring the dynamic force under 3000rpm or 4000rpm working condition.

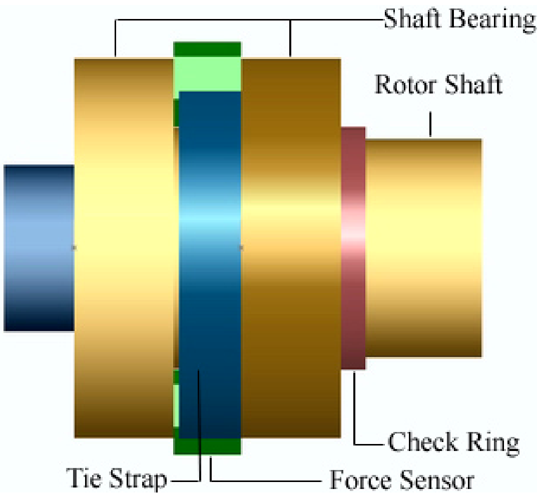

Fig.6 The schematic diagram

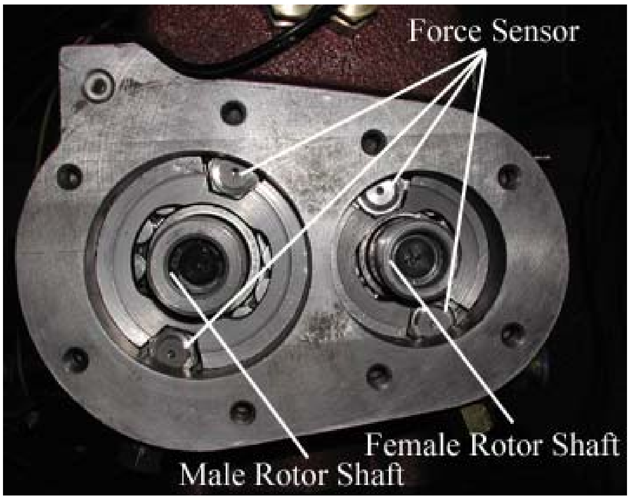

Fig.7 Force Sensors installed in bearing chamber

4. RESULTS AND DISCUSSION

The working condition of the twin screw air compressor for testing is that the suction temperature is 20℃; the suction pressure and discharge pressure is 0.1MPa and 1.1MPa respectively; the rotation speed of male rotor is 3000rpm.

The forces and the Z-axis torsional moment result from the compression gas in the working chamber are computed by the FEM under the above working condition. The axial force and radial force acting on the male rotor and the female rotor are shown in Fig.8 and Fig.9 respectively. From the two figures, we can see that the axial force acting on the male rotor is the maximum and on the female rotor is the minimum in all forces. Also, the radial force acting on the discharge end of the rotor is much larger than that on the suction end. It can be seen that the forces are periodic functions with respect to the male rotor angle due to the symmetry of the rotor.

In addition, the torsional moment about the axis of rotation for male rotor and female rotor is shown in the Fig.10. It can be seen from the chart the moment for the male rotor is much larger than that for the female rotor.

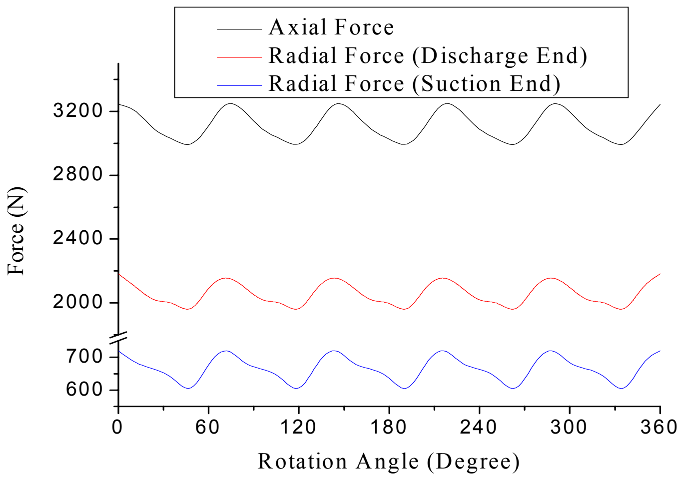

Fig.8 Forces on male rotor

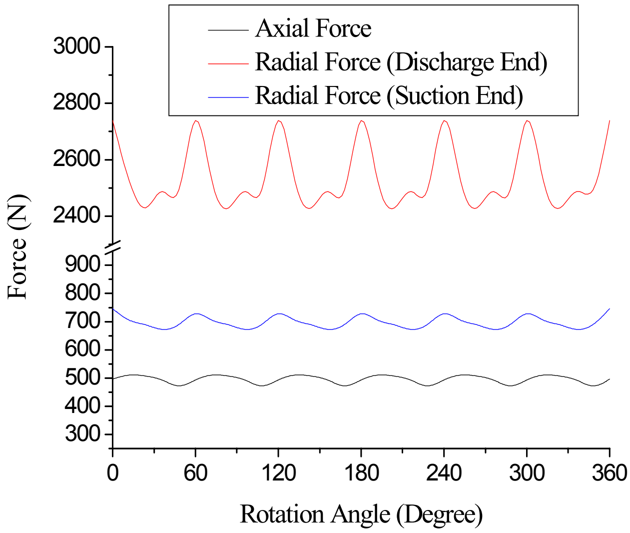

Fig.9 Forces on female rotor

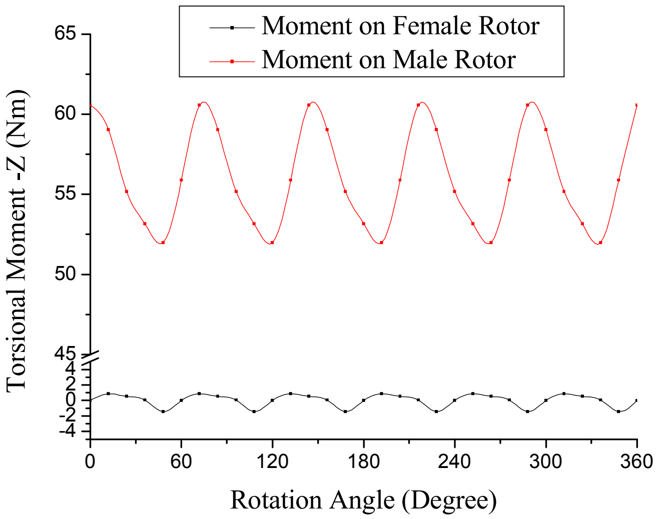

Fig.10 Torsional moment

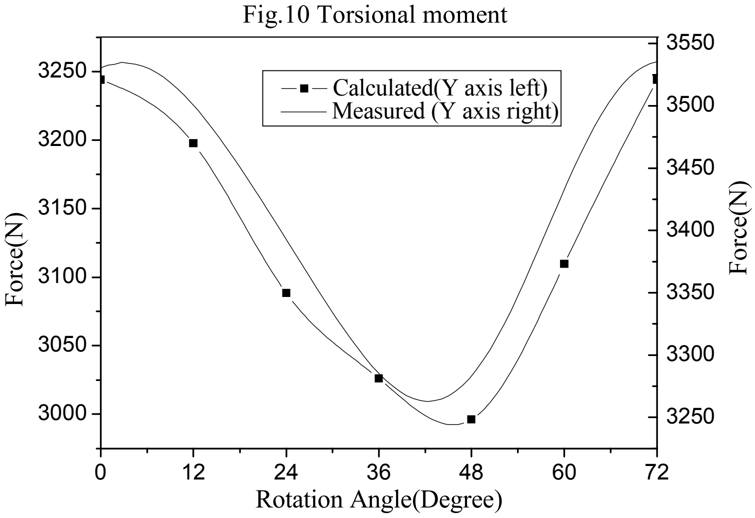

Fig.11 Comparison of the axial forces on male rotor

The measured result of the axial force acting on the male rotor is shown in the Fig.11, compared with the calculated result. It can be seen that the tendency of the calculated result of the axial force changing with the male rotor rotation angle is coinciding with the measured force in substance. It is known that the axial force consists of two parts. One is resulted from the gas acting on the discharge and suction end surface, the other is owing to the gas acting on the rotor lobe surface. So the difference values between the measured result and calculated result can be partly explained by the fact that the imperfect pressure is loaded on the discharge end surface, which can not be measured by the pressure sensor or calculated exactly.

The average moment measured is 63.7 Nm. The average moment calculated is 56.6287 Nm, which is not considering the moment resulted from the friction between the male and female rotor. The error between the two above moments is 11.1%. However, the mechanical efficiency of the compressor is 91.5%, which can be calculated by the indicated power and the shaft power. The indicated power of the compressor calculated by the indicator diagram recorded is 18.326 kW. The shaft power measured is 20.027 kW. The power due to the friction is 8.5% of the shaft power. Certainly, the friction also wastes the moment that is 8.5% of the moment measured. So it can be believed that the calculated result can be coincided with the measured result.

5. CONCLUSIONS

A method for computation of compression loads in the twin screw compressor is provided using the FEM. Thus, it can be applied to arbitrary rotor profiles and various working conditions. In order to verify the method, a test rig for measuring the indicator diagram, axial force, moment and power is set up. The result calculated by the FEM shows good agreement with the measured result. So it is useful in obtaining forcing input for simulating dynamics of the rotors during operating. Also, it is helpful for designer determining the bearing loads in the screw compressor designing.

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