Study on the performance prediction of dry twin screw vacuum pump

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Abstract. Dry twin screw vacuum pump, which generates no oil and gas during operation, inherits the advantages of screw machinery, such as high reliability, stable medium conveying, small vibration, simple and compact structure, convenient operation, etc. which has been widely used in petrochemical and new or high-technology industries, like vacuum heat treatment, nuclear research, micro electro mechanical systems, nanotechnology, precision manufacturing and national defense technology. In this paper, the geometric feature of the twin screw vacuum pump such as contact line, meshing line and chamber volume between teeth is analysed, respectively. In order to study the micro working process within the dry twin screw vacuum pump, a test rig has been designed and built and the p-V indicator diagrams of the working process of dry twin screw vacuum pump have been recorded successfully under various working conditions. Based on the indicator diagrams, the working process within the dry twin screw vacuum pump is analysed in more detail, and especially, the effects of the suction pressure as well as rotational speed on the performance of dry twin screw vacuum pump, such as pumping speed and ultimate pressure, is thus investigated. The results presented in this paper are helpful to deepen understanding of the working process inside dry twin screw vacuum pump and offers a good reference for to improve design and optimize such machines.

1. Introduction
The dry twin screw vacuum pump, which generates no oil and gas during operation, is a novel positive-displacement device capable of “pumping” gas. Its core element consists of a pair of inter-meshing screw rotors rotating in opposite direction about parallel axes, and it removes gas from a sealed working chamber in order to leave behind a partial vacuum. The meshing of a pair of conjugated male and female rotors inside the housing of the dry twin screw vacuum pump allows suction, compression and discharge in the screw channel and the space inside the housing wall. Additionally, the dry twin screw vacuum pump inherits the advantages of screw machinery, such as high reliability, stable medium conveying, small vibration, simple and compact structure, convenient operation, etc. So, it is widely used in low and medium-low vacuum applications, such as semiconductor industry, metallurgical industry, nanomaterials industry, chemical industry, pharmaceutical industry and high-technology industry, et al.

In recent years, a lot of investigations have been done on the performance prediction of the dry twin screw vacuum pump under different operating conditions. Ohbayashia[1] established a method of the performance prediction and proposed a way to design the pump that satisfies specific requirements. The leaks flow through clearances between the screw rotor and housing, and clearances between two
meshing rotors were estimated with the results based on the linearized BGK model. Pfaller[2] presented a method based on the evolutionary calculation algorithm for the energetic optimization of the dry twin screw vacuum pump, accordingly changing the pitch of a rotor at constant design volume. Feng[3] and Huang[4] analyzed the deformation of rotor and the performance of the dry twin screw vacuum pump. Compared with the experimental results, the calculated results obtained from the performance prediction method can agree well. Zhao[5] and Wang[6] derived a theoretical formula to calculate the leakage in the dry twin screw vacuum pump, but there was no experimental validation.

Although much research have been performed on the theoretical and experimental study of the dry twin screw vacuum pump, there is seldom analysis on the thermodynamic process of the dry twin screw vacuum pump based on the recorded p-V indicator diagram. In this paper, the geometric feature of the dry twin screw vacuum pump, such as contact line, meshing line and working volume between teeth is analyzed, respectively. In order to study the micro working process within the dry twin screw vacuum pump, a test rig has been designed and built and the p-V indicator diagrams of the working process of dry twin screw vacuum pump have been recorded successfully under various working conditions. Based on the indicator diagrams, the working process within the dry twin screw vacuum pump is analyzed in more detail, and especially, the effects of the suction pressure as well as rotational speed on the performance of the dry twin screw vacuum pump, such as pumping speed and ultimate pressure is thus investigated. The results presented in this paper are helpful to deepen understanding of the working process inside dry twin screw vacuum pump and offers a good reference for to improve design and optimize such machines.

2. Geometrical parameters

The dry twin screw vacuum pump is a type of fluid-extracting apparatus, which consists of two partly overlapping cylinders and two same-shaped meshing rotors. The two rotors rotate in opposite directions around their axes and are synchronized by a pair of gears and maintain only tiny clearances during operation. The rotor profile plays a significant role on the performances and applications of the dry twin screw vacuum pump since it permits the rotors to remain meshed with each other.

A novel rotor profile, which is completely conjugate, is proposed, as shown in Figure 1. Taking the male rotor as an example, it contains 5 curves: extended epicycloid ab, root circular arc bc, pseudo Archimedes cd, envelope curve of pseudo Archimedes de and head circular arc ea. The equations as well as design procedures of the rotor profile can be found in the paper by Yang[7] published in 2015.

![Figure 1. The profile of the rotors.](image)

2.1. Rotor Parameters

The design parameters of the male and female rotors are shown in Table 1 commonly, because their parameters are completely the same. The rotor pitch is divided into two sections: one section of the pitch is 88mm, and the other section of the pitch is 60 mm.
Table 1. Geometry parameters of screw rotor

| Item                  | Value |
|-----------------------|-------|
| Outer radius/ mm      | 130   |
| Root radius/ mm       | 66    |
| Rotor length/ mm      | 340   |
| First rotor pitch/ mm | 88    |
| Second rotor pitch/ mm| 60    |
| First rotor length/ mm| 200   |
| Second rotor length/ mm| 140   |

2.2. Calculation of the sealing line

The sealing line has a significant effect on the thermodynamic performance of the twin screw vacuum pump. The equation of the sealing line can be obtained by the simultaneous of meshing condition and the equation of the rotor profile. Figure 2 shows the sealing line of the rotor profile, where \( Z \) is the axial length of the rotor and \( Y \) is the upright length of the rotor. The section \( LM \) and \( PQ \) are generated by the meshing of point and the extended epicycloid; the section \( MN \) and \( OP \) are generated by the meshing of the head and root circular arc; the section \( NO \) is generated by the meshing of the pseudo Archimedes and the envelope curve of pseudo Archimedes.

2.3. Calculation of chamber volume

The operation of the twin screw vacuum pump is based on the change of the chamber volume enclosed by the male rotor, female rotor and the housing (working volume). The chamber volume of the twin screw vacuum pump can be expressed by:

\[
A = \sum_{i=1}^{n} \frac{1}{2} \int_{t_i}^{t_{i+1}} \left[ \dot{y}_i \dot{x}_i - \dot{x}_i \dot{y}_i \right] dt
\]

Where \( A \) is the tooth area of the tooth, \( x_i = x_i(t) \), \( y_i = y_i(t) \), are the parametric equation of the \( i \)-th curve of the rotor profile, \( \dot{x}_i = dx_i(t)/dt \) and \( \dot{y}_i = dy_i(t)/dt \) are the derivative of the parameter, \( t_i \) and \( t_{i+1} \) are the range of parameter.

\[
V_0 = \frac{T}{2\pi} \int_{0}^{\theta_{Max}} A(\theta) d\theta
\]
Where $V_0$ is the chamber volume, $T$ is the pitch of the rotor and $\theta$ is the rotation angle of the rotor.

The chamber volume of the dry twin screw vacuum pump versus the rotation angle of the male rotor is shown in Figure 3. Obviously, the maximum volume of the working chamber is 0.767 L. And the twin screw vacuum pump contains six processes during operation: suction, transportation, internal compression, transportation, compression and discharge.

![Figure 3](image)

**Figure 3.** Chamber volume varies with the rotation angle. (①—suction, ②—transportation, ③—internal compression, ④—transportation, ⑤—compression, ⑥—discharge)

3. **Experimental setup**

3.1. **Test rig**

A schematic diagram of the dry twin screw vacuum pump test rig is shown in Figure 4 and the photograph of which is shown in Figure 5. The dry twin screw vacuum pump prototype is driven by a variable-speed motor. A tacho-torque meter with an uncertainty of ±0.1% reading is coupled between the motor and dry twin screw vacuum pump for measurements of the shaft speed and power. The variable rotation speed of the male rotor is obtained by a frequency-converter. The pressure in the test cover is measured by a PSG 500 type vacuum gauge made by Inficon. Its measurement range is from 0 Pa to $10^5$ Pa and the accuracy is less than 2% of reading. The revolving type flow meter with an uncertainty of 0.5% is used to measure the vacuum pump flow rate. Data obtained from the measurement system mainly include pumping speed, ultimate pressure and pressures at various positions.
3.2. Recording of the p–V indicator diagram

As we know, p–V indicator diagram is an important and efficient method to investigate the thermodynamic performance of displacement machines. In order to study the micro working process within the dry twin screw vacuum pump, a test rig has been designed and built. Firstly, the installment of the pressure sensors is discussed. Five GE pressure sensors and one Kulite pressure sensor are installed in consecutive positions in its housing to measure the p-V diagram of the studied dry twin screw vacuum pump. The model UNIK 5000 pressure sensors are made by GE Group and have an accuracy of less than ±0.04% of full scale and a response frequency of 3.5 kHz. The model XTL-140M pressure sensor is made by Kulite Group and has an accuracy of ±0.3% of reading and a response frequency of 10 kHz. Each pressure sensor has some overlap with the following pressure sensor in order to continuously monitor pressure changes in the compression process, as shown in Figure 6. However, since there is no position for installing the pressure sensor in the compressor discharge housing, the pressure in the late part of discharge process cannot be measured in this study. Therefore, for this part, the discharge pressure is considered approximately equal to the atmosphere pressure.
The installation of pressure sensors on the casing of dry twin screw vacuum pump is shown in Figure 7. The sealing ring can prevent the ambient gas from going into the working chamber. There should have some distance between the sensor film and the pressure-leading system to avoid the damage of sensor. The design of pressure-leading system can be referred to the following equations[8]:

$$\xi = 2 \sqrt{\frac{3}{\pi} \frac{\mu}{\rho c} \frac{\sqrt{IV}}{r^3}}$$  \hspace{1cm} (3)

Where $\xi$ is damp coefficient, $\mu$ is the dynamic viscosity, $r$ is the radius of pressure-leading pipe, $l$ is the length of leading pipe, $V$ is the volume of the sensor pocket, $\rho$ is the density, $c$ is the speed of sound.

$$\omega_1 = \sqrt{\frac{3\pi}{2} \frac{rc}{\sqrt{IV}}}$$  \hspace{1cm} (4)

$$\frac{p_1}{p_2} = \frac{1}{\sqrt{1 - \left(\frac{\omega_2}{\omega_1}\right)^2 + 4\xi^2 \left(\frac{\omega_2}{\omega_1}\right)^2}}$$  \hspace{1cm} (5)

$$\varphi = -\arctan \left[ \frac{2\xi \left(\frac{\omega_2}{\omega_1}\right)}{1 - \left(\frac{\omega_2}{\omega_1}\right)^2} \right]$$  \hspace{1cm} (6)

Where $p_1$ is the pressure of the sensor film, $p_2$ is the pressure within the working chamber, $\omega_1$ is the angular frequency of the pressure signal of the sensor film, $\omega_2$ is the angular frequency of the pressure signal within the working chamber.

In general, with the increase of the angular frequency of the pressure signal within the working chamber $\omega_2$ and decrease of damp coefficient $\xi$, the dynamic performance of the leading system increases and the distortion of the pressure signal decreases.

Secondly, in order to determine a phase position of the recorded pressure, a photoelectric sensor is used. When the dry twin screw vacuum pump working process begins, the photoelectric sensor will produce an impulse. Based on the pressure signal and the position signal, the pressure distribution versus the rotational angle of the rotor can be obtained. By transformation between the rotational angle of rotor and the volume of working chamber as explained by Xing[9], the diagram of pressure

![Figure 7. Installation of pressure sensors.](image-url)
distribution versus chamber volume can be carried out. The pressure signal is collected and analyzed in the dynamic data acquisition system NI-PXIe1078. The model of data acquisition card is PXIe-6356. This card can provide 8-differential and 16-single ended measurement. The differential measurement is chosen in order to eliminate the impact of common-mode voltage on the signal. Because the signal to be measured is floating type, bias resistor should be applied to satisfy the bias current path requirement. In the experiment, one resistor connected between the negative input and the com terminal is required because the input signal of data acquisition card is DC-coupled and the impedance of pressure sensor is low. Moreover, the same isolated power source is used to supply the power of pressure sensors and the data acquisition card to improve accuracy of measurement. Figure 8 shows typical p-t measurements from the signal collecting and analyzing system when the compressor is operating at a suction pressure of 4kPa and rotation speed of 3000 r/min.

![Figure 8](image)

**Figure 8.** A typical p-t diagram obtained in the experiment.

4. Results and discussion

4.1. Effects on p–V indicator diagram

As we know, p–V indicator diagram is an important and efficient method to investigate the performance of displacement machines. The change of the gas pressure in the working chamber can be described in detail by the p–V indicator diagram. In addition, it can be used to estimate the indicated power consumption of the dry twin screw vacuum pump.

The p–V indicator diagrams of the dry twin screw vacuum pump with variable suction pressures are shown in Figure 9. Here, the rotation speed of the rotor is 3000 r/min and the suction pressures are 4kPa and 6kPa, respectively. It can be seen from Figure 9 that the dry twin screw vacuum pump with the suction pressure of 6kPa has a higher pressure almost during the whole working process than that of 4kPa. This can be explained by the fact that the larger the suction pressure is, the larger the back flow of gas is. It also can be seen that the pressures in the discharge process with variable suction pressures are almost equivalent.

It also can be seen that the area of the p–V diagram is different for different suction pressures. And the area for the case with 6kPa suction pressures is larger than that with 4kPa suction pressures. It is thought that the pressure within the working chamber becomes higher, causing the indicated power to rise up, as the suction pressure increases.
Figure 9. The diagrams of dry twin screw vacuum pump with variable suction pressures.

Figure 10 shows the p–V indicator diagrams of the dry twin screw vacuum pump at different rotation speeds of 3000/min and 2700/min, and a suction pressure of 4kPa. As seen from Figure 10, the pressure during the whole working process of rotation speed of 2700 r/min is higher than that of 3000 r/min. This is because of the fact that the heat generated during the whole working process of dry twin screw vacuum pump increases as the rotation speed increases and the temperature difference between the housing and rotors increases with the rotation speed, so the clearances between them become narrow, which results in smaller backflow occurs from the discharge chamber into working chamber and also lead to larger pressure increase. Moreover, the pressure level inside the working chamber under high speed is also higher than that under low speed. It’s can be partly explained by the fact that a reduced fraction of a total slip passes through the internal clearance as rotational increases.

Figure 10. The diagrams of dry twin screw vacuum pump at variable rotation speeds.

4.2. Pumping speed

The pumping speed of the dry twin screw vacuum pump, which is based on the ISO 1607 standard[10], is measured by varying the frequency from 40 to 50 Hz with a 5 Hz interval. The changes of pumping speed with the rotation speed and suction pressure are shown in Figure 11. It can be seen that the pumping speed increases sharply when the suction pressure is less than about 200 Pa with the increase of the suction pressure and then decreases slowly when the suction pressure is in other ranges at the
same rotation speed. Moreover, it can be seen that the pumping speed increases monotonously with the rises of the rotation speed at the same suction pressure. So, the increases of pumping speed with rotation speed can be explained by the increases of volumetric efficiency with rotation speed. Obviously, the suction pressure corresponding to the maximum value of the pumping speed is about 200 Pa.

![Figure 1. Pumping speed for different suction pressures.](image1)

**Figure 11.** Pumping speed for different suction pressures.

4.3. *Ultimate pressure*

The experimental results of ultimate pressure for different rotation speeds are shown in Figure 12. It can be observed that the ultimate pressure goes down rapidly with the increment of speed in a certain range. As the speed is large enough, the changing rate of ultimate pressure with speed becomes is less and less. It can be explained by the two reasons: one is that the temperature of the rotors rises higher than that of the housing because the housing are cooled more easily than the rotors. The heat generated during gas compression phase increases as the rotation speed increases and the temperature difference between the housing and rotors increases with the rotation speed, and the clearances between them become narrow, which results in the back flow of gas lessen; two is that the external leakages go down with the increment of speed, resulting in an increase in the volumetric efficiency, so the ultimate pressure decrease. The minimum ultimate pressure of the dry twin screw vacuum pump is 0.7Pa.

![Figure 2. Ultimate pressure.](image2)

**Figure 12.** Ultimate pressure.
5. Conclusions
In order to study the micro working process within the dry twin screw vacuum pump, a test rig has been designed and built and the p-V indicator diagrams of the working process of dry twin screw vacuum pump under various working conditions have been recorded successfully. Based on the indicator diagrams, the working process within the dry twin screw vacuum pump is analyzed in more detail, and especially, the effects of the suction pressure as well as rotational speed on the performance of dry twin screw vacuum pump is thus investigated. Some main conclusions are drawn as detailed below:

(1) The suction pressure and rotational speed have a direct influence on the p-V indicator diagrams of the dry twin screw vacuum pump. The pressure curve during the whole working process has a higher level under the condition of higher rotational speed and higher inlet pressure.

(2) The pumping speed of the dry twin screw vacuum pump increases with the increase of the suction pressure at the same rotation speed. Also, it increases monotonously with the rise of the rotation speed at the same suction pressure.

(3) The ultimate pressure goes down rapidly with the increment of speed in a certain range. As the speed is large enough, the changing rate of ultimate pressure with speed becomes is less and less.

(4) The integrated experimental studies of the dry twin screw vacuum pump make it possible to create experimental data bank concerning pressure dependence in working chambers on rotors rotating angle. These results can be used for further development of the pump design and optimization with the help of mathematical model.

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ISO 1607-1:1993 Standard Positive-Displacement Vacuum Pumps—Measurement of Performance Characteristics. Part 1: Measurement of Volume Rate of Flow (Pumping Speed). ISO 1607-2:1989 Standard Positive-Displacement Vacuum Pumps—Measurement of Performance Characteristics. Part 2: Measurement of Ultimate Pressure (International Standardization Organization).