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Theoretical and Experimental Studies on Oil Injected Twin Screw Air Compressor under Unload Conditions

Theoretical and Experimental Studies on Oil Injected Twin Screw Air Compressor under Unload Conditions

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Abstract. The working process of twin screw air compressor under unload conditions was analyzed. Mathematical model describing the working process of oil injected twin screw compressor under unload conditions was established based on the laws of perfect gas and standard thermodynamic relations, considering the effects of oil injection, gas-oil heat transfer, internal leakage etc. A simulation model of screw air compressor system based on MATLAB was built up, and the effects of rotational speed and discharge pressure etc. on the performance under unload conditions were analyzed. The experiment was carried out to validate the model by a comparison between the results of the simulation model and testing. The simulation and testing results both showed that volumetric efficiency increased with the increase of rotational speed, when the discharge pressure was reduced. The shaft power increased with the increase of discharge pressure and rotational speed. The flow rate increased with the reduction of discharge pressure and increase of rotational speed. This model can be used for performance prediction under unload conditions and it provided reference for the high efficiency and energy saving operation of screw compressors.

1. Introduction

The twin screw compressors are rotary positive displacement machines. A couple of intermeshing twin screw rotors is placed in a suitable chamber to produce several volume-reducing cavities for gas compression. They can operate efficiently at high speeds over a wide range of flow rates and operating pressures. In addition, they are compact, simple and reliable. Therefore, the twin screw compressors are widely used in the air conditioning systems, refrigeration systems, mechanical vapor compression systems and pneumatic transport applications, taking into account their advantages [1-5].

A number of papers have been presented on the behavior of twin screw compressors in recent years. Fujiwara and Osada investigated the heat transfer coefficient and flow coefficient experimentally and used in mathematical model for twin screw air compressors [6]. Stosic et al. established a mathematical model to simulate the working process of screw refrigeration compressors, and studied the influence of fluid injection and fluid leakage upon the screw compressor thermodynamic process experimentally and theoretically [7, 8]. Seshaiah et al. developed a mathematical model for low cost air compressors adopted for different gases and analyzed the effect of some of the operating and design parameters on compressor performance theoretically and experimentally [9, 10]. Fleming et al. established a mathematical model of the working process and studied the effect of internal leakage and gas-oil heat transfer etc. on screw refrigeration compressor performance [11, 12]. Chen et al. described a mathematical model of the screw refrigeration compressor on the basic of radial discharge area and effective by-pass area to study the performance of compressor with slide valve under part load conditions by theory and experiment [13]. Krichel and Sawony presented a simulation model to describe the dynamic behavior of twin screw air compressors and verify the model by experimental study of three types of screw compressors. [14]. Chamoun et al. developed a new model of twin screw compressor with water injection for high temperature heat pump by using Modelica and investigated the performance of the compressor in different configurations [15]. A thermodynamic model of water injection twin screw vapor compressor was presented by Tian et al. and performance of a twin screw water vapor compressor

Was researched [16]. Li et al. established a mathematical model of water-injection twin screw compressor. In order to verify the accuracy of the model, the performance of a prototype twin screw air compressor was tested [17]. Liu et al. proposed a mathematical model to predicting the performance of twin screw refrigeration compressors and verified the model by experiment [18].

However, most previous studies of the twin screw compressor were focus on the working process under full load conditions. Research on the performance of twin refrigeration compressor under part load conditions has also done in the literature. However, the structure of twin screw refrigeration compressors is also different from that of twin air compressors.

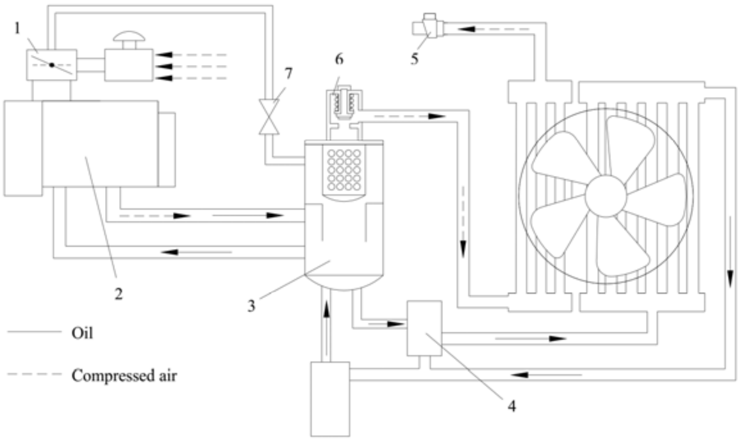
Since the compressed air requirement varies, the twin screw air compressor must limit its supply to reduce energy input and prevent the buildup of pressure under low compressed air demand periods. In order to meet the load variation, the compressed air system is frequently operated under unload conditions. The twin screw air compressors consume a lot of energy under unload conditions even if the compressors don’t supply compressed air to the air network. Thus it is necessary to research the performance of twin screw air compressor under unload state to improve energy efficiency. However, few of the previous studies focused on the performance of screw compressors under unload conditions until now.

Considering the lack of study on the screw air compressor under unload conditions, a mathematical model for describing the working process inside twin screw air compressors under unload conditions is presented. The model correlates some operating parameters and some key design parameters of the twin screw air compressor. In addition, an experimental study on performance of a prototype twin screw air compressor under unload condition is carried out. The model is able to predict the compressor’s performance under unload condition.

2. Mathematical modeling of compressor cycle

2.1. Working process analysis under load/unload conditions

The working process of twin screw compressor is shown in Fig. 1. During the start-up process, the inlet valve 1 opens slowly and air is sucked in. When the inlet valve 1 opens, compressed air begins to fill the separator. During the start-up phase, the pressure is low, so the minimum pressure valve closes, and air is released into the ambient atmosphere through the throttle valve. When switching to load conditions, the pressure in the separator rises, so it activates the minimum pressure valve. When the pressure is high enough, and the air is blown through the minimum pressure valve and outlet throttle valve to the air network. The pneumatic resistance across the minimum pressure valve is not considered for simplicity. When the air supplied by the compressor is more than the air consumption of system, the system pressure will increase. If the system pressure reaches the preset unload pressure, the controller sends a signal to the inlet valve and the compressor turn into unload process. Then, the inlet valve closes, but a small flow of air is still sucked in through the hole in inlet valve. In this way, the oil cycle runs continuously to keep the compressor lubricating and cooling at unload state. The pressure air in the separator is discharged to the inlet port of air compressor through valve 7. Thus, the minimum pressure valve closes due to the pressure drop and air delivery to net is stopped.



1-Inlet valve, 2-Twin screw compressor, 3-Oil/air separator, 4-Thermovalve, 5-Outlet throttle valve A, 6-Minimum pressure valve, 7-Blowdown valve

Figure 1. System of oil injected screw compressor

2.2. Modeling of suction process

The working cycle of screw compressor can be divided into suction, compression, and discharge processes. The spaces formed between the intermeshing rotors form a series of working chambers. Gas is contained in these working chambers. At the beginning of suction process, there is a starting point for each chamber and the trapped volume is initially zero. The volume of the chamber increases with the rotation of screw rotors. At the end of the suction process, the volume of chamber has reached its maximum value. The working chambers are filled with gas as they pass the suction port. The connection of the inlet valve and chambers is closed, forming a compression chamber with further rotation. As the rotors turn to compress the gas, the compression chamber volume decreases, compressing the gas toward the discharge port. The gas exits from the compressor as the compression chamber passes the discharge port and discharge continues till the male rotor lobe completely disengage from female rotor groove. This process is quasi-continuously through the high rotational speed of the rotor.

The inlet temperature of oil and gas, and the pressure drop across inlet port are assumed to be constant in the suction process, duo to the pressure and temperature fluctuations during suction process are generally small.

The inlet valve is fully opened under load conditions, and the inlet pressure is identical with ambient pressure (assuming that the pressure loss of air flowing through the inlet valve is neglected).

The inlet valve is closed under unload state, and only a small amount of air is sucked into the compressor through holes in the inlet valve in this state. This leads to the inflow of oil, which reduce the temperature rise of compressor and lubricate the rotor motion. As the inlet valve closes, the amount of the air that has to be sucked in is reduced and pressure values immediately after the inlet valve (suction pressure) are under ambient pressure. It’s calculated as below:

$$P_s = P_{atm} - K_v \frac{{v_{in}}^2}{2g} \tag{1}$$

The mass flow rate of gas that pass through the inlet valve and is inducted into the working chamber will not be reduced. Based on law of conservation of mass, the mass flow rate is expressed as:

$$m_s = m_{in} = \frac{P_{in}\dot{Q}_{in}}{RT_{in}} == \frac{P_s\dot{Q}_s}{RT_s} \tag{2}$$

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By assuming that there is no temperature increase across the valve, the velocity of air immediately before the inlet valve is calculated by the equation below:

$$v_{in} = \frac{\dot{Q}_{in}}{A_{in}} = \left(1 - \frac{K_v v_{in}^2}{2RT_{in} g}\right)\frac{\dot{Q}_s}{A_{in}} \tag{3}$$

Owing to the heat transfer of the compressed gas in the compression process, the temperature of the rotors and the casing in suction cavity which are covered with a film of oil leaks from the compression working chamber is higher than the temperature of intake air during suction process. The intake air is preheated and the temperature of intake air rises. This process is assumed to be an isobaric process at pressure P_s.

The average temperature of air at the end of suction process may be expressed as follow [9]:

$$T_e = \frac{hA(T_{oil} - T_s)t_r}{c_p M_l} + T_s \tag{4}$$

At the end of the suction process, the total mass of air in the suction cavity is the sum mass of the leaked air and the intake air, which can be calculated by:

$$M_{s1} = M_s + M_{il} \tag{5}$$

At the end of the suction process, the leakage mass of air in the suction cavities is calculated by:

$$M_{il} = m_l t_r \tag{6}$$

The gas mass (including leakage gas) in the working chamber at the end of suction process can be described as follow:

$$M_{s1} = \frac{P_s V_t}{RT_e} \tag{7}$$

2.3. Modeling of compression and discharge process

The pressure of gas in the working chamber is increased and thermodynamic properties of the oil and gas are changed continuously by the rotation of rotors.

The following assumptions are made in the calculations of the working process:

1. Oil is an incompressible fluid, and working gas is an ideal gas with constant specific heat.
2. Oil and gas never change phase.
3. Pressure, oil temperature and gas temperature are homogeneous throughout the working space at any stage.
4. Heat transfer between oil and gas and is in proportion to the temperature difference between them.
5. Oil droplets are non-deformable spheres of equal radius, droplets interaction and the effect of turbulence on the droplets is not considered.

According to the first law of thermodynamics, the internal energy of gas through the working chamber is described by:

$$\frac{dU}{d\varphi} = \frac{dM_i}{d\varphi}h_i - \frac{dM_o}{d\varphi}h_o - P\frac{dV}{d\varphi} + \frac{dQ}{d\varphi} \tag{8}$$

Based on the conservation of mas, gas mass changing due to leakage in the working chamber can be written as:

$$\omega \frac{dM_g}{d\varphi} = m_{gi} - m_{go} \tag{9}$$

Conservation law of oil mass in the working chamber is expressed by:

$$\omega \frac{dM_l}{d\varphi} = \frac{dM_{li}}{d\varphi} - \frac{dM_{lo}}{d\varphi} \tag{10}$$

The gas temperature change rate is written as follow:

$$\frac{dT_g}{d\varphi} = \frac{1}{M_g R}\left(-RT_g \frac{dM_g}{d\varphi} + P\frac{dV_g}{d\varphi} + V\frac{dP}{d\varphi} \right) \tag{11}$$

The rate of oil temperature change can be written by:

$$\frac{dT_l}{d\varphi} = -\frac{T_l}{M_l}\frac{dM_l}{d\varphi} + \frac{T_{li}}{M_l}\frac{dM_{li}}{d\varphi} - \frac{T_{lo}}{M_l}\frac{dM_{lo}}{d\varphi} + \frac{hA}{M_l c_l}\left(T_g - T_l\right) \tag{12}$$

The rate of gas pressure change is written by:

$$\frac{dP}{d\varphi} = -\frac{1}{V_g}\left(\frac{kPdV}{d\varphi} + \frac{k-1}{\omega}Q_c + \frac{k-1}{\omega}Q_l \right) \tag{13}$$

Q_c is the heat transfer between gas and casing, which is expressed by:

$$Q_c = hV_t^{2/3}(T_g - T_c) \tag{14}$$

The heat transfer between the oil and gas can be given as follow:

$$Q_l = m_l c_p (T_g - T_l) \tag{15}$$

2.4. Leakage model

The internal geometry of twin screw compressor is such that they have clearances between the rotors and their housing, clearances between two rotors. Because of the pressure difference, the mixture of gas and oil in the working chamber move from the high pressure side to the low pressure side through these clearances. These clearances compose the leakage paths of twin screw air compressor. The major leakage paths are leakage through inlet end face, outlet end face contact line, blow hole and rotor tip.

The flow through the leakage paths should be treated as two-phase flow, and the mass flow of gas and oil leakage through the leakage paths can be defined by[19]:

$$m_g = \alpha A_c v_g \rho_g \tag{16}$$

$$m_g = \alpha A_c v_g \rho_g \tag{17}$$

$$m_l = \left(1-\alpha\right) A_c v_l \rho_l \tag{18}$$

The velocity of oil and gas may be written by:

$$v_g = C\sqrt{\frac{2kR\left(T_g - T_l\right)}{k-1}} \tag{19}$$

$$v_l = v_g / S \tag{20}$$

$$\alpha = \left(1 + \frac{1-\chi}{\chi}\frac{\rho_g}{\rho_l}S\right)^{-1} \tag{21}$$

$$S = 0.4 + 0.6\left(\frac{\rho_l}{\rho_g} + 0.4\frac{1-\chi}{\chi}\right)^{\frac{1}{2}}\left(1 + 0.4\frac{1-\chi}{\chi}\right)^{\frac{1}{2}} \tag{22}$$

The geometric parameters of compressors and the changes in the compressor working chambers are related to the rotation angle of male rotor. All the equations obtained above are used for calculating the working process of twin screw compressor under unload conditions. At the beginning of calculation, geometric parameters such as volume of working chamber and leakage area etc. and the initial values of running parameters such as rotation speed, suction pressure and temperature etc. have to be defined. Then all the changes in the compressor working chambers can be solved by step-by-step calculation with Runge-Kutta procedure.

2.5. Power and efficiency
The thermodynamic work of the compressor can be written as:

$$W_t = \frac{k}{k-1}P_s V_t\left[\left(\frac{P_c}{P_s}\right)^{\frac{k-1}{k}} - 1\right] + \left(P_d - P_c\right)V_d \tag{23}$$

$$P_d = P_s \cdot \varepsilon^k \tag{24}$$

$$V_d = V_t / \varepsilon \tag{25}$$

The theoretical indicated power required for the air screw compressor is given by the expression:

$$N_t = W_t \frac{n}{60} \tag{26}$$

The power dissipated by oil injected depends on the pressure difference between the discharge pressure and the pressure at the injection position, and the power loss can be calculated as:

$$N_{oil} = \dot{Q}_{oil}(P_d - P_{inj}) \tag{27}$$

The effects of viscous friction on the power lost can be represented as follows:

$$N_{fr} = \beta\mu V_t \omega^2 \tag{28}$$

On the basis of equations (26), (27) and (28), the shaft power of twin screw compressor is calculated as:

$$N_s = N_t + N_{oil} + N_{fr} \tag{29}$$

The volumetric efficiency can be determined by:

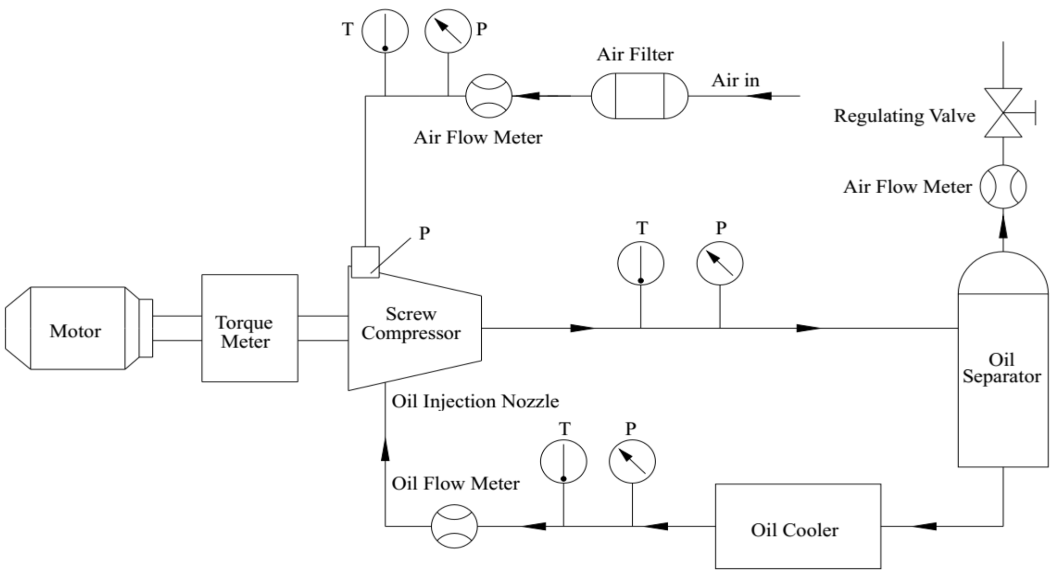
$$\eta_v = \frac{m_s - m_{gt}}{m_s} \tag{30}$$

3. Experimental studies

3.1. Test rig

The mathematical model must be verified before it being used for performance prediction. In this case, the test to study the performance of the oil injected twin screw air compressor under unload conditions is carried out. A basic piping and instrumentation diagram of the test rig with twin screw compressor is shown in Fig. 2. The air was taken into working chambers through the air filter and the oil was injected into working chamber through the injection nozzle with the rotation of the main rotor. The oil/air fluid was compressed in the compressor and discharged into an oil/air separator. From there, the separated air was cooled further and flowed to the network. The separated oil was flowed out of the separator through the cooler and then reinjected into the compressor duo to the pressure difference between the pressure in the compressor working chamber and the discharge pressure.

The compressor was driven by a variable speed electrical motor, with its speed controlled by a frequency converter. The rotational speed and shaft power consumption of compressor were measured using a torque meter which was placed between the compressor and motor. The ambient pressure, suction pressure, discharge pressure of air and pressure of injected oil were measured by pressure transducers. The temperature of ambient air, discharged oil/air fluid temperature and oil temperature before injecting into the working chamber were measured by temperature transducers. The volumetric flow rate of inlet air and discharge air were measured by turbine flow meters. The flow rate of oil was measured by oil flow meter.



P – Pressure sensor T – Temperature sensor

Figure 2. The oil injected screw compressor test system

Table 1 presents the main parameters of twin screw air compressor for test. The clearance value of the leakage paths is listed in Table 2.

Table 1. Main parameters of the twin screw air compressor

Parameter	Value
Number of male rotor lobes	5
Number of female rotor lobes	6
Outer diameter of the male rotor /mm	128
Outer diameter of the female rotor /mm	101
Rotor length /mm	175
Wrap angleof male rotor / °	300
Built in volume ratio	6
Rotation speed of the male rotor /rpm	3000
Supplied oil temperature /K	306

Table 2. Clearances at different leakage paths

Leakage paths	Value/area
The contact line /mm	0.027
The blow hole /mm^2	2
The rotor tip /mm	0.03
The outlet end face /mm	0.03
The inlet end face /mm	0.4

3.2. Model validation

Fig. 3 presents the comparisons of the measured and calculated shaft power consumption of twin screw compressor under unload conditions with various discharge pressure, where the suction pressure, the rotational speed are kept constant at 0.083MPa, 2000 rev/min and kept constant at 0.061MPa, 3000 rev/min respectively. It can be seen that the power consumption increases from 2.43kW to 3.23kW when the discharge pressure rise from 0.21MPa to 0.35MPa, where rotational speed is at 2000 rev/min. It means that there is more shaft power consumption of the compressor at higher discharge pressure where the rotational speed is kept constant. This is due to the fact that more shaft power is required to deliver per unit volume of air at higher discharge pressure. As the rotational speed increases, the shaft power increases. This may be attributed to the fact that more quality of compressed air quality is delivered in per unit time as the rotational speed increases.

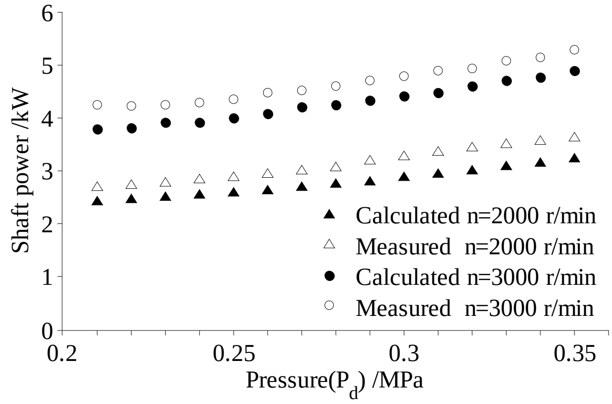


Figure 3. Shaft power for various discharge pressure

The measured and calculated flow rate of screw compressor with various discharge pressure are shown in Fig. 4. And the suction pressure, the rotational speed are kept constant at 0.083MPa, 2000 rev/min and 0.061MPa, 3000 rev/min respectively. As indicated in Fig. 4, the flow rate decrease as the discharge pressure rises, but the decrement rate is bigger and bigger. This is because of the working chamber is connecting with discharge port at the end of compression stage, and the discharge pressure causes the gas leak to the working chamber quickly. The gas leakage is more at higher discharge pressures, so that the flow rate drops. On the other hand, it is observed that flow rate increase at higher rotational speed. The reason for this is that the theoretical volume of working chambers keeps constant, so the theoretical displacement per revolution of the compressor keeps constant. Because of leakage and other factors, the increase rate for flow rate is smaller and smaller.

As shown in Figs 3 and 4, the errors between the calculated and measured shaft power are less than 3.2%. In addition, the actual flow rate of screw air compressor agrees closely with the calculated flow rate and the errors are less than 3%.The results show good agreement between the predicted and measured values. The difference in experimental data and theoretical values is attributed to wall friction and other frictional losses.

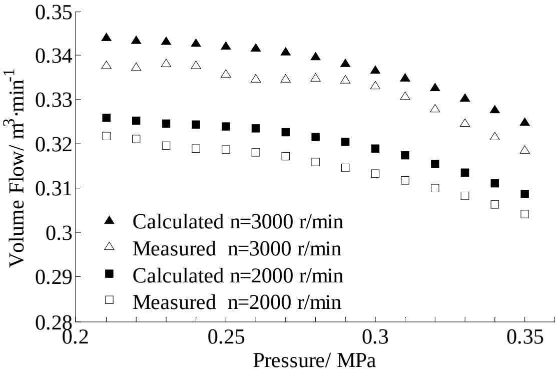


Figure 4. Variation of gas flow rate with different discharge pressure

4. Discussion

The influence of discharge pressure on the P–V diagram is shown in Fig. 5. The compressor keeps a constant rotational speed of 2000 rev/min and a constant suction pressure of 0.083MPa. As seen from Fig. 5, with increased discharge pressure, pressure in the working chamber is nearly the same over a wide range in the suction process and the beginning of compression process. This happens because the discharge pressure has little effect on these stages. But when close to the discharge process, it is observed that pressure in the working chamber with discharge pressure is at 0.3MPa, slightly higher than that with discharge pressure, which is at 0.2MPa. This is attributed to the higher discharge pressure.

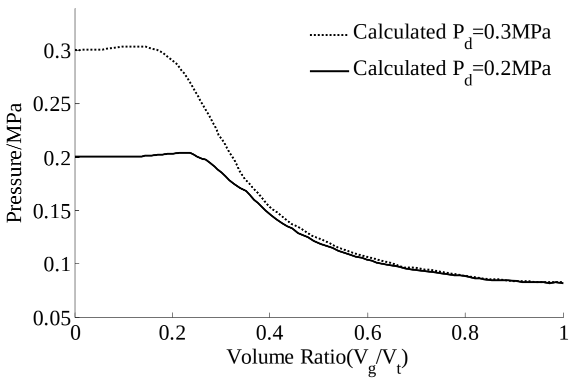


Figure 5. P-V indicator diagram for various discharge pressure of screw compressor

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Fig.6 shows the influence of suction pressure on the P–V diagram, where the rotational speed and the discharge pressure are kept constant at 2000 rev/min, 0.3 MPa respectively. It can be seen that the area enveloped by P-V diagram is smaller with the decrease of suction pressure. It means that the compressor's shaft power decreases with the reduction of suction pressure. As the decrease of suction pressure, the pressure in the working chamber changes faster during the discharge process. This is because the suction pressure is lower, pressure inside the working chamber is also lower, so the leakage between working chamber and discharge chamber is increased with the same discharge pressure.

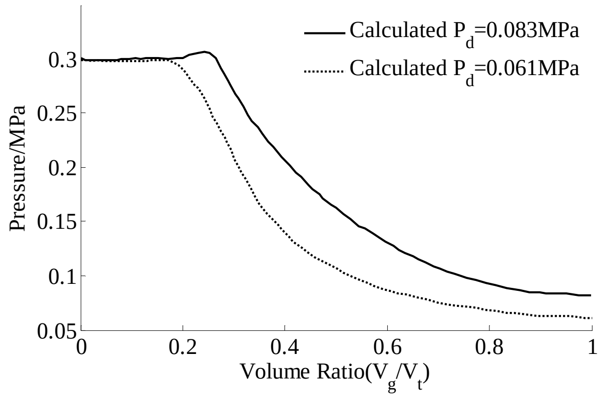

Figure 6. P-V indicator diagram for various suction pressure of screw compressor

The influence of discharge pressure on the volumetric efficiency at 0.083MPa, 2000 rev/min and 0.061MPa, 3000 rev/min is shown in Fig.7. It is observed that the volumetric efficiency of screw compressor decreases at higher discharge pressure. It's owing to the fact that the gas leakage rate depends on the pressure difference during compression process. If the discharge pressure increases, the pressure difference between suction and discharge pressure increases, so the volumetric efficiency slightly decreases. What's more, the volumetric efficiency increase at higher rpm can be seen in figure. Because the time for leakage will be shortened at higher rpm, and the amount of leakage will be reduced.

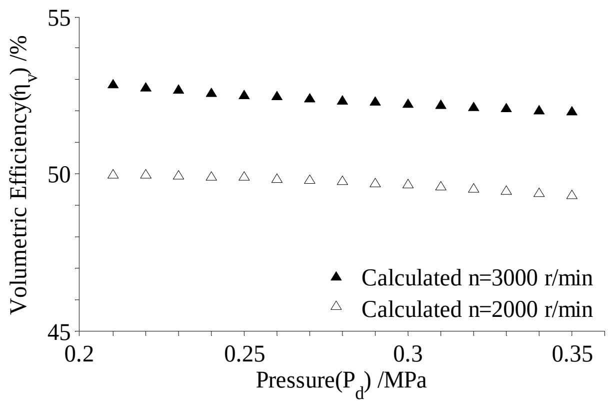

Figure 7. Volumetric efficiency versus differential discharge pressure

The percentage of shaft power consumption under the unload conditions and full load conditions at differential pressure is shown in Fig.8, where the suction pressure, the rotational speed are kept constant at 0.083MPa, 2000 rev/min and kept constant at 0.061MPa, 3000 rev/min respectively. Under the full load conditions, the compressor keeps a constant discharge pressure of 0.7MPa and a constant suction pressure of 0.1MPa. As indicated in Fig.8, as the discharge pressure rise, the percentage of shaft power consumption under unload conditions increases at the constant rotational speed. Moreover, the percentage of shaft power consumption increases with reduction of rotational speed when the discharge pressure is kept constant. In order to obtain a lower percentage of shaft power consumption under unload conditions, the discharge pressure should be as low as possible in a proper range. Because the flow rate of oil for circulating may not enough with a too low discharge pressure.

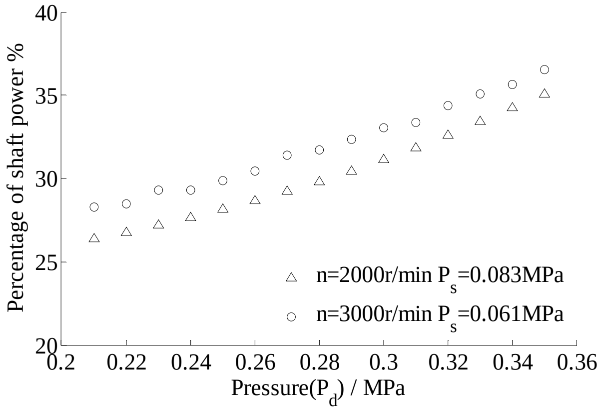

Figure 8. The percentage of power consumption versus differential discharge pressure

5. Conclusion

After the analysis of the working process of twin screw air compressor under load/unload conditions, a mathematical model of screw air compressor has been developed to study the influence of some operating and design parameters on the compressor performance under unload conditions. Experimental study on the performance of a screw air compressor under unload conditions has been carried out to verify the model. The calculated results show a reasonable agreement with the data measured by the experiment. The errors between the model calculating data and experimental data are less than 3% for the volumetric flow rate and less than 3.2% for the shaft power under unload condition. The model can predict the performance of twin screw air compressor under unload conditions accurately and it is helpful for the compressor design and optimization.

The theoretical and experimental results show that the rotational speed and the discharge pressure have a direct influence on the performance of screw air compressor under unload conditions. As the discharge pressure decreases, the volumetric efficiency will improve and the flow rate will increase. As the increase of the rotational speed, the flow rate and volumetric efficiency will be greater. Wherein, influence of rotational speed on the volumetric efficiency is more sensitive than that of discharge pressure. The shaft power consumption of the compressor will be less with the decrease of discharge pressure and rotational speed.

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Nomenclature

A	heat transfer area
A_c	leakage clearance area
A_{in}	cross sectional area of the piping leading to the valve
C	flow coefficient
c_l	specific heat of lubricating oil
c_p	specific heat of gas at constant pressure
g	gravitational constant
h	heat transfer coefficient between gas and oil
H_g	enthalpy of gas
K_v	resistance coefficient
k	ratio of specific heats
M_g	mass of gas in the working chamber
M_{il}	interlobe leakage mass of air that leaks back to the suction cavity
M_l	mass of oil in the working chamber
M_s	theoretical mass of inducted gas
M_{s1}	mass of gas in the working chamber at the end of suction process
m	leakage mass flow rate through flow path
m_s	mass flow rate of air through the inlet valve
m_{gt}	total mass leakage rate of gas
m_{gt}	total mass leakage rate of gas
N_{fr}	friction power loss
N_{oil}	power loss by oil injected
N_s	shaft power
N_t	theoretical indicated power
n	rotational speed
P	pressure in the workingchamber
P_{atm}	ambient pressure
P_c	pressure at the end of the compression process
P_d	discharge pressure
P_{inj}	pressure of gas at the injection position
P_s	suction pressure
Q_c	heat transferred between gas and casing
Q_l	heat transferred between gas and oil
$\dot{Q}_{in}$	volumetric flow rate immediately before the valve
$\dot{Q}_{oil}$	volumetric flow rate of injected oil
$\dot{Q}_s$	volumetric flow rate immediately after the valve
R	specific gas constant
S	slip factor
T_e	gas temperature at the end of suction process
T_g	gas temperature going in to the working chamber
T_{in}	gas temperature immediately before the valve
T_l	temperature of oil in the working space
T_{oil}	mean temperature of leaked oil in suction process
T_s	inlet gas temperature
t_r	time required for suction process
U_g	internal energy of gas
V	volume of the working chamber
V_d	volume of rotor chamber at the end of the compression process
V_g	volume of gas in the working chamber
V_t	geometrical volume of working chamber
v	velocity of gas or oil
v_{in}	velocity immediately before the inlet valve
W_t	thermodynamic gas work

Greek symbols

α	void fraction coefficient
β	coefficient for mechanical loss
χ	ratio of gas in the gas-oil mixture
ε	volume ratio
φ	main rotor angular position
η_v	volumetric efficiency
μ	dynamic viscosity of oil
ρ_g	density of gas
ρ_l	density of oil
ω	angular rotational speed

Subscripts

i	gas or oil going into the working chamber
o	gas or oil going out of the working chamber
g	gas
l	oil