1990

Research on Discharge Ports of Oil-Flooded Rotary Compressors

L. Hongqi
Xi'an Jiaotong University

J. Guangxi
Xi'an Jiaotong University

RESEARCH ON DISCHARGE PORTS OF OIL-FLOODED ROTARY COMPRESSORS

Li Hongqi. Jin Guangxi
Xi'an JiaoTong University
People's Republic of China

ABSTRACT

In the discharge process of a rotary compressor, the resistance of gas-oil two-phase flow through the discharge port can't be ignored.especially with oil-injected. Therefore, it's always an important problem in this kind of compressor design that the discharge port should be determined reasonably. In this paper, taking an oil-flooded single screw compressor as an example, developing a computer model, the effect of discharge port on the performance of an oil-flooded rotary compressor has been analyzed in detail, also, the optimal outlet position under various working conditions have been presented.They have been proved by test in the end of the paper.

INTRODUCTION

In a rotary compressor, gas discharge is generally coercive, which is very different from a piston-compressor. The time of discharge beginning is only determined by built-in volume ratio, and has nothing to do with cavity pressure.When outlet position isn't designed properly, there will be extra energy losses due to over or less compression. The position of outlet port determines the beginning of discharge process, usually it's designed by the principle -- built-in pressure ratio should be equal to that of outer one's. However, this is only suitable for ideal condition.The discharge port resistance is neglected. In fact, the compression chamber pressure is higher than discharge pressure, then the port position designed by that principle isn't optimal. In this paper, we take an oil-flooded single screw compressor as an example. By applying variation mass thermodynamic theory and the basic two-phase flow tenet to the computer model of working process and optimal design, we examined discharge process in detail. The oil-gas two-phase flow through discharge port was analyzed, and optimal positions of outlets were presented under various conditions. This has been proved by test. In short, this paper is attempted to provide some valuable references to the design of discharge port.

BASIC THEORY

During the process of discharge, the groove pressure was affected by two factors. one is the pressure drop due to the diffluence of gas and oil, the other is the pressure rising due to the convergence of control volume. The real pressure is a result of these two aspect's interaction.

According to the first law of thermodynamics and equation of state for ideal gas, we can deduce the variation rate of pressure:

$$\frac{dp}{d\theta_2} = -KP\left\{\frac{1}{v}\frac{dv}{d\theta_2} + \frac{1}{\omega_2 m_g}\left[\frac{dm_{go}}{dt} - \frac{dm_{gi}}{dt} + \frac{dm_{gl}}{dt}\right]\right\} \qquad (1)$$

where

θ_2--gaterotor rotation angle.

v--control volume

ω_2--the angle speed of gaterotor

m_g--mass of gas

dm_{go}/dt--discharge mass flow rate of gas

dm_{gi}/dt--back flow rate of gas

dm_{gl}/dt--leakage of gas

k--adiabatic exponent

p--groove pressure

The flow through discharge port is regarded as two-phase flow through an orifice[1].

$$\frac{dm_{go}}{dt} = \frac{\alpha A \cdot x \cdot \sqrt{2\rho_l \Delta p_{TP}}}{(1-x)\theta_0 + x \cdot \sqrt{\rho_l/\rho_g}} \qquad (2)$$

where

α--flow coefficient

A---area of discharge port

ρ--oil density

ΔP_{TP}--pressure drop of two-phase flow through an orifice

x--- dryness

ρ_g---gas density

θ_0---correction coefficient

The leakage of gas is rather less than exclusion gas during outlet stage. Therefore, the effect of leakage can be neglected.

According to the above equations, we can obtain the relationship between thermodynamic parameters in control volume and the variation angle of gaterotor, also the power consumption and efficiencies of compressor. For a given working condition, we set the power consumption as an objective function, optimizing the gaterotor angle at the beginning of discharge process, then the optimal outlet position is obtained.

TEST INSTALLATIONS

An air compressor with revolving ring was used in our test to examine the theoretical calculation model. By shifting the revolving ring to change the outlet position, we measured the optimal setting for various working conditions. Fig.1 is the experimental system.

where

1 --- D.C. machine

2 --- single screw compressor

3 --- gas tank, oil seperator

4 --- oil cooler

5 --- sensor of torque and rotation speed

6 --- display panel of torque and rotation speed

7 --- filter

8 --- turbo flow indicator

11.12 --- turbo flow meter

13 --- oil pressure gauge

15 --- testing point for oil temperature

16 --- oil pump

20. 21 --- testing points for temperature and pressure

22 --- exhauster

23 --- nozzle

24. 25 --- testing points for temperature and pressure

27 --- cooling water

9.10.14,17,18.19,26 --- valves

RESULTS ANALYSIS

At the beginning of discharge. $dv/(v d\theta_2)$ is the predominant factor, since αA is small and flow resistance is strong, then outgoing gas dm_{go}/dt is minor. From equation(1), it is known that $dp/d\theta_2 > 0$, the groove pressure will be rising continuously. Only when αA is big enough, does the pressure decrease to backpressure gradually. Fig.2 shows the theoretical value for the discharge pressure in groove, when the discharge port was designed by the principle of equipollent pressure ratio, oil injection and backpressure are kept constant. It may be expected that immediately at the opening of the outlet port a pressure increase will occur, hence, the discharge process should be brought forward to reduce the extra energy losses when designing. All the optimal outlet positions offered in the paper are expressed by $\Delta\theta_2 (= \theta_{2d} - \theta_{2d'})$ which is the early opening angle as compared with that when inner and outer pressure ratio is equal. There, θ_{2d} is the gaterotor angle when the pressure ratios are equal, and $\theta_{2d'}$ is the optimal.

The rotor revolution N1 affects discharge process strongly. ω_2 is inversely proportional to displacement according to the equation(1). When ω_2 rises, the over compression in discharge process becomes more serious (Fig.2). To reduce the extra energy losses, $\Delta\theta_2$ should be larger. Fig.4 and Fig. 5 show the variations of $\Delta\theta_2$ and N1 according to different oil injection and outer pressure ratio. The curves are theoretical values, and the points are the results of test. Known from the figures, the higher N1, the larger $\Delta\theta_2$. And when N1 tends to zero, $\Delta\theta_2$ tends to zero. It is just at this situation that the position designed by equipollent pressure ratio principle would be ideal.

The oil-injection effect on discharge process is indicated by the dryness of two-phase flow. Transform equation (2) as follow:

$$\frac{dm_{go}}{dt} = \frac{\alpha A \cdot \sqrt{2 \rho_l \Delta p_{TP}}}{(1/x - 1)\theta_0 + \sqrt{\rho_l / \rho_g}} \qquad (3)$$

and $x = m_g / (m_g + m_l)$

where m_l is the mass of oil.

Known from the above, the more oil mass flow rate, the less dm_{go}/dt, and the over compression in discharge process is more serious. Thereby, the larger $\Delta\theta_2$ should be. Fig.3 shows the calculation results of pressure change in groove with certain rotation speed, outer pressure ratio and various oil mass flow rate. Seeing from Fig.5, under the condition of 3000 r.p.m. and

3.316 outer pressure ratio(ε_p), $\Delta\theta_2$ is 1.65° while oil mass flow rate is $4\times10^{-3}m^3/min$, and when oil mass flow rate is $20\times10^{-3}m^3/min$, $\Delta\theta_2$ is 1.99°.

The back pressure Pd also affects discharge process somewhat. Shown clearly in Fig.4 and Fig.5, the higher Pd, the larger $\Delta\theta_2$. Keep $4\times10^{-3}m^3/min$ oil flow rate and 3000 r.p.m. constant, then, $\varepsilon_p=5.429, \Delta\theta_2=1.80°$; $\varepsilon_p=3.316, \Delta\theta_2=1.65°$.

CONCLUSIONS

1. Been existing the flow resistance during discharge process, it is not reasonable to design outlet position using equipollent pressure ratio principle. The discharge should be brought forward to decrease the extra energy losses.

2. Factors which have primary effect on discharge process are rotative speed, oil-injection, designed back pressure and gas leakage, etc.

3. The over compression in discharge process is rising with the increase of rotative speed, the higher rotative speed, the larger $\Delta\theta_2$ should be.

4. Oil mass flow rate affects the dryness of two-phase flow, influencing discharge process indirectly. The more oil mass flow rate, the larger $\Delta\theta_2$.

5. $dv/(vd\theta_2)$ is influenced by designed back pressure, higher back pressure leading to larger $\Delta\theta_2$.

REFERENCES

1. 林宗虎，《气、液、固多相流测量》，中国计量出版社，1988。
2. Bein. T.W., and Hamilton, J.F."Computer Modeling of an Oil-Flooded Single Screw Air Compressor",ICECP,1982

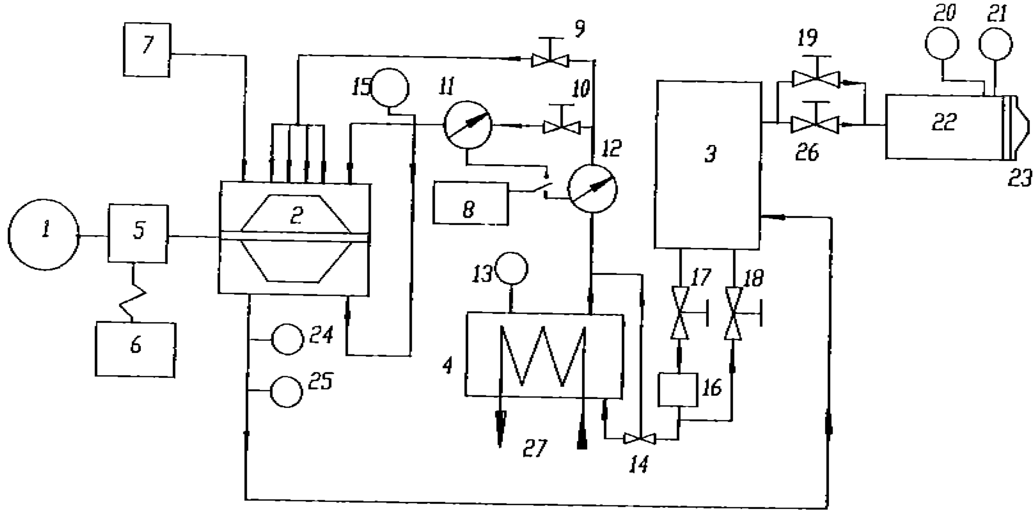
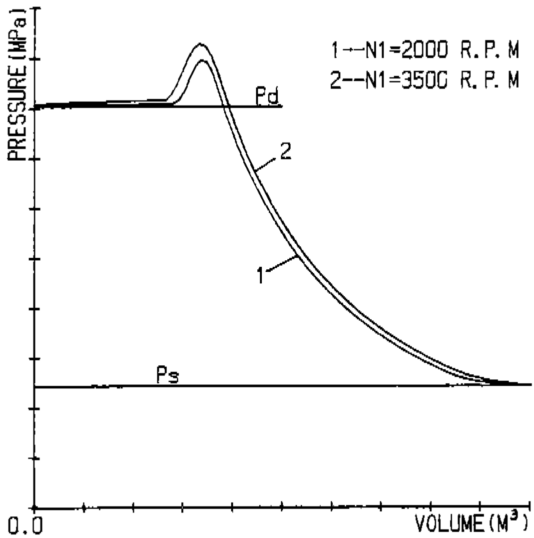
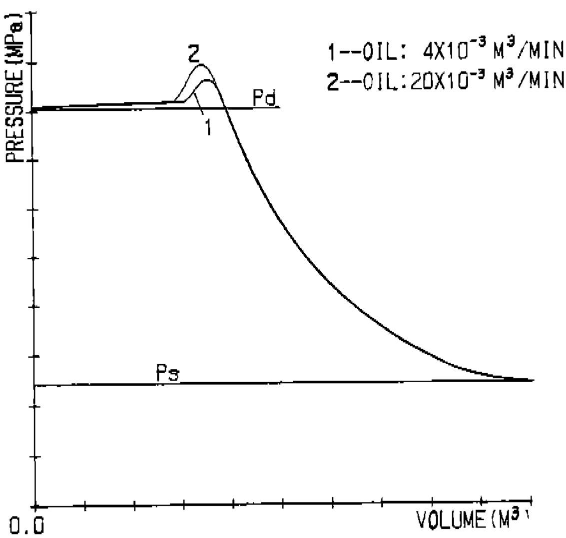


Fig. 1 EXPERIMENTAL SYSTEM



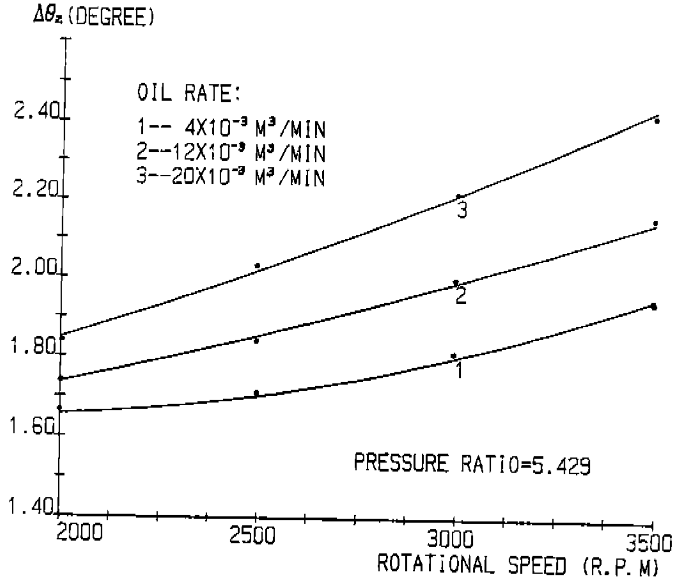
PRESSURE (MPa)
1--N1=2000 R.P.M
2--N1=3500 R.P.M
Pd
2
1
Ps
0.0
VOLUME (M³)

Fig. 2



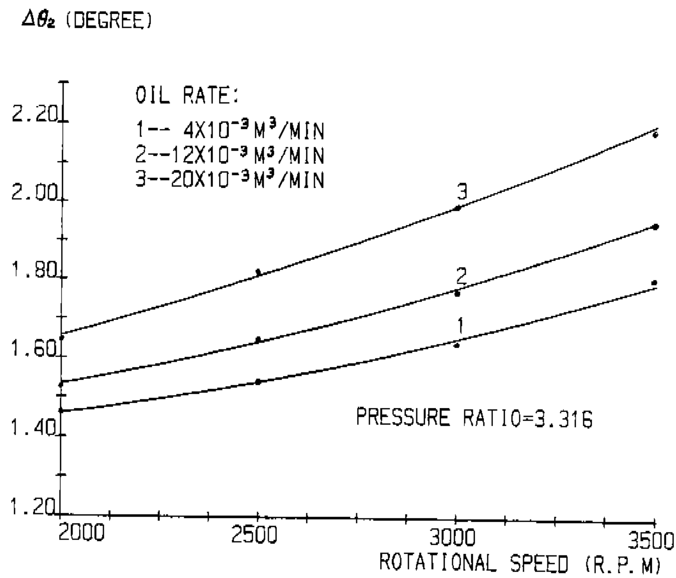
PRESSURE (MPa)
1--OIL: 4X10⁻³ M³/MIN
2--OIL:20X10⁻³ M³/MIN
2
Pd
1
Ps
0.0
VOLUME (M³)

Fig. 3



Δθ₂ (DEGREE)
OIL RATE:
1-- 4X10⁻³ M³/MIN
2--12X10⁻³ M³/MIN
3--20X10⁻³ M³/MIN
2.40
2.20
2.00
1.80
1.60
1.40
3
2
1
PRESSURE RATIO=5.429
2000
2500
3000
3500
ROTATIONAL SPEED (R.P.M)

Fig. 4



Δθ₂ (DEGREE)
OIL RATE:
1-- 4X10⁻³ M³/MIN
2--12X10⁻³ M³/MIN
3--20X10⁻³ M³/MIN
2.20
2.00
1.80
1.60
1.40
1.20
3
2
1
PRESSURE RATIO=3.316
2000
2500
3000
3500
ROTATIONAL SPEED (R.P.M)

Fig. 5