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Back-Pressure Mechanism of Scroll Compressor

Z. Qian
Xian Jiaotong University

Z. Zhang
General Machinery Research Institute; P. R. China

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Qian Zhongliang
Xi'an Jiaotong University
Xi'an, Shaanxi, China

Zhang Zhiheng
General Machinery Research Institute
Anhui, China

ABSTRACT

On the principle of changeable mass thermodynamics, a thermodynamic model of working process for scroll compressor with self-adjusting back-pressure mechanism is established in this paper. General motion law of back-pressure ports position is developed by solving thermodynamic model of compressor and carrying out dynamical computation. In this method, the guarantee of higher volumetric and mechanical efficiency is regarded as the confined condition.

SYMBOLS

b: polar radius of the centre of the backpressure port, m

β: polar angle of the centre of the backpressure port, (°)

r_o: eccentricity of the crank, m

θ: angle of rotation (°)

e: radius of the backpressure port, m

$S(\theta)$: connected area between the backpressure chamber and compression chamber, m^2

m: mass of the gas in the compression chamber, kg

u: specific internal energy of the gas in the compression chamber, kJ/kg

h_i: specific enthalpy of the gas flowing into the compression chamber, kJ/kg

h_o: specific enthalphy of the gas flowing out of the compression chamber, kJ/kg

ω: angular velocity, rad/s

dm_i: mass of the gas microelement flowing into the compressor chamber, kg

dm_o: mass of the gas microelement flowing out of the compression chamber, kg

dm_b: mass of the gas microelement flowing between the backpressure chamber and compression chamber, kg

T: temperature of the gas in the compression chamber, K

v: specific volume of the gas in the compression chamber, m^3/kg

P: pressure of the gas in the compression chamber, (10^5Pa)

P_b: pressure of the gas in the backpressure chamber (10^5Pa)

a: radius of basic circle for involute, m

φ_A: expanding angle of point A on the Wrap, rad

d: starting angle of the involute, rad

b_o: polar radius of the backpressure port tangential to the Wrap

δ: increment of b_o, which is chosen according to the machining requirements of backpressure port, m

ψ: included angle between an arbitrary point on the circule of the backpressure port and the horizontal axis, rad

INTRODUCTION

One of the key technologies of the scroll compressor is the appropriate balance of the axial force, which not only gurantees that the mating scrolls do not separate in the compression and prevents gas from radial leakage, but also ensures that the acting force on the contact surfaces of the mating scrolls is not too great, and prevents the mechanical officiency decreasing. Currently, there are mainly three methods to balance the axial gas force, i.e., exerting an axial bearing counter-force, a spring force or a backpressure gas force on the back of the orbiting scroll. The last method can not only compensate automatically the wear between the mating scrolls, but also adjust the pressure of the backpressure gas whenever the working condition changes. Therefore it has found wide application in scroll compressors.

When the third method is adopted to balance the axial gas force, the central compression chamber and backpressure chamber are connected by the backpressure port on the orbiting scroll, and the backpressure of the gas in the backpressure chamber depends wholly on the position and geometrical dimensions of the port. This paper geometrically studies the orbit of the backpressure port relative to the static scroll and the law of the flow area change of the port; it finds out the relationship between the pressure in the backpressure chamber and the position and flow area of the backpressure port by applying the basic law of thermodynamics and the principle of mass conservation; it also carries out dynamic computation to determine the optimal position and geometrical dimensions of the backpressure port with the confining condition of enough volumetric and mechanical effeiency of the compressor.

LAW OF MOTION OF THE BACKPRESSUR PORT AND CHANGE OF THE FLOW AREA

Mass exchange of the actuating medium between the compression chamber and backpressure chamber of the scroll compressor is carried out through the backpressure port. Mass flow rate between the two chambers is mainly determined by the pressure ratio $\tau_i(\theta)$ and flow area $S(\theta)$, and the pressure ratio $\tau_i(\theta)$ between the two chambers is connected closely with the flow area $S(\theta)$ and the position and diameter of the backpressure port. To learn about the pressure change in the backpressure chamber when the compressor is working, we must find out the orbit of the backpressure port relative to the static scroll and solve for the law of the flow area of the backpressure port changing with the orbiting angle.

Usually, two backpressure ports are made in the end plate of the orbiting scroll in the scroll compressors with self-adjusting backpressure mechanism, which have a diameter of 2 mm and are 180° out of phase. During the operation of the orbiting scroll, the motion of the backpressure port's center relative to the static scroll is governed by the same law as the relative motion of an arbitrary point on the orbiting scroll. The motion orbit is shown in Fig. 1.

The coordinates of the port center on the orbiting scroll are $(b_{OI}\cos\beta, -b_{OI}\sin\beta)$ and $(-b_{OI}\cos\beta, b_{OI}\sin\beta)$ respectively. Now examine the port in the fourth quadrant. Projected to the static scroll, the orbit of the port center is a circle whose center is at $(b_{OI}\cos\beta, -b_{OI}\sin\beta)$ and radius equals the eccentricity of the crank r_o. Thus, the equation is:

$$
\begin{aligned}
x &= b_{OI}\cos\beta + r_o\cos\theta \\
y &= -b_{OI}\sin\beta - r_o\sin\theta
\end{aligned}
\tag{1}
$$

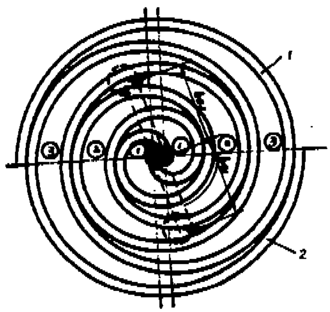


Fig. 1 Orbit of the center of the backpressure port
1.orbiting scroll 2.static croll

In addition, the orbit of an arbitrary point on the boundary circle of the port is also a circle of the same radius r_o, and only the center of the circle is in a different position, as shown in Fig. 2. The equation of its orbit is,

$$x = b_{OI}\cos\beta + r_0\cos\theta + e\cos$$
$$y = -b_{OI}\sin\beta - r_0\sin\theta - e\sin \qquad (2)$$

During the operation of the orbiting scroll, the backpressure port may be covered partially or totally by the wall of the Wrap on the static scroll (See the dash area in Fig. 2), which may result in the change in the flow area $S(\theta)$ between the backpressure chamber and compression chamber. It may be acquired by solving the insolute equation of the static scroll and Eq.(2) simultaneously.

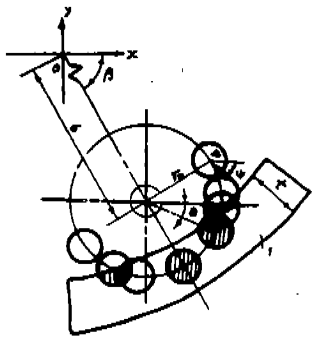


Fig. 2 Orbit of the backpressure port and law of the flow arta change

1. static scroll

Assume the port radius e and chamber volume V_d constant. The pressure in the backpressure chamber is mainly determiined by the position of the port. In a motion period, the port does not move in one compression chamber only, which is determined by the characteristic of the backpressure chamber for balancing the axial force. If the backpressure port moves in the first compression chamber or the second compression chamber, the gas pressure in the backpressure chamber, $P_b(\theta)$ has a range of $P_2(\theta) < P_b(\theta) < P_d$; and such a high backpressure would increase the contact wear between the end surfaces of the mating scrolls, and fail to ensure a high mechanical efficiency. If the port moves in the outermost compression chamber, then $P_3(\theta) < P_b(\theta) < P_2(\theta)$; and such a low backpressure could not ensure the fit of the orbiting scroll to static scroll in operation and could hardly ensure the volumetric efficiency. Therefore, the backpressure port can only move in the second or third compression chambers. Thus, with β determined, selection of b should satisfy the followin conditions:

A. When $0 \leqslant \theta \leqslant \beta$, the port is in the second compression chamber, or is covered by the static scroll

B. When $\beta \leqslant \theta \leqslant 2\pi$ the port is in the third compression chamber, or is covered by the static scroll

The position of the backpressure port in the compression chamber with different angle of rotation θ is shown in Fig. 3.

It can be seen from Fig. 3 that given β, b is

$$b = b_o + \delta$$

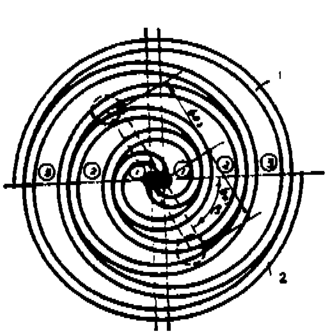
where $b_o = \sqrt{1 + \varphi_A^2} - r_o + e$

and can be gained by

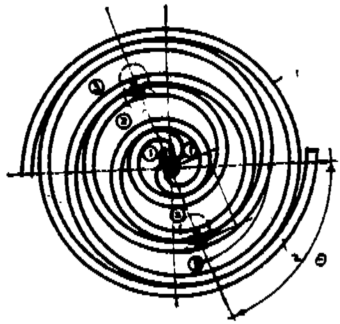
$$tg\,B = \frac{\varphi_A(\cos(\varphi_A + \alpha) - \sin(\varphi_A + \alpha)}{{}_A\sin(\varphi_A + \alpha) + \cos(\varphi_A + \alpha)}$$

ANALYSIS OF THE THERMODYNAMIC PROCESS

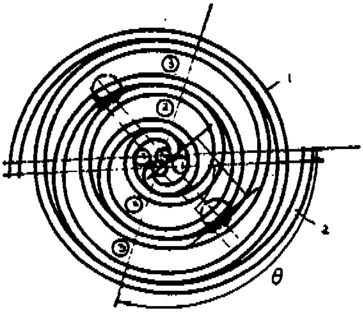
In order to determine the optimal position and geometric dimensions of the backpressure port, it is necessary to find out the pressure of the gas in the compression and backpressure chambers and law of its change, and analyze the axial force and overturning moment on the orbiting scroll. Therefore the thermodynamic model of the chambers must be established and solved.



a) Position of the port when $\theta < \beta$



b) Position of the port when $\theta = \beta$



c) Position of the port when $\theta > B$

Fig. 3 Analytical gragh of the relationship between b and β

1. orbiting scroll
2. static scroll

1. Thermodynamic Model of the Compression Chamber

The compression chamber connected with the backpressure chamber is taken as the control volume, the connected system is shown in Fig. 4. Assume that the volumes of the suction and discharge chambers are indefinitely great, and gas flow in the suction, discharge and leakage process and mass exchange between the two chambers is all steady flow, and ignore the effect of the lubricating oil on the performance of the actuating medium and potential energy and kinetic energy of the medium, then, we have the following equation by applying the first law of thermodynamics:

$$d(mu) = dQ + dQ_b^* - h^* dm_b + h_i dm_i - h_o dm_o - dw \tag{3}$$

where $d\theta_b^*$ is the heat transfer microelement between the compression chamber and backpressure chamber.

dQ is the heat transfer microelement between the compression chamber and the outside system.

h^* is the specific enthalpy of the gas flowing between the two chambers.

dm_b is positive, and $h^* = h$; when $P(\theta) > P_b(\theta)$,

dm_b is negative, and $h^* = h_b$. when $P(\theta) < P_b(\theta)$,

dw is the compression microelement work.

$$dw = - PdV_c ,$$

and V_c is the volume of the compression chamber.

When the rotating velocity of the crank is constant, $\theta = \omega\tau$, $d\theta = \omega d\tau$. Hence Eq.(3) can also be expressed as the following differential form of θ.

$$m \frac{du}{d\theta} + u \frac{dm}{d\theta} = \frac{dQ}{d\theta} + \frac{dQ_b^*}{d\theta} - h^* \frac{dm_b}{d\theta} + h_i \frac{dm_i}{d\theta} - h_o \frac{dm_o}{d\theta} - P \frac{dV_c}{d\theta} \tag{4}$$

Apply the total derivative relationship:
$\frac{du}{d\theta} = (\frac{\partial u}{\partial T})_v \frac{dT}{d\theta} + (\frac{\partial u}{\partial v})_T \frac{dv}{d\theta}$ and $h_o = h$. Then Eq.(4) can be converted into:

$$\frac{dT}{d\theta} = \Big[\frac{dQ}{d\theta} + \frac{dQ_h^*}{d\theta} - h^*\frac{dm_b}{d\theta} + h_i\frac{dm_i}{d\theta} - h\,\frac{dm_o}{d\theta} - P\frac{dv_c}{d\theta} - u\frac{dm}{d\theta} - m\,(\frac{\partial u}{\partial v})_T\,\frac{dv}{d\theta}\Big] / \Big(m(\frac{\partial u}{\partial T})_v\Big) \qquad (5)$$

where $\frac{dv}{d\theta} = d(v_c/m)/d\theta = \frac{1}{m}\frac{dv_c}{d\theta} - \frac{v_c}{m^2}\cdot\frac{dm}{d\theta}$ (5)

$$\frac{dm}{d\theta} = \frac{dm_i}{d\theta} - \frac{dm_o}{d\theta} - \frac{dm_b}{d\theta}$$

and $m = m_{(o)} + \int_0^\theta (\frac{dm}{d\theta})\,d\theta$

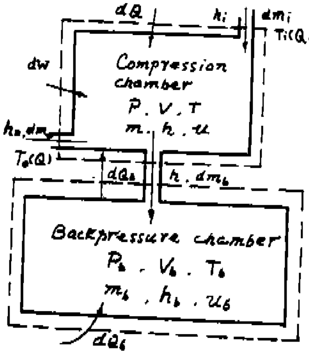


Fig. 4 Thermodynamic model

The above Eq.(5) is thus the basic equation of the thermodynamic model of the compression chamber.

2. Thermodynamic Model of the Backpressure Chamber

Here the backpressure chamber indicated in Fig. 4 is taken as the control volume for analysing the thermodynamic process of the backpressure chamber. Apply the first law of thermodynamics. Then,

$$d\,(m_b u_b) = dQ_b - \Sigma\, dQ_b^* + h^* dm_b \qquad (6)$$

where dQ_b is the heat transfer microelement between the backpressure chamber and system. When the chamber absorbs heat, dQ_b is positive; otherwise, it is negative.

Apply the total derivative relationship. Then Eq.(6) can be written

$$\frac{dT_b}{d\theta} = \Big(\frac{dQ_b}{d\theta} - \Sigma\frac{dQ_b^*}{d\theta} + h^*\,\frac{dm_b}{d\theta} - u_b\,\frac{dm_b}{d\theta} - m_b\,(\frac{\partial u_b}{\partial v_b})_{T_b}\,\frac{dv_b}{d\theta}\Big) / \Big[m_b\,(\frac{\partial u_b}{\partial T_b})_{v_b}\Big] \qquad (7)$$

where $\frac{dv_b}{d\theta} = \frac{d(v_d/m_b)}{d\theta} = \frac{-v_d}{m_b^2}\,\frac{dm_b}{d\theta}$;

$$m_b = m_b(o) + \int_0^\theta \frac{dm_b}{d\theta}d\theta$$

where v_d is the volume of the backpressure chamber.

Ignore the effect of oil deposition on the volume, then

$$v_d = C$$

Eq. (7) is thus the basic equation of the thermodynamic model of the backpressure chamber.

The scroll compressor is characterized with the continuous multi-chamber compression, and the parameters of the gas condition in the backpressure chamber are in periodical variation. Therefore the compression process must satisfy the following convergence conditions:

$$\begin{aligned} m_3(360^\circ) &= m_2\,(0^\circ) \\ T_3(360^\circ) &= T_2\,(0^\circ) \\ P_b(0^\circ) &= P_b(360^\circ) \\ T_b(0^\circ) &= T_b(360^\circ) \end{aligned} \qquad (8)$$

where $m_3 . T_3$ are the mass and temperature of the gas in the third compression chamber respectively. m_2 and T_2 are the mass and temperature of the gas in the second compression chamber respectively.

Combining the equations (5), (7) and (8), we obtain the basic equations of the thermodynamic model of the scroll compressor with self-adjusting backpressure mechanism. Solving them, we can find out the pressure of the gas in the compression and backpressure chambers and law of its change.

DETERMINATION OF THE OPTIMAL POSITION OF THE BACKPRESSURE PORT

1. Minium Backpressure P_{bmin} to Ensure the Mating Scrolls do not Separate

Prerequisites of determination of the port position is the calculation of the minimum backpressure P_{bmin} which ensure the mating scrolls do not separate in the operation, to test whether the parameters of the position (b,β) meet the requirements or not. Therefore the axial gas force F_t and overturning moment exerted on the orbiting scroll should be solved for from the pressure distribution in the compression chamber obtained by solving the above thermodynamic model. Then P_{bmin} can be obtained from the balance of the axial forces on the orbiting scroll.

The pressure distribution on orbiting scroll and self-adjusting backpressure mechanism is shown in Fig. 5. Since the contact surfaces of the mating scrolls are not very wide, we can consider it to be linear distribution of pressure here. Then

$$F_{po} = \frac{1}{2}(P_b + P_a)(A_b - A_c) \qquad (9)$$

where A_b is the axial projected area of the orbiting scroll. $A_b = \frac{\pi}{4} D^2$ (where D is the diameter of the end plate of orbiting scroll.)

And A_c is the axial projected area of the suction pressure, compression chamber pressure, and central chamber pressure acting on the orbiting scroll.

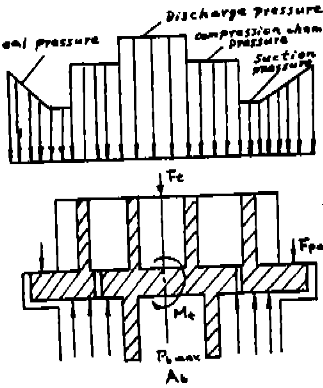


Fig. 5 Sketch of the self-adjusting backpressure mechanism and pressure distribution on orbiting scroll

Assume that the backpressure chamber pressure is just in equilibrium with the axial force and overturning moment at a instant. Then the axial force equilibrium equation of the orbiting scroll is is

$$P_{bmin} \cdot A_b = F_t + F_{po} + \frac{2M_t}{D} \qquad (10)$$

Substitute Eq.(9) into Eq. (10), and note that at the moment, $P_b = P_{bmin}$. Then,

$$P_{bmin} = \frac{2F_t}{A_b+A_c} + \frac{A_b-A_c}{A_b+A_c} \cdot P_s + \frac{4M_t}{D(A_b+A_c)} \qquad (11)$$

From Eq.(11), under given working conditions, with A_b and A_c constant and M_t only varying slightly, the variation of P_{bmin} is mainly affected by the axial force F_t. When F_t increases, P_{bmin} increases as well. When $\theta = \theta^*$ (where θ^* is the exhaust angle of the compressor), and F_t is the maximum, P_{bmin} reaches its maximum as well.

2. Determination Method of the Port Position

Minimum backpressure P_{bmin} provides the basis for testing whether the chosen port position (b,β) meets the requirements or not. If with the first chosen β, the thermodynamic and dynamic computations prove that $P_b(\theta)$ is greater than P_{bmin}, the determination of the position is then reasonable. Otherwise, β should be increased until the above requirements are met. However it should be noted that too high a chamber pressure $P_b(\theta)$ would increase the friction between the contact surfaces of the mating scrolls and result in the decline in the mechanical efficiency of the compressor η_m.

Therefore the determination of the port position should conform to the following principle.

When selecting the proper β, we should first guarantee $P_b(\theta) \geq P_{bmin}$ and the volumetric efficiency η_v, and at the same time restrict $P_b(\theta)$, to prevent the apparent decline in the mechanical efficiency η_m. The relationship between the polar angle of the port β and the efficiency of the compressor is shown in Fig. 6. It would be most ideal if the chosen β could always ensure $P_b(\theta) = P_{bmin}$; however, it is impossible to do so because $P_b(\theta)$ and P_{bmin} both vary with θ, and their variations are both very complex. Therefore, the following condition should be satisfied in the determination of the backpressure port position (b, β):

$$\beta_{min} \leq \beta \leq \beta_{max}$$

where β_{min} is the minimum β to ensure the mating scrolls do not separate.

and β_{max} is the maximum β to ensure the mechanical wear will not be too great.

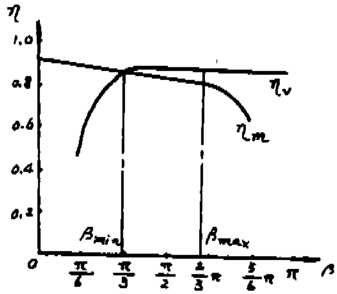


Fig. 6 Relationship between the polar angle of the port and efficiency

CASE ANALYSIS

The above method has been applied to analyse the backpressure port position in our scroll compressor, and the result is shown in Fig. 7.

From Fig. 7, when $\beta = 30^\circ$ or 45°, it cannot be ensured that $P_b(\theta) \geq P_{bmin}$; therefore it is not reasonable that $\beta = 30^\circ$ or 45°. when β 60°, it is ensured that $P_b(\theta) \geq P_{bmin}$; therefore $\beta_{min} = 60^\circ$ is appropriate to choose. When β continues to increase, $P_b(\theta) - P_{bmin}$ will increase as well. From Fig. 8, we know η_m will decrease, and when $\beta = 105^\circ$, η_m will have had an apparent decrease. Therefore the selection of β should satisfy: $60^\circ \leq \beta \leq 105^\circ$. Considering the reliability of the compressor, it is proper to select $\beta = 75^\circ$. Analysis and computation show that the proper positions of the pair of backpressure port centers on the orbiting scroll are at O_1(34mm,75°) and O_2(34mm,255°) respectively, as shown in Fig. 9.

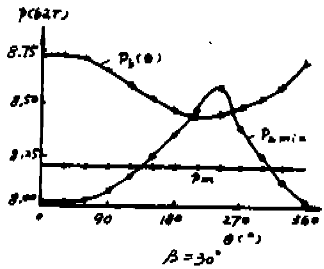


Fig. 7 Law of $P_b(\theta)$, P_{bmin} and P_{bm} changing with the angle of retation θ

(P_m is the average gas pressure in the compression chamber connected with the backpressure chamber in a cycle)

CONCLUSION

1. Backpressure mechanism is widely applied to balance the axial gas force in scroll compressors, which guarantees the compressor's ability to adjust to the changing working conditions.

2. Reasonableness of the backpressure mechanism is mainly demonstrated in

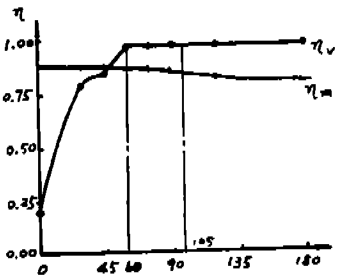


Fig. 8 Effect of B on the volumetric efficiency and mechanical efficiency

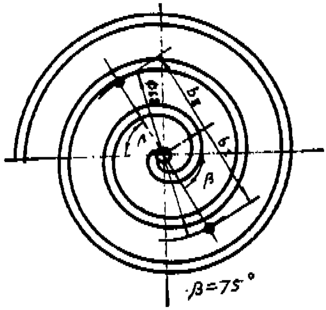


Fig. 9 Position of the backpressure port in the speciman compressor

1. orbiting scroll

the ability of the pressure in backpressure chamber to balance the axial gas force exerted on the orbiting scroll.

3. With the given chamber volume and port radius e, the gas pressure in the backpressure chamber depends on the position of the port (b, β).

4. b is determined by β, which has a proper range of $\beta_{min} \leq \beta \leq \beta_{max}$, thus ensuring the great volumetric and mechanical efficiency of the compressor.

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