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ABSTRACT

In the present paper, the critical geometric parameters of the single screw compressor related to the diameter ratio of engagement pair λ_d are analyzed. The correlation equation of the diameter ratio λ_d and the center-line spacing coefficient λ is presented. The effects of λ_d on the economy and operation performances of the single screw compressor are investigated. Base on the results obtained by the numerical simulation, the suitable range of λ_d is suggested to improve the performance of single screw compressor in the actual application.

1. INTRODUCTION

The engagement pair of the main screw and gate-rotor is the key part of the single screw compressor. The profile and the geometric parameters of the engagement pair affect directly the performance and the reliability of the machine. In the present paper, a theoretical analysis is performed to investigate the diameter ratio of engagement pair and its effect on the operation performance of the single screw compressor.

The flute number Z_1 and the diameter d_1 of the main screws and the teeth number Z_2 and the diameter d_2 of the gate-rotors are the critical parameters of engagement pair. Commonly the choice of Z_1 and Z_2 can be determined by the compression ratio. When the compression ratio is ranged from 7 to 10 for the standard combination, Z_1 and Z_2 are chosen to be 6 and 11, respectively. Under this condition, the choice of the diameter ratio of engagement pair $\lambda_d = d_2 / d_1$ emerges as an important factor for the single screw compressor design. In the design of earlier single screw compressor, the diameters of main screw and the gate-rotor are equal, i.e. $\lambda_d = 1$. However, the design of $\lambda_d > 1$ is gaining popular gradually in the industry application. The reason is deserved to be thinking deeply.

For a fixed diameter of main screw d_1, the advantages of increasing the diameter of gate-rotor d_2 are the increased penetration of the gate-rotor teeth into the main screw and the swept flute volume. The parameter λ_d is restricted by many factors such as the engaging angle in working process, the utilization of flute, structure and machining, etc. The present study is performed to determine the diameter ratio of engagement pair λ_d of the single screw compressor. The numerical simulation validates the results obtained by the theoretical analysis.

2. GEOMETRIC PARAMETERS ABOUT λ_d

2.1 The Numbers of Flutes Z_1 and Teeth Z_2

The basic geometric structure of the engagement pair of the single screw compressor is shown in Fig. 1. In order to operate equably the two parts of the engagement pair of single screw compressor, the number of the main screw Z_1 should be even. The superposed area of the out rounds of gate-rotor and the out edge of main screw increase with the increase of the numbers of flutes Z_1, meanwhile, the percentage of the flute and the swept flute volume decrease. The results obtained by Chen *et al.* (1981) showed that the number of flutes Z_1 is related to the compression ratio E. For the ranges of $E = 2 \sim 4$，$7 \sim 10$ and $10 \sim 16$, the numbers of flutes Z_1 are 4, 6 and 8, respectively.

The teeth number of the gate-rotor Z_2 should be a prime number with Z_1. Since every tooth of gate-rotor sweeps the flutes by turns, therefore, the engagement pair is abraded evenly during the process of engaging. Under this condition, the precision of machining and assembly can be easily controlled.

The maximum flute volume can be obtained when the number of the simultaneous engaging teeth between the gate-rotor and the flute is $Z_1 / 2$, i.e. the closed flute volume is located at the diagonal position. The compression ratio for the number of the main screw $Z_1 = 6$ and the simultaneous engagement teeth number of 3 is the most popular design in the application. Therefore, the combination of teeth numbers of the main screw and the gate-rotor is 6 / 11, which is the standard combination.

2.2 Engaging Angle α

As shown in Fig. 1, the engaging angle α is the angle of the gate-rotor turning from penetrating into the flute to quitting out of the flute. For the standard combination, the maximum engaging angle is obtained at the time when the main screw turns half cycle, i.e. the gate-rotor turns the angle of $6\pi / 11$. In fact, the engaging angle α is approximately $\pi / 2$ due to the space occupied by the two gate-rotors.

2.3 Top Width of Main Screw Tooth Δ

Referring to Fig. 1, the top width of main screw tooth Δ has the effects on the leakage, the swept flute volume and the tooth intensity of the single screw compressor. The seal performance of the single screw compressor can be improved by increasing the top width Δ. However, the swept flute volume decreases with the increase of the top width Δ. In addition, the tooth intensity of main screw will be too low and the bottom of the tooth will be too narrow if the value of Δ is very small, which are the difficulties in the process of machining. Generally, the coefficient of the top width of the main screw tooth $\xi = \Delta / d_1$ is ranged from $0.10 \sim 0.25$ (Sun, 1988).

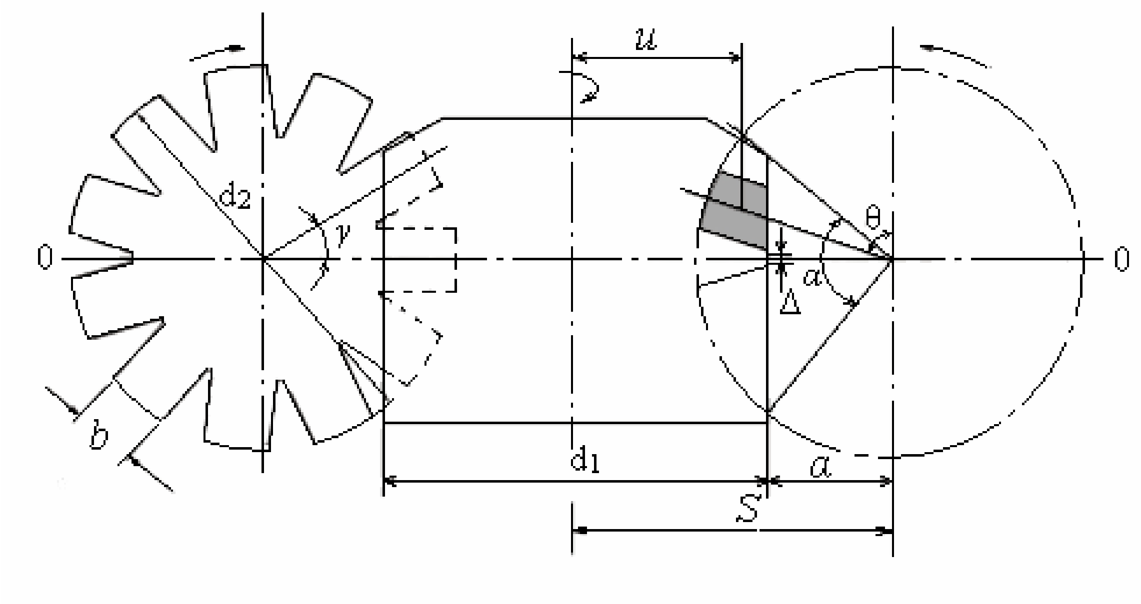

Figure 1 The basic geometry of the engagement pair of the single screw compressor

2.4 Center-Line Spacing S

The center-line spacing of the engagement pair $S = \lambda d_1$, where λ is the coefficient of the center-line spacing. The value of S depends on the degree of penetration of gate-rotor into flute, the engaging angle α and the flute volume. Fig. 2 shows the relation between the relative degree of penetration h and the relative flute volume V_Σ .

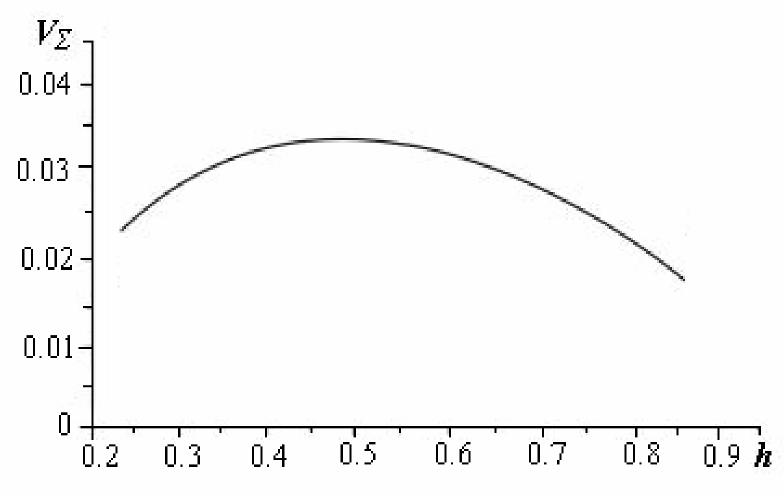

Figure 2 Relation between the degree of penetration h and the flute cubage V_Σ

It can be seen that the swept flute volume V_Σ increases first and then decreases with the increase of the degree of penetration h. This because that the flute width will be decreased to ensure the adequate width of Δ while the degree of penetration is continuously increased. Under the condition of $\lambda_d = 1$, the maximum value of the swept flute volume V_Σ is obtained at the degree of penetration $h = 0.45$ (Chen *et al.*, 1981). Correspondingly, the coefficient of center-line spacing is optimal, i.e. λ equals λ_{opt}. Based on the geometry structure of the engagement pair as shown in Fig. 1, λ_{opt} is obtained to be 0.8. The determination of λ_{opt} under the condition of $\lambda_d \neq 1$ will be analyzed in the next section.

3. RELATIONSHIP BETWEEN λ_d AND λ_{opt}

The swept flute volume of the engagement pair can be expressed as

$$V = f(A, u) \tag{1}$$

where A is the engaged tooth area of gate-rotor and u is its moment about the main screw axis. The maximum value of the swept flute volume V_{max} can be calculated by

$$(A, u)_{max} = \frac{bh_{max}(d_1 - h_{max})}{2} \tag{2}$$

where h_{max} is the maximum value of the degree of penetration of gate-rotor into flute and b is the width of gate-rotor tooth. Submitting the relevant geometric parameters of the engagement pair into Eq. (2), the above equation can be rewritten as

$$(A, u)_{max} = \frac{d_1^3}{8}(2e\lambda - C)\left[1 - (\lambda_d - 2\lambda)^2\right] \tag{3}$$

with

$$C = \sin(\gamma / 2) + \xi \cos(\gamma / 2) \tag{4}$$

and

$$e = \sin(\gamma / 2) \tag{5}$$

where γ is the division angle of gate-rotor. Calculating the extremum of Eq. (3) at $\frac{\partial[(SR)_{\max}]}{\partial \lambda} = 0$, the result gives

$$\lambda_{opt} = \frac{(2e\lambda_d + C) \pm \sqrt{(2e\lambda_d + C)^2 - 3e(e\lambda_d{}^2 + 2\lambda_d C - e)}}{6e} \tag{6}$$

where λ_{opt} is the optimal value of λ. Fig. 3 presents the relationship between λ_d and λ_{opt} for $\alpha \approx \pi / 2$ and $\xi = 0.2$ at the standard combination. It can be seen that for $\lambda_d = 1$, the optimal value λ_{opt} equals 0.8, i.e. the center-line spacing $S = 0.8d_1$ and shows an agreement with the analysis presented earlier. It is obviously that the optimal value λ_{opt} varies with the variations of the diameter ratio of engagement pair λ_d and the center-line spacing S. For a fixed d_1, both the structure sizes of the compressor and the swept flute volume V decrease with the decrease of λ_{opt} and S for $\lambda_d < 1$. For the condition of $\lambda_d > 1$, the swept flute volume V increases with increasing the degree of penetration of gate-rotor. However, it should be noted that the structure size of the machine is also increased at the same time.

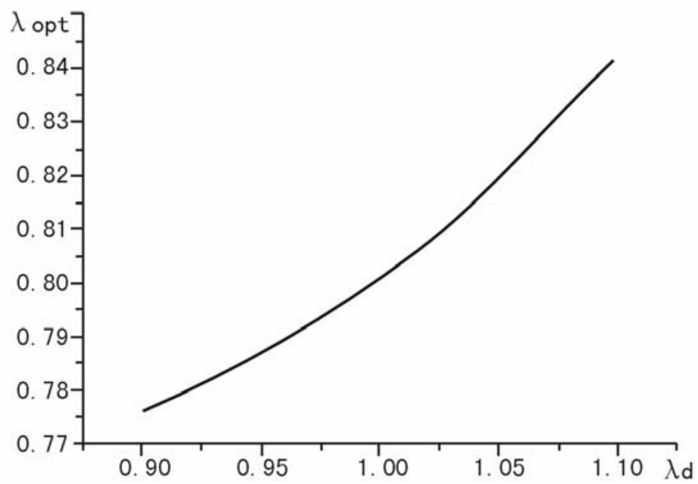

Figure 3 Relationship between λ_d and λ_{opt}

4. CALCULATION EXAMPLE

To investigate the effects of λ_d on the performance of the machine, a FORTRAN programme (Wu and Jin, 1988) is developed to simulate the working process of the single screw professor. The parameters of the prototype (oil-flooded and air-cooled) used in the simulation study are: (1) the volume rate of flow $Q = 3$ m^3/min; (2) the discharge pressure $P = 0.7$ MPa; (3) the volumetric efficiency $\eta = 0.85$; and (4) the rotation speed $n = 2940$ rpm.

In the calculation, the value of S/Q is introduced to evaluate the effects of λ_d on the performance and the structure size of the single screw compressor. This index represents the ratio of the center-line spacing to the volume rate of flow. A smaller value of S/Q means better economic performance of the single screw compressor. The results of the numerical study of the single screw compressor are tabulated in Table 1. It can be seen that the value of S/Q decreases gradually with increasing the diameter ratio of engagement pair for $\lambda_d > 1$. However, a too small value of S/Q will result in the difficulties of supporting structure design and the machining techniques of the engagement pair in the actual application. Therefore, based on the above analysis and calculation, $\lambda_d = 1.00 \sim 1.10$ can be considered as the suitable range for designing the engagement pair in the single screw compressor. It is also noted that the effect of the diameter ratio of the engagement pair on the specific energy requirement N/Q of the machine is less significant than that on the ratio of S/Q. The value of the specific energy requirement N/Q changes slightly with the variation of the diameter ratio of the engagement pair λ_d, since the shaft power N also increases with the increase of the volume rate of flow Q.

Table 1 Comparison of the economy performance of the single screw compressor with different λ_d

Parameters	λ_d	0.90	1.00	1.05	1.10	1.15	1.20
	λ_{opt}	0.774	0.800	0.816	0.833	0.855	0.867
d_2/d_1 (mm/mm)		143/160	160/160	167/160	176/160	183/160	192/160
S (mm)		124	128	131	134	137	139
Q (m^3/min)		2.51	3.08	3.41	3.70	3.98	4.34
N (KW)		14.42	17.68	19.61	21.23	22.85	24.95
S/Q [mm/(m^3/min)]		49.40	41.56	38.42	36.22	34.42	32.03

5. CONCLUSION

In this paper, a comprehensive analysis is performed to determine the diameter ratio of engagement pair λ_d of the single screw compressor. The results show that the diameter ratio of the engagement pair λ_d is related to the degree of penetration of the gate-rotor teeth to the flute and the value of the center-line spacing coefficient. An optimal value of the center-line spacing coefficient λ_{opt} corresponds to a diameter ratio of the engagement pair λ_d. For a fixed diameter of main screw d_1 and $\lambda_d > 1$, the ratio of S/Q increases with the increase of λ_d in a certain range. The simulated results indicate that the suitable value of λ_d can be chosen from the range of $1.00 \sim 1.10$. However, it should be noted that the value of the specific energy requirement N/Q changes slightly with the variation of the diameter ratio of the engagement pair λ_d, since the shaft power N also increases with the increase of the volume rate of flow Q.

NOMENCLATURE

A	engaged tooth area of gate-rotor	(mm^2)
C	factor	(-)
d_1	diameter of main screw	(mm)
d_2	diameter of gate-rotor	(mm)
e	factor	(-)
E	compressor ratio	(-)
h	degree of penetration	(-)
n	rotation speed	(rpm)
N	power	(W)
P	pressure	(Pa)
Q	flow volume rate	(m^3/min)
S	center-line spacing	(mm)
u	exposed tooth area of gate-rotor	(mm^2)
V	swept flute volume	(mm^3)

Subscripts

opt	optimal value
max	maximum

V_{Σ}	total flute volume	(mm^3)
Z_1	flute number	(-)
Z_2	teeth number	(-)
α	engaging angle	(º)
γ	division angle of gate-rotor	(º)
λ	coefficient of center-line spacing	(-)
λ_d	diameter ratio of engagement pair	(-)
ξ	coefficient of top width of main screw tooth	(-)
Δ	top width of main screw tooth	(mm)

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