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NEW ROTOR PROFILE AND ITS PERFORMANCE PREDICTION OF SCREW COMPRESSOR

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ABSTRACT

Improvement of screw compressor is dependent on advantage of rotor profile to a great extent. On the basis of studying the basic geometry and engagement characteristics of rotor profile, two types of new profiles are generated, all characteristic parameters of new rotor profiles are derived and construction parameters of new rotor profiles are optimized in this paper. The performance prediction of new rotor profiles shows that the performances of new profiles developed by the authors is better than traditional profile's.

1. INTRODUCTION

Since the invention of screw compressor, through continual improvement in rotor profile, machining method of rotor and whole system, the efficiency of screw compressor has been improved and has approached to or over the reciprocating compressor's, so sorew compressor is able to compete with reciprocating compressor.

After the first generation profile (symmetrical circular arc rotor profile) and the second generation profile (asymmetrical rotor profile) were designed, in the late 1970's a new asymmetrical profile(the third generation) developed. In comparison with the first generation profile, the efficiency of the second generation profile has increased by 8% to 12%, and the efficiency of the third generation profile has increased by 10% in comparison with the second one.

Reviewing improvement process of screw compressor, it can be known that the improvement of rotor profile has largely increased the efficiency of screw compressor. Two new protiles showing better performance have been developed in this paper.

2. THE BASIC GIST OF MODERN ROTOR PROFILE

For compressing gas economically, rotor profile must consist of several sequential rotor curve, relevant rotor curve of female and male rotor must meet basic law of rotor mesh and conjugating curve of known rotor curve is derived according to conjugating principle. The above described gists are indispensable conditions of rotor profile. Moreover, because rotor profile largely affects the efficiency, volume and machining of screw compressor, a newly designed profile must meet the following valuation basis.

2.1 Blow-Hole

Blow-hole results in forfeit ability of axial seal in screw compressor. Symmetrical circular arc profile has large blow-hole area, while point engagement epitrochoid's profile has the best axial seal characteristics. Because of above factor, in comparison with symmetrical circular arc profile, the efficiency of previous asymmetrical profile increases by about 10%. Since the back profile of all kinds of asymmetrical rotor profile consists mainly of epitrochoid, their efficiency approximately are the same. It is desirable that the rotor profile has smallest blow-hole area.

2.2 Contact Line Between Female and Male Rotors

Contact line is the line of linking the points which of helix surface generated by end profiles meet engaging principle. The surfaces of male and female rotors must have clearance for ratation, the theoretic contact line becomes real leakage area. In order to decrease leakage, the contact line length must be as short as possible. The equation of contact line is obtained by the following equation according to mesh principle

$$\vec{V} \cdot \vec{N} = 0$$

where $\vec{V}$: relative velocity of helix surface in contact point
$\vec{N}$: normal of contact point

2.3 Lobe Combination

The efficiency of screw compressor depends on tip linear speed to a great extent. Therefore, lobe combination 5/6 are better than 4/6, for 5/6 tip linear speed of female rotor approachs the male rotor's, the same linear tip speed of female and male rotro can improve the efficiency of screw compressor. In a word, the lobe combination of modern rotor profile tends to 5/6 instend of traditional lobe combination 4/6. Moreover, as the male lobe number increases, the discharge port opens earlier by an increase in the port opening angle. This allows increasing large axial and radial discharge port with increaseing male lobe number[2]. Considering these, the lobe combination of new rotor profile is taken 5/6 in this paper.

2.4 Confining Volume

Suction and discharge sides consist in confining volume which has a bad effect on delivery volume, shaft power and increases noise of screw compressor, confining volume in discharge side has larger effect on performance than confining volume in suction side. As mentioned above, point engagement epitrochoid can decrease blow-hole area, but has larger contact line length and confining volume. Confining volume probably causes oil impact which will endanger safety operation of oil flooding screw compressor. The effect of confining volume on performance can be relieved by designing an unload groove on side cover.

2.5 Area Utilization Coefficient

Area utilization coefficient indicates the utilizating rate of total area in rotor nominal diameter and is a function of rotor profile and its construction parameters (for example addendum radius R, lobe number, etc.). In a word, in order to increase area utilization coefficient one should design proper lobe combination and addendum radius.

2.6 Machining of Screw Rotor

The output of screw compressor increase in the recent two decades, so it is important to improve machining productive and precision of screw rotor. Besides high efficiency, the screw rotor must have fine cutting characteristics.

2.7 Streamlined Rotor Profile

Because screw compressor has high tip speed, modern new rotor profile is usually streamlined in order to decrease pressure loss of gas flow.

All points above mentioned are foundation to evaluate the characteristics of various rotor profiles, although some is mutual contradictory. For example, point engagement epitrochoid can decrease blow-hole area, but increases contact line length and has confining volume. Streamlined rotor profile can decrease dynamic loss of gas and oil, but increases blow-hole area. Basides rotor profile, with the improvement of machining technology the engaging clearance between two mating screw rotor can be properly arranged, this is also an important factor to improve the efficiency of screw compressor. Therefore, the design of new rotor profile must consider above all factors. It is no question to consider above all factors in design of new rotor profile though some of the factors can not be calculated quantitatively up till now.

3. NEW ROTOR PROFILE AND THE CALCULATION OF GEOMETRICAL CHARACTERISTICS

In this paper screw rotor profile is generated by deriving the conjugate profile of a known rotor profile according to conjugate principle. Useing this method and considering all factors in section 2, two types of rotor profiles (JD1, JD2) are developed.

3.1 JD1 Rotor Profile

JD1 rotor profile is shown in Fig.1

JD1 is a bilateral asymmetric epitrochoid ellipse profile. Forepart of the rotor profile, which because of in suction pressure has no need of axial seal, is designed as ellipse and ellipse evolute so that screw compressor has a short contact line length and is easy to machine. The back part of the profile is designed as point engagement epicycloid so that axial leakage is decreased to minimum (blow-hole area ≈ 0) and the efficiency of internal compression is enhanced. The authors use circular arcs to connect smoothly the forepart and the back park of the profile. The lobe combination of JD1 rotor profile is $Z1/Z2=5/6$, the gas velocity through discharge port and specific power is lower than the traditional rotor profile. In Fig.1, the addendum radius H of the female rotor is decided by consideration of blow-hole area, and the addendum radius R of the male rotor is decided by consideration of utilization coefficient, stiffness of female rotor and the performance of screw compressor.

In order to optimize parameters of rotor profile, the H and R are optimized . The results show that as R is decreased, the length of contact line is shorten, blow-hole area, area utilization coefficient are decreased, and discharge port area is enlarged. As H is decreased, discharge port area decreases slightly, while the other conclusions are similar to above.

3.2 JD2 Rotor Profile

JD2 rotor profile is shown in Fig.2.

JD2 is an unilateral asymmetry circular arc--circular arc evolute profile. Because all the rotor curves consist of circular and circular arc evolute, the contact line length of JD2 is shorter. Moreover, the male and female rotors of JD2 engage continually and smoothly, so JD2 can decrease noise, protect oil film, and improve lubricating characteristics of contact surface or line. All curves of JD2 are connected smoothly, this is beneficial to reduction of gas flow resistance.

On the basis of engaging characteristics of circular profile, the main considerations for selecting all radius of circular arc are: R6 and R7 are decided by decreasing blow-hole area, R4 is decided by the stiffness of female rotor and area utilization coefficient, R5 is decided by decreasing the length of contact line, R3 is decided by decreasing the length of contact line and balancing the shape of screw rotor.

3.3 Analysis and Comparision of Rotor Profiles

Geometrical characteristics of JD1, JD2 and traditional rotor profile (unilateral asymmetry epitrochoid--circular arc profile TR2) are calculated in this paper, the results are shown in Tab.1 and Tab.2.

Tab.1 Comparision of JD1, JD2 and TR2 Profile

Profile	Contact line Length (mm)	blow-hole area (mm^2)	area utiligation coefficient
JD1	526	1.08	0.436
JD2	530	1.03	0.390
TR2	632	0.77	0.469

Tab.2 Discharge port area (mm^2) of JD1, JD2 and TR2 Profile

Profile	operating condition						
	refrigeration			air-conditioning			air compressor Pd = 7 bar(g)
	R717	R12	R22	R717	R12	R22	
JD1	2603	2755	2968	6973	7505	7897	1135
JD2	2656	2812	3032	7138	7682	8083	1129
TR2	2941	3081	3348	7727	8302	8724	1260

Tab. 1 shows that the contact line length of JD1 and JD2 is much shorter than TR2, the blow-hole area of JD1 and JD2 is larger than TR2 profile, the area utilization coefficient of JD1 and JD2 is smaller than TR2 profile's, but the decrease of JD1's area utilization coefficient is small. Tab. 2 shows that for three operating conditions and four working substances, the discharge port areas of JD1 and JD2 are smaller than TR2 profile's. The leakage loss across clearance of contact line under an large pressure difference is the most important one, so maybe the overall performance of JD1 and JD2 is better than TR2 profile's.

4. PERFORMANCE PREDICTION

In order to calculate quantitatively the performance of new rotor profile compressor, the adiabatic efficiency and volumetric efficiency are derived by simulating working processes of screw compressor. Fig.3 shows the flow chart of computer program. Fig.4 to Fig.7 show the calculating results.

Fig.4 to Fig.7 show that among the three rotor profiles the performance of JD1 is the best, the performance of JD2 is better than TR2 the profile's. The η_i and η_v of JD1 increase by 6.1% to 14% and 2.1% to 8.0%, η_i and η_v of JD2 increase by 5.3% to 8.2% and 1.8% to 7.2% in comparasion with the TR2 profile.

5. CONCLUSIONS

5.1 Two types of new rotor profiles (JD1 and JD2) are developed in this paper. In order to design rotor profile with excellent performance the main foundation of designing new rotor profile is: forepart of rotor profile has short contact line length, backpart of rotor profile has proper contact line length and small blow-hole area.

5.2 The construction parameters of JD1 and JD2 is derived and optimized, the performance of JD1 and JD2 is predicted by computer simulation.

5.3 The results of performance prediction show that the performance of JD1 and JD2 is better than TR2 the second generation rotor profile,the volumetric efficiency $\eta_v$of JD1 increases by 2.1% to 8.0%, adiabatic efficiency η_i of JD1 increases by 6.1% to 14%, while η_v of JD2 increases by 1.8% to 7.2% , η_i of JD2 increases by 5.3% to 8.2%.

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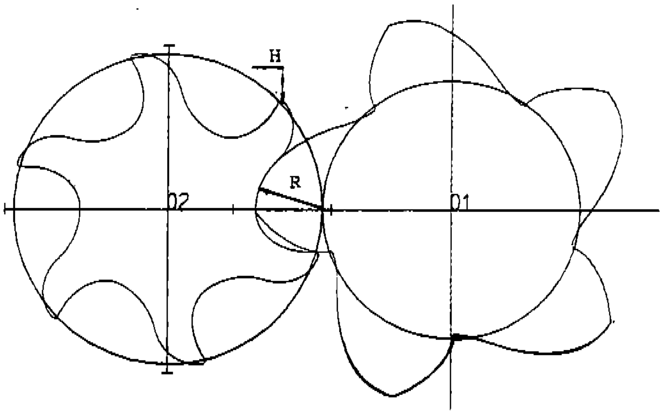

Fig.1 JD1 rotor profile

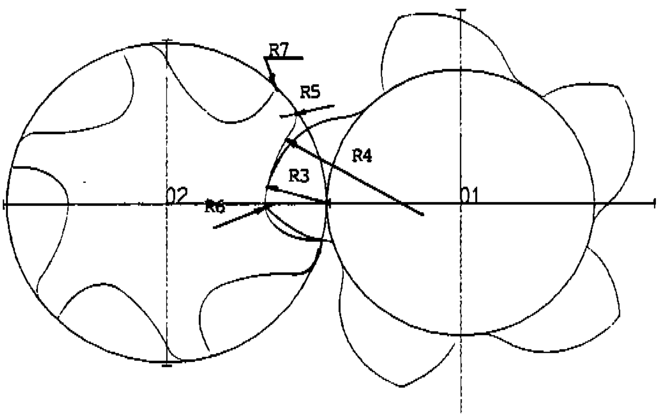

Fig.2 JD2 rotor profile

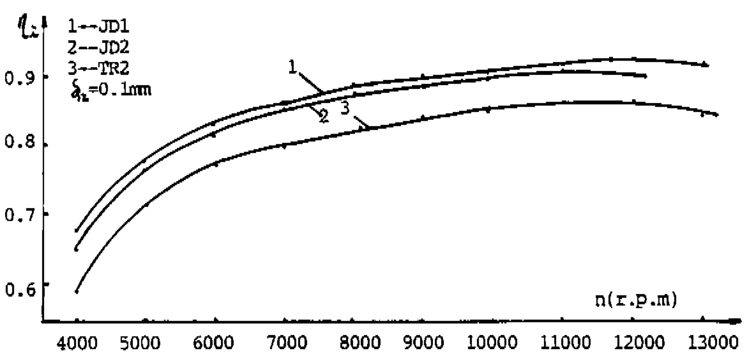

Fig.7 Adiabatic efficiency versus rotation speed of male rotor

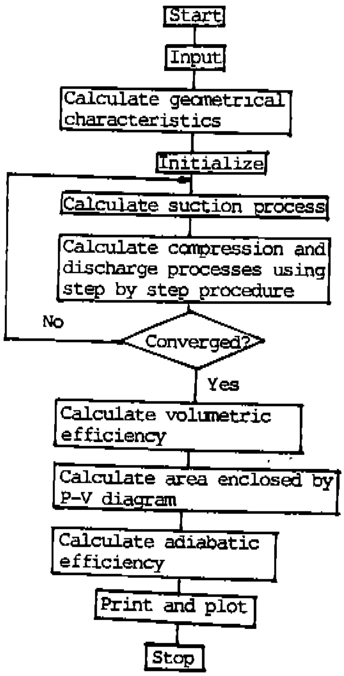

Fig.3 Flow chart for performance simulation

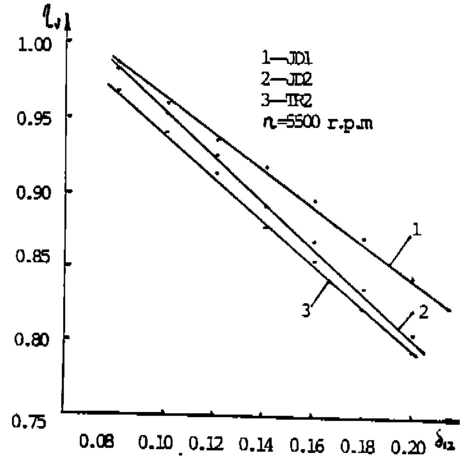

Fig.4 Volumetric efficiency versus engagement clearance

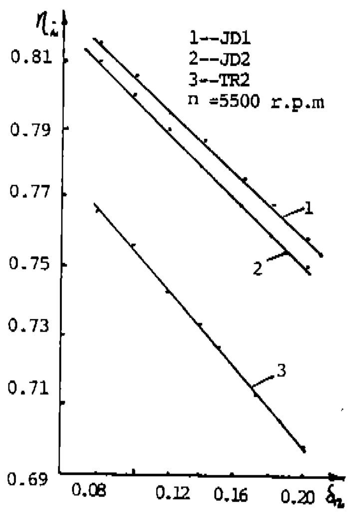

Fig.5 Adiabatic efficiency versus engagement clearance

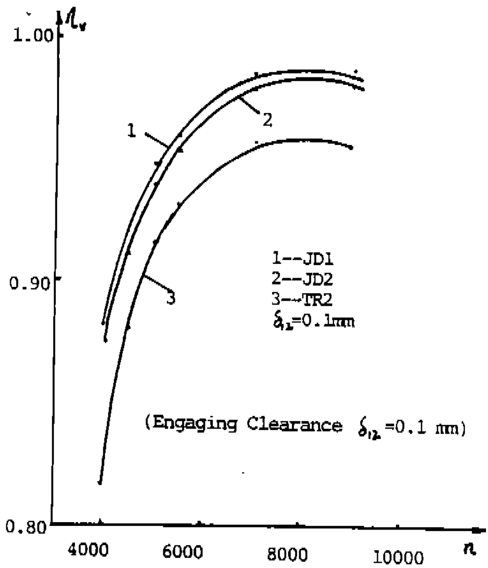

Fig.6 Volumetric efficiency versus rotation speed of male rotor