2002

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C19-4

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Abstract

Through the performance test of screw air compressor by changing work conditions, analyzing the test results, investigating other air compressor basic parameters, analyzing various factors which affecting the performance of screw air compressor, several suggestion and reference about the auxiliary equipment design and choice of screw air compressor are presented in this paper.

Nomenclature

N	Male rotor rotation rate	rpm
P_s	Suction pressure	kPa
T_s	Suction temperature	℃
P_d	Discharge pressure	MPa
T_d	Discharge temperature	℃
T_{oil}	Injected oil temperature	℃
Q	Delivery capacity：	m^3/min
N_z	Shaft power:	kW
η_v	Volume efficiency	
η_{ad}	Adiabatic efficiency	
q_v	Specific power	$kW/m^3.min^{-1}$
D	nominal diameter of intake Controlling valve	mm
d	confining flow hole diameter	mm

Introduction

With lots of unique advantages such as compactness, light in weight, simple in structure, no wearing parts，reliable and easy in operation，screw air compressors now have been being applied more and more in the fields where middle or larger delivery capacity and middle or lower gas pressure are needed, especially in small compact space. As we know, screw air compressor unit comprises air end and auxiliary equipment such as oil separator, oil cooler, suction control valve and air filter etc. The design, manufacture and assembly of air end are important certainly, but the design and choice of auxiliary equipment is also the key factor that decides the performance of screw compressor.

In July, last year, owing to practical application demand，we carried out a lot of performance tests on a set of 3.5 m^3/min screw air compressor under various work conditions, solved many problems arising during experiments，analyzed test results，investigated other air screw compressor basic parameters, and we obtained some new ideas and acquaintance about screw compressor design.

Prototype geometric parameters：

Rotor center distance： A=80mm
Male rotor outside diameter： $D1_O$=112mm
Female rotor outside diameter： $D2_O$=96mm
Rotor length： L=185mm
Distortion angle of rotor： τ 1=296O
Male rotor led： T1=225mm
Area utilization factor： Cn=0.485
distortion coefficient： C_Φ=0.977

Prototype main performance parameters：

Male rotor rotation speed： 4010rpm
delivery capacity： ≥3.5m^3/min
discharge pressure： 0.7MPa
Specific power： ≤6.6kW/m^3.min^{-1}
Linear velocity： 23.5m/s
bare unit noise： ≤82dB(A)

General performance test system

The compressor is tested according to the methods and specification of international standard ISO 1217-96 displacement compressor ---Acceptance test[1] and China National standard [2], GB3853-83"Methods of Performance Test for General Displacement Compressor”. Test system diagram refers to figure 1.

Variable speed performance Test results:

In order to examine the prototype reliability and supply some credibility useable data for the prototype remodeling design, the general performance of the prototype was tested under various Male rotor rotation rates by changing the rotation speed of motor through the frequency modulator. The test results are shown in Table 1.

Table 1：

N	2899	3499	4010	4500	4990
P_s	101.17	101.17	101.17	101.17	101.17
T_s	32.2	34.2	34.4	34.4	35.3
P_d	0.7	0.7	0.7	0.7	0.7
T_d	77	81	84	95	100
T_{oil}	51.0	54.0	60.0	61.6	63.7
Q	2.44	2.99	3.44	3.85	4.01
N_z	15.74	19.32	22.51	25.69	29.69
η_v	0.7655	0.7761	0.7801	0.7787	0.7308
η_{ad}	0.7423	0.7410	0.7318	0.7176	0.6467
q_v	6.41	6.46	6.54	6.67	7.40
Note： D=45mm d=4mm					

Variable discharge pressure performance test results

In order to understand further the prototype's adaptive ability to variable work conditions and provides the solutions to solve the problems arising in service; we tested the general performance of the prototype under variable discharge pressure. The test results are shown in Table 2.

Table 2

N	4000	4000	4000	4000
P_s	101.08	101.08	101.08	101.08
T_s	29.7	31.8	31.9	30.6
P_d	0.5	0.7	0.8	1.0
T_d	86	88	91	94
T_{oil}	59	59.6	62.1	64.7
Q	3.45	3.43	3.09	2.93
N_z	18.78	22.3	23.72	26.83
η v	0.7843	0.7798	0.7025	0.6710
η_{ad}	0.7241	0.7359	0.6709	0.6336
q_v	5.44	6.50	7.68	9.16
Note： D=45mm d=4mm				

In addition, we installed a suction air filter before the suction-controlling valve of the compressor and tested its noise according to [3] GB7022-86 "Determination of sound power level for noise emitted by displacement compressors-Survey method". The average value of Sound pressure level of the compressor tested is 86 dB (A) and no satisfaction with the design requirement.

Test result analyzing

From Table 1, it is observed that the volumetric efficiency of this compressor is increasing with the rotation speed N increasing from 2899 to 4010rpm and reaches its maximum value when the rotation speed N is about 4000rpm; but the volumetric efficiency of this compressor is decreasing when the rotation speed N is increasing from 4010 to 4990rpm, it decreasing sharply especially from 4500 to 4990rpm. The above phenomena aren't consistent completely with the theoretical analysis. From the theory, the volumetric efficiency of the compressor should increase continually, instead of decreasing with the rotation speed N increasing when the leak clearance area of screw compressor is constant, we think above phenomena result from the smaller flow area of air suction because of unsuitable suction controlling valve of the compressor.

Because the nominal diameter of suction controlling valve installed on the compressor is too small, the air flowing speed in suction port is too large and the pressure loss of air inlet increases, these result in suction gas flow insufficiency and lead to the decreasing of delivery capacity (no up to design value 3.5 m^3/min). In particular, when rotation speed N increase to a certainty value, the volumetric efficiency of the compressor would decrease. That is explanation about the test results listed in table 1.

Because the nominal diameter of suction controlling valve installed on the compressor is too small, the suction gas flow speed would increase rapidly, and the suction resistance would also increase with the rotation speed N increasing, at the same time, they would cause power consumption extra increase. That is to say, the Specific power of this compressor would increase with the rotation speed N increasing. When rotation speed N increase to a certainty value, the specific power would exceed design specified value. Unsuitable suction controll valve of the compressor is an important factor that brings high noise of the compressor.

From Table 2 it is observed that the delivery, volumetric efficiency and adiabatic efficiency of the compressor are all decreasing with discharge pressure increasing; while the changing trend of specific power is opposite. The test result is consistent with theory. Because leakage increasing with discharge pressure increasing, it results in delivery capacity decreasing and power consumption increasing, so volumetric efficiency and adiabatic efficiency decrease whereas specific power increase with discharge pressure increasing. From the other viewpoint, the above changing is too great to conform theory and experience. The past experience and theory [4] tell us that the volumetric efficiency and adiabatic efficiency ought not to have such great changing when the discharge pressure changes from 0.5MPa to 1.0MPa. We think these phenomena result from two aspects as following: (1) it is the nominal diameter of suction controlling valve is too small that cause suction resistance become greater especially when discharge pressure is higher. (2) it is the flowing diameter of oil return pipe is too large installed in oil separator that results in delivery capacity loss and extra power consumption.

While the oil accumulated at the bottom of filter element in oil separator is led into suction chamber through the oil return pipe from the bottom of filter element, a part of high pressure gas enter into suction chamber where it expands to low pressure gas, so the power consumption will increase and suction capacity decrease. The larger the flowing diameter of oil return pipe and the higher the discharge pressure, the more the gas entering into suction chamber through the oil return pipe and the more greatly the delivery loss and extra power consumption. That is explanation about the test results listed in table 2.

Through above analyses, we knew clearly the existing problem, so we installed a new suction controlling valve with nominal diameter of 65mm instead of the old one, and added a small hole(Φ0.8mm)in the oil return pipe. In addition, we wrapped the suction pipe with the noise silencing material and tested the general performance of the compressor. The test results are shown in Table 3.

Table 3

N	4010	Q	3.60	η_v	0.8164
P_s	101.36	T_d	91	η_{ad}	0.7724
T_s	33.8	T_{oil}	60.0	q_v	6.21
P_d	0.7	N_z	22.36	L_{PA}	81
Note: D=65mm d=0.8mm					

The test results listed in table 3 meet with the design requirements.

Fore and Aft improvement, the relation of the volumetric efficiency and the adiabatic efficiency to the rotation speed is illustrated in Fig.2 and to the discharge pressure in Fig.3.

It's observed that the performance of improved compressor is much more superior to that of the unimproved one.

Conclusion

Through the experimental study on the prototype and investigation of the other specific screw air compressors, we could draw the following conclusions:

The better performance of screw air compressor unit not only depend on the design, manufacture, and assembly of the air end, but also depend on the design and choice of the auxiliary equipment of screw air compressor.

1. The suction controlling valve should be chosen with reference to the flowing Velocity V_{in} of air intake.

   V_{in} could be obtained from the following equation:

$$V_{in} = \frac{Q \times 10^6}{15 \pi D^2} \quad (m/s)$$

   V_{in} should be (10~ 20)m/s.

2. Installing or changing pipe line structure（as installing a small hole plate in the oil return pipe line）could improve the performance of compressor unit.

References:

[1] ISO 1217-96 displacement compressor---Acceptance tests 1996-09-15

[2] China National standard GB 3853-83 "Methods of Performance Test for general displacement compressor”.

[3] China National standard GB7022-86 “Determination of sound power level displacement compressors-Survey method”

[4] Dingguo Deng, Pengcheng su 1982 “Rotary Compressor” Xi’an Jiaotong University

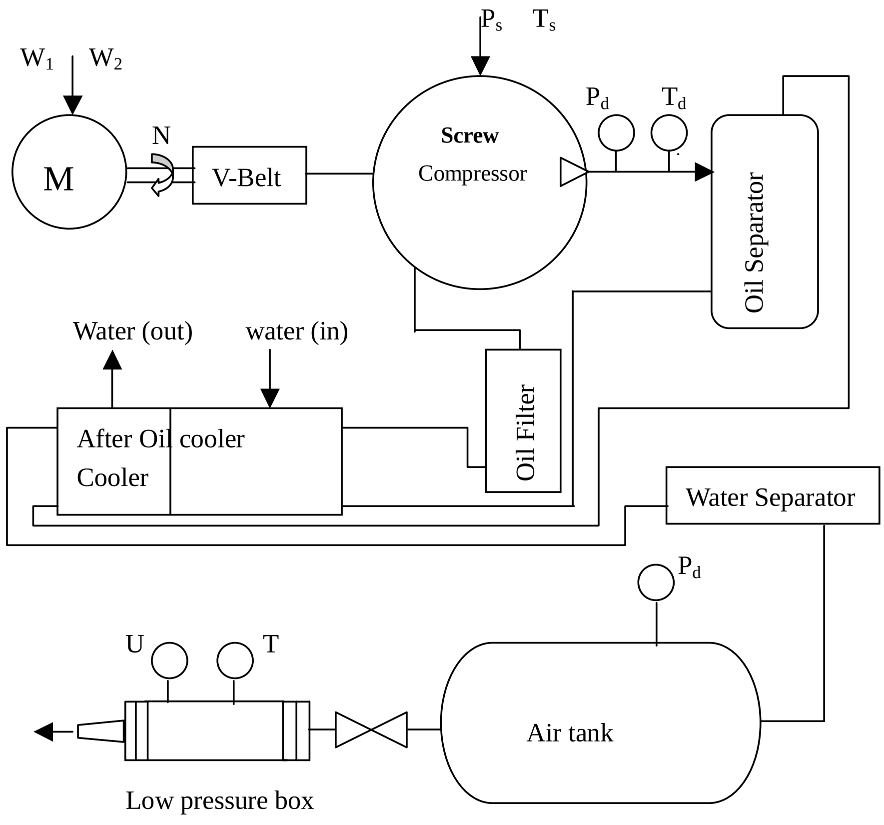

Figure 1 Test system diagram

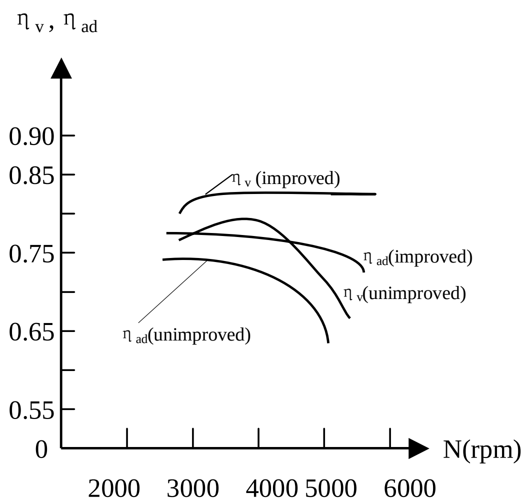

Fig. 2 Relation of the volumetric efficiency and the adiabatic efficiency to the male rotor rotation speed

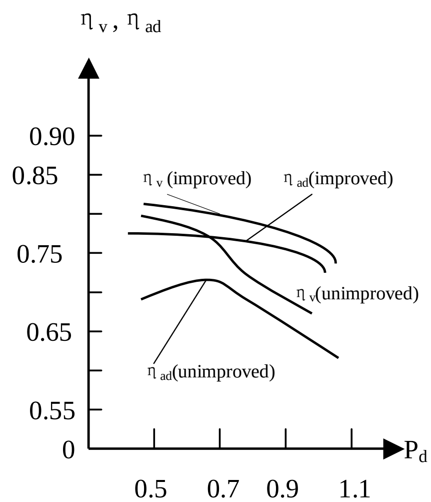

Fig.3 Relation of the volumetric efficiency and the adiabatic efficiency to the discharge pressure