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DESIGN AND CALCULATION ON A MINIATURE HIGH-PRESSURE COMPRESSOR USED IN CLOSED THROTTLE REFRIGERATOR

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ABSTRACT

Design and calculation on a miniature high pressure compressor used in closed throttle refrigeration system has been introduced. Choice of thermodynamic parameters, determination of system resonant frequency, design of motor, stability of piston-shaft moving, sealing and valve structure have been analysed in this paper.

0. INTRODUCTION

In the field of infra-red technique and cryogenic technique, throttle refrigerators have been used widely which are a kind of throttle cyclic system in light of J-T refrigeration effect and high pressure gas has been filled in a bottle to supply for the system or a miniature high pressure compressor gives off high pressure gas directly to refrigerator at throttle pressure. In this paper, a kind of miniature compressor driven by electro-magnetic resonant linear motor has been studied.

1. THEORY AND GENERAL STRUCTURE

The compressor driven by electro-magnetic resonant motor is used to raise gas pressure on the basis of electro-magnetic and mechanical resonance. According to driving mechanism, there are two sorts of moving coil and moving soft iron yoke. The structure introduced in this paper is moving coil structure, whose coil flowed with alternating current is exerted with a sine reciprocating force in magnetic field which is sent off by permanent magnet to make piston-shaft reciprocating suction, compressing and discharging gas. In the light of working theory, the multi-stage cylinders of miniature compressor could not be ranged in angle. Four-stage compression is chosen in the design and four cylinders are ranged symmetrically at two sides of driving system linearly. Compared with usual reciprocators, the compressor driven by electro-magnetic linear motor has special suspension between piston and cylinder, and it has fewer parts and components, flank force acted on piston is much less, and clearance sealing is easily realized. Reciprocating friction force is reduced, wear and noise are also reduced, piston stroke can be adjusted and capacity can be adjusted from...
0 to 100% at continual operation, permanent magnet field gives off stable magnetic flux. The maximum power could not exceed 120 W. Fig.1 shows the general structure.

2. CHOICE OF THERMODYNAMIC PARAMETERS

In comparison with reciprocator of crank-rocker mechanism, the compressor driven by electro-magnetic resonant motor is an special kind of reciprocator which has high velocity and short stroke. So it has characters of general reciprocator: cylinder volume efficiency reduces as a result of clearance volume, spring force of suction valve, pressure pulsation of valve chamber, heat exchange between sucked gas and cylinder and gas leakage. In design and calculation, all above factors are expressed as follows:

\[ \eta_d = \lambda_v \cdot \lambda_p \cdot \lambda_t \cdot \lambda_l \]  \hfill (1)
\[ \lambda_v = 1. - a \cdot (\varepsilon^{1/m} - 1) \]  \hfill (2)
\[ \lambda_v = 1 - a \cdot \left( \frac{Z_4}{Z_3} \cdot \varepsilon^{1/m} - 1 \right) \]  \hfill (3)

where,  
\( \lambda_v \) ---- volume coefficient;  
\( \lambda_p \) ---- pressure coefficient;  
\( \lambda_t \) ---- temperature coefficient;  
\( \lambda_l \) ---- leakage coefficient;  
\( a \) ---- relative clearance volume;  
\( \varepsilon \) ---- compression ratio;  
\( Z_4 \) ---- expansion quality index at expansion end;  
\( Z_3 \) ---- compression quality index at compression end;  
\( m \) ---- expansion index.

The concrete clearance volume of high pressure stages are too small, so valve design and range are difficult. Pressure difference of high pressure stage is greater and its leakage coefficient is larger. So sealing problem has been carefully considered and generally speaking, the volume efficiency are small in accordance with (1), (2) or (3). It is the feature of this miniature compressor.

3. MOTOR MOVING BALANCE AND RELATIVE CLEARANCE VOLUME

Due to sine movement, electrically driving force can perform equal work at positive and negative stroke, that is equal to output power. The mechanical moving part of the compressor designed by us could be simplified as vibration of single degree of freedom with damping. In order to keep compressor to work accurately, motor moving balance position
must be duplicated with motionless position, that is the amplitude of two side is equal, or work performed by piston in positive and negative stroke is equal. This is easily done in symmetry range and same stage compressor. The stability calculation of symmetry range and same stage compressor driven by electro-magnetic resonant motor refers to [1]. In design of our compressor, the first and the fourth stage are ranged in one side, opposite to the second and the third stage in other side. Its moving balance calculation is complex. On the other hand, average force in reciprocating course must be equal, considering keeping reciprocating stroke equal and equal work powered by electrically driving force in each stroke. According to the former, cylinder range could keep moving balance in accordance with or near motionless position.

Considering energy balance equation, output work powered by electrically driving force and friction work from damping is equal or approximate in positive and negative stroke. They are neglected in energy equation. The work produced from atmosphere at area difference between first and second stage piston: Respective stage compression and expansion work needs to be considered attentively. Therein, \( W_1, W_2, W_3, W_4 \) are each stage algebra sum of compression and expansion work. For example, \( W_1 \) is expressed as follows:

\[
W_1 = V_{k1} \cdot \left( 1 - \delta_{oi} \right) \cdot \frac{\gamma_1 \cdot P_1 \cdot V_{k1} \cdot \gamma_1 - 1}{\left( \varepsilon_1 \cdot (I + \delta_{oi}) \right) \cdot \left( \frac{\mu_{oi}}{\mu_{oi}} \right) - 1}
\]

there, \( \delta_{oi} \) ---- suction relative pressure loss;
\( \delta_{oi} \) ---- overall relative pressure loss;
\( P_1 \) ---- suction pressure;
\( V_{k1} \) ---- stroke volume;
\( \gamma_1 \) ---- compression index;
\( \varepsilon_1 \) ---- nominal compression ratio.

Among (4), relative clearance volume is an adjustable parameter. Adaptable value adjustment could make energy balance on condition that mass flow of each stage don't change. The change of clearance volume is relevant to valve design. So choice of relative clearance volume is not only thermodynamic but also with regard to considering moving balance.

4. RESONANCE AND AMPLITUDE

Just as said above, the mechanical moving part could be simplified as forced vibration with damping, as shown in fig. 2(a). Fig. 2(b) shows armature diagram. Armature mass includes coil, piston-shaft and part of diaphragm suspension system. \( K \) represents composite rigidity coefficient of working spring and equivalent gas spring. \( c \) is equivalent effective damping coefficient and \( f \) shows electro-magnetic driving force. Equation (6) is general mathematical mode of compressor linearly driven by moving
coil.

\[ U = i R_e + B_e l e Z + L_e \frac{d i}{d t} \]
\[ m \ddot{Z} + C Z + K Z = f_m \]
\[ f_m = i B_e l e \]

there, \( R_e \)--corresponding resistance;
\( L_e \)--corresponding length;
\( B_e \)--corresponding inductance;
\( K \)--magnetic induction strength.

In order to make motor output power more useful, mechanical vibration system must work in resonance, that is, intrinsic frequency of mechanical vibration is equal to or near motor moving frequency. Raising motor efficiency is relevant to calculation of corresponding gas spring.

By virtue of calculation, gas force to armature varies with piston displacement during performance, as shown in Fig. 3. \( X \) represents piston displacement. The diagram shows one period. The linked line from point 1 to 3 is representative approximately of equivalent gas spring rigidity curve. The gas spring rigidity line crossed with displacement axis at middle point. That is amplitude of two side is equal.

Although piston amplitude of two side is equal in theory, in practice, collision of piston with cylinder end would take place. Particularly at start when each stage pressure has not been built up and motor is not steady in operation, this case would easily take place. So amplitude should be controlled by mechanical method. The structure of diaphragm suspension system matched with stroke controllers could effectively prevent the collision of piston with cylinder end.

In light of gas rigidity and suspension diaphragm system rigidity, which are paralleled connection, composite rigidity of mechanical system could be calculated and intrinsic frequency of mechanical vibration could be obtained, added with vibrator mass. The result has been compared with driving frequency. If they are not equal, the stroke determined at first (double amplitude) or vibrator mass or driving frequency (can be controlled) needs to be adjusted in order to change mechanical system rigidity and intrinsic frequency and to match driving frequency.

5. RESONANT MOTOR DESIGN

Motor useful output power is formulated as follows:

\[ P = U I \cos(\theta) \]

there, \( U \)--effective input voltage;
\( I \)--effective input current;
COS(θ)----power factor.

The rise of efficiency is associated with many aspects. Design of magnet path is an important respect. Moving coil flowed with alternating current produces negative electrical potential in magnetic field is hoped and the value is greater as soon as possible. In the meantime, moving coil produces self-induction electrical potential is not hoped, since self-induction reduces considerably motor power factor. Accurate design of magnet path, including wiring and winding, could eliminate self-induction opposite to electrical source. So power factor could be raised.

Magnetic field is produced from Nb-Fe-B permanent magnet. This kind of magnet has good features and is adaptable to linear motor with small power.

6. SEALING AND VALVE STRUCTURE

Clearance sealing technique has been used in our design. Fig. 4 shows the sketch of leakage chamber. Actual sealing structure have been calculated and leakage flow is controlled in permission.

Valve structure has been treated elaborately, in particular, the third and the fourth stage suction valve structure effectively reduce expansion volume of high pressure gas in clearance volume and raises gas flow.

A patent as regards the miniature high pressure compressor has been applied for. All above have been written in patent disclosure.

7. CONCLUSION

We have designed a high pressure miniature compressor which has been used as a prototype for throttle refrigerator. The prototype performance has verified that all above design thoughts are correct. However, we must improve technologically to make it a perfect product.

REFERENCES:
Fig. 1 sketch of general structure

Fig. 2(a) simplification of mechanical system

Fig. 2(b) armature diagram
Fig. 3 relation between force and displacement

Fig. 4 sketch of leakage chamber