A Theoretical Study on the Novel Structure of Vane Compressor for High Efficiency

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Abstract. Aiming at the problem of excessive mechanical loss of the conventional vane compressor, this paper proposes a novel vane compressor structure. This compressor can significantly reduce the mechanical frictional loss through converting sliding friction between vane tip and cylinder into rolling friction by using a rolling bearing. The structure and operation principle are introduced in this paper, and mechanical friction loss calculation models of these two kinds of compressor are theoretically analyzed. The results show that mechanical loss of the novel vane compressor can be reduced by nearly 38\% under the same working conditions. At the same time, the actual tested results indicated that the total power consumption of compressor decreased 160.1W (6.89\%), and the COP increased by 11.89\%.

1. Introduction

With the advantages of simple structure, easy processing, self-balance of rotor, low noise, less vibration, little torque ripple, low cost of manufacturing and so on, vane compressor is widely used in the fields of air compressor, vacuum pump, small refrigeration, air conditioning equipment and automotive air conditioning compressors.

The main disadvantage of conventional vane compressor is that mechanical friction between vane tip and cylinder is more severe. Reference [1] discussed the optimal back pressure of the vane and accordingly designed a segmented throttling back pressure chamber. References [2] pointed out that the frictional loss of vane tip can account for about 87.1\% of the total mechanical loss of the compressor. References [3-5] analyzed the back pressure acting on the tail of vane, indicated that under the premise of ensuring the contact between the vane tip and the inner wall of cylinder, reduced
the contact force could decrease the friction loss and thus improve the performance of compressor. Reference [6] set three back pressure states, by comparing those different influences on motion state and characteristics of the vane, it is shown that constant back pressure is more conducive to improve the motion characteristics, and then improve the performance of compressor. The above studies have focused on how to reduce the power consumption of vane tip. Therefore, the frictional power consumption and reliability between vane tip and cylinder are the biggest bottlenecks limiting the application of the vane compressor in a wider field.

This paper proposes a new type of vane compressor. By changing cylinder of the conventional vane compressor into a cylinder and rolling bearing, the vane tip contacts inner ring of the bearing, and keep a certain clearance between inner wall and cylinder. The inner ring of bearing rotates together with vane under the action of friction, which reduces the relative speed of the vane tip and the inner wall of cylinder. Therefore, the frictional power consumption of the vane tip is reduced, and improving the defects of the conventional vane compressor to some extent, and then expanded its application range.

2. Basic structure
Figure 1 shows the basic structure of a novel high efficiency vane compressor (HEVC). Compared with the conventional vane compressor (CVC) 2 rolling bearings are added on and beneath the cylinder. HEVC mainly consists of a shell, a motor and a compression mechanism. The compression mechanism is accommodated in the bottom of shell, and it is composed of a main bearing, cylinder, sub bearing, shaft, two rolling bearings and three vanes (Figure 2). The suction port and discharge port are set in the cylinder radial. Low-pressure refrigerant is sucked into the compression mechanism. Then it is pressurized to high pressure in compression mechanism and discharged into the interior of shell. Finally, the refrigerant is discharged to the outside of compressor through the gap of the motor and discharge pipe.

![Figure 1. Basic structure of HEVC](image-url)
Figure 2 shows the basic structure of the compression mechanism. There are 3 vane grooves in the shaft, vanes are assembled in each vane groove.

Figure 2. Basic structure of compression mechanism

Figure 3. Assembly of the vane tip with cylinder and rolling bearing

Figure 3 shows the assembly of vane tip with cylinder and rolling bearing. The cylinder and rolling bearing are assembled coaxially. And the diameter of cylinder is smaller than that of the inner rings. So the vane tip contacts the inner ring of bearing, then forms a clearance $\delta_c$ with the inner wall of cylinder. If $\delta_c$ is too small, it may lead vane to contact with the cylinder when the former parts are subjected to deformation, which cannot reduce the friction. If $\delta_c$ is too large, it will cause leakage. Therefore, $\delta_c$ has an optimal range.

3. Operating principle

Figure 4 shows the compression process of HEVC. During the operation of compressor, the friction force of vane tip between inner ring drives rolling bearing to rotate. Vane tip doesn’t contact with cylinder, and there is no friction between them. As a result, friction loss and wear of vane tip can be reduced effectively which will improve HEVC’s reliability at the same time.

Figure 4. Compression process of HEVC
4. The analysis of mechanical loss

Compared with conventional vane compressor, HEVC converts part of the sliding friction between vane tip and cylinder into rolling friction by introducing rolling bearings, which reduces frictional power loss significantly. But, the inner ring end and the rolling bearing itself bring extra power loss. Therefore, it is necessary to analyze the overall mechanical loss.

The compressor has 3 kinds of lubrication: boundary, mixed, fluid friction. Because of the complexity of frictional process, a part of the load is supported by solid contact (boundary friction) and remainder by a fluid film. So it is very difficult to determine the coefficient of friction in mixed frictional process. Therefore, the friction at each location is assumed to ideal state: boundary or fluid friction. When friction meets the following conditions: convergence space, lubricating oil and speed difference, the fluid dynamic pressure lubrication can be used to calculate the conditions. The rest is calculated according to the boundary friction.

Including friction loss due to viscous drag between the kinematic pair and lubricating oil in a shell chamber, 9 locations where frictions occur are assumed as shown in Table 1, Figures 5 and 6. A part of structure parameters and its variable or range is shown in Table 2.

![Figure 5. Locations of some friction](image1)

![Figure 6. Friction locations of vane and van tip](image2)
Table 1. Friction models assumed

| No. | Locations of friction | Model       |
|-----|----------------------|-------------|
| 1   | Shaft and Main Bearing | Fluid       |
| 2   | Shaft and Sub Bearing  | Fluid       |
| 3   | Rotor end and Main Bearing End (Upper Thrust Bearing) | Fluid       |
| 4   | Rotor end and Sub Bearing End (Lower Thrust Bearing) | Boundary    |
| 5   | Vane end and Bearing End | Boundary    |
| 6   | Vane and Vane Groove  | Boundary    |
| 7   | Vane Tip and Inner Ring(Cylinder) Wall | Fluid       |
| 8   | Inner Ring End        | Fluid       |
| 9   | Rolling Bearing       | Other       |

4.1. Shaft and Main Bearing
The main bearing is a typical journal bearing to be used for the compressor. Based on the above hypotheses of fluid motive power lubrication, the Reynolds equation for fluid frictional loss is written as [7],

\[ W_1 = \frac{2\pi \eta \omega_s^2 R_s^2 h_{mb}}{\delta_{sm}} \] (1)

where \( \eta \) is the dynamic viscosity of lubricating oil, \( \omega_s \) is the angular velocity of shaft, \( R_s \) is the radius of shaft, \( h_{mb} \) is the height of main bearing, \( \delta_{sm} \) is the oil film thickness between the long part of shaft and the main bearing.

4.2. Shaft and Sub Bearing
Similar to the main bearing, Petroff’s equation is also applied for the sub bearing,

\[ W_2 = \frac{2\pi \eta \omega_s^2 R_s^2 h_{sb}}{\delta_{ss}} \] (2)

where \( h_{sb} \) is the height of sub bearing, \( \delta_{ss} \) is the oil film thickness between the short part of shaft and the sub bearing.

4.3. Rotor end and Main Bearing End (Upper Thrust Bearing)
Since the gap and speed difference between rotor end and main bearing end, and the load is light. It is assumed that a better oil film can be formed here. So the fluid friction is expressed,

\[ W_3 = \frac{\pi \eta \omega_s^2 (R_t^2 - R_s^2)(R_t^2 + R_s^2)}{2\delta_{tm}} \] (3)

where \( R_t \) is the outer radius of rotor, \( \delta_{tm} \) is the oil film thickness of upper thrust bearing.
4.4. **Rotor end and Sub Bearing End (Lower Thrust Bearing)**
Due to the action of motor and shaft, lower thrust face forms a metal-to-metal contact with the cylinder. The formation of oil film is more difficult. So the boundary friction is introduced,

\[ W' = \frac{2\mu_{lb}\omega FR(R^1 - R^0)}{3(R^1 - R^0)} \] (4)

where \( \mu_{lb} \) is the friction coefficient of lower thrust bearing, \( F \) is the gravity of motor and shaft.

4.5. **Vane end and Bearing End**
Similar to chapter 4.3, the load between vane end and bearing end is slight, so fluid friction should be considered, that is

\[ W_5 = \frac{3\eta hL \nu_v^2}{\delta_{vm}} \] (5)

where \( b \) is the thickness of vane, \( L \) is the length of vane, \( \nu_v \) is the absolute speed of vane, \( \delta_{vm} \) is the oil film thickness between vane end and bearing end.

4.6. **Vane and Vane Grooves**
In course of motion, vanes form the line-contact with vane grooves due to gas force acting on both sides of the vane. It is difficult to form an oil film. So the calculation here is based on boundary friction, that is

\[ W_6 = 3(\mu_{sv}f_1\nu_v + \mu_{sv}f_2\nu_v) \] (6)

where \( \mu_{sv} \) is the friction coefficient between vane and vane grooves, \( f_1 \) is the force of the front part of vane groove on the vane, \( f_2 \) is the force of the rear of vane groove on the vane, \( \nu_v \) is the relative speed of vane and vane groove.

4.7. **Vane Tip and Inner Ring (Cylinder) Wall**
Due to centrifugal force and oil pressure which come from vane tail, vane tip will cling and relative slide along the inner ring wall. Thus, a line contact is formed between vane and inner ring (cylinder) wall. Similar to 4.6, the boundary friction is introduced,

\[ W_7 = 3\mu_{vi}f_{vt}\nu_t \] (7)

where \( \mu_{vi} \) is the friction coefficient between vane and inner ring, \( \nu_t \) is the relative speed of vane tip and inner ring wall, and \( f_{vt} \) is the gas force between the head and the rear of vane.

4.8. **Inner Ring End**
The compression mechanism has 2 rolling bearings, and each inner ring has 2 friction pairs. The calculation model of each friction pair are the same. And the inner ring has gap with cylinder or bearing, with lubricating oil and speed difference. It is assumed that a better oil film can be formed here, so the calculation is based on fluid friction. Formula for a single friction pair is shown as \( W' \), and the total friction loss is \( W_b \), which are shown as,
\[ W = \frac{\pi \eta (R_o^4 - R_i^4) \omega_i^2}{\delta_{im}} \]  
(8)

\[ W_s = 4 \frac{\pi \eta (R_o^4 - R_i^4) \omega_i^2}{\delta_{im}} \]  
(9)

where \( R_o \) is the outer radius of inner ring, \( R_i \) is the inner radius of inner ring, \( \omega_i \) is the angular velocity of inner ring, \( \delta_{im} \) is the oil film thickness between inner ring end and main bearing end.

### 4.9. Rolling Bearing

Because of the complex motion state of rolling bearings, their frictional torque calculations are currently based on empirical formulas. This paper refers to the calculation model of SKF (Sweden) [8]. The friction loss is made up of rolling friction torque \( (M_{rf}) \), sliding friction torque \( (M_{sf}) \), seal element friction torque \( (M_{sef}) \) and the frictional torque \( (M_{drag}) \), caused by drag loss, vortex, splash, etc.

\[ M = M_{rf} + M_{sf} + M_{sef} + M_{drag} \]  
(10)

\[ W_q = \frac{Mn}{9550} \]  
(11)

**Table 2.** Shows the related structure parameters in the equation and its variable or range.

| Parameters | Range  | Parameters | Range  | Parameters | Range  |
|------------|--------|------------|--------|------------|--------|
| \( \eta \) (mm\(^2\)/s) | 8.31   | \( R_o \) (mm) | 70     | L (mm)     | 21     |
| \( h_{sb} \) (mm)   | 60     | \( R_i \) (mm) | 60     | \( \mu_{th}, \mu_{vi}, \mu_{vs} \) | 0.1    |
| \( h_{mb} \) (mm)   | 40     | F (N)      | 20     | \( \delta_{vm}, \delta_{im}, \delta_{sm} \) (mm) | 0.01-0.02 |
| \( R_t, R_s \) (mm) | 21.5   | b (mm)     | 5      | \( \delta_{vm}, \delta_{im}, \delta_{sm} \) (mm) | 0.02-0.03 |

**Table 3.** Working conditions and related parameters

| Working condition | Related parameter |
|-------------------|-------------------|
| Refrigerant       | R410A             |
| Condensation Temp (°C) | 54.4          |
| Evaporation Temp (°C) | 7.2               |
| Displacement (cc)  | 42.8              |
| Cylinder Diameter (mm) | 60               |
| Rotation Speed (r/min) | 1800         |

According to the above formulae and parameters, the mechanical loss between HEVC and conventional vane compressor is calculated, as shown in Table 4.
Table 4. Comparison of the mechanical loss

| Friction pairs category | HEVC   | Conventional |
|-------------------------|--------|--------------|
| W_1 (W)                 | 24.28  |              |
| W_2 (W)                 | 14.78  |              |
| W_3 (W)                 | 8.36   |              |
| W_4 (W)                 | 1.73   |              |
| W_5 (W)                 | 2.34   |              |
| W_6 (W)                 | 122.55 |              |
| W_7 (W)                 | 40.39  | 272.11       |
| W_8 (W)                 | 40.90  | /            |
| W_9 (W)                 | 20.65  | /            |
| W (W)                   | 275.99 | 446.16       |

On the basis of other conditions remain the same, the calculation results indicated that the mechanical loss of the vane tip and the vane flanks are larger than other ones in CVC, accounting for 61% and 27.5% of the total mechanical loss, respectively. The vane tip mechanical loss of the HEVC decreased 231.72W (85.2%) compared with that of the CVC, and the total mechanical loss of compressor decreased 170.17W (38%).

5. Prototype verification experiment

The performance of CVC and HEVC were verified experimentally with the same structural parameters and working conditions to confirm the above conclusions.

The CVC and HEVC are designed according to the parameters in Table 2, respectively, and the difference of them is that the HEVC replaces the cylinder of CVC with the rolling bearing and cylinder. Figure 7 indicates two schematic diagrams showing the structure of the cylinder and the rolling bearing after dissecting the two prototypes.

![Figure 7](image.png)

Figure 7. The schematic diagram of cylinder and rolling bearing after dissecting the two prototypes.
After completing the production of the two prototypes, the performance tests of the two prototypes were carried out under the operating conditions of air conditioning shown in table 3. The test results are as follows.

Table 5. The measured data of CVC and HEVC and their difference.

| Type    | Refrigerating Capacity | Power  | COP   |
|---------|------------------------|--------|-------|
| CVC     | 6386.3                 | 2323.4 | 2.75  |
| HEVC    | 6653.4                 | 2163.3 | 3.08  |
| Difference | 267.1            | -160.1 | 0.3   |
| %       | 4.18%                  | -6.89% | 11.89%|

Figure 8. The theoretical and measured power difference of CVC and HEVC.

Because the premise of the calculation is that the friction pair operates with an ideal condition, but in the actual test, the internal system of compressor is complicated, with the temperature, pressure and refrigerant state affecting each other. Therefore, the theoretical calculation results and the actual test results have a tiny difference. But the two compressors are compared under the same premise, so the change trend is same.

From the front test results, under the design working conditions, the total power consumption of compressor decreased 160.1W (6.89%), and the total power consumption consists of motor power, shaft power, compression power and so on. The refrigerating capacity of compressor increased 267.1W (4.18%), so the COP increased by 11.89%.

6. Conclusion
This paper introduced a novel high efficiency vane compressor, and its characteristics have been analyzed. Mainly as follows:

Vane compressor has the advantages of high volumetric efficiency, small size and little torque ripple. Because of high sliding friction between vane and cylinder, mechanical friction loss of the conventional vane compressor is large, so COP of compressor is low, which limits the scope of its application.

The striking feature of HEVC is the use of rolling bearings, which can translate sliding friction between vane tip and cylinder inner wall into rolling friction. Due to the effect on performance, the
clearance $\delta_c$ and assembly process are the key research directions in the future.

Compared with conventional vane compressor, the theoretical mechanical loss of HEVC can be reduced by 170.17 W (38%), and the actual tested results indicated that the total power consumption of compressor decreased 160.1 W (6.89%), and the COP increased by 11.89%.

Finally in order to ensure the reliability of rolling bearings, it is necessary to design the lubricating oil circuit for rolling bearings.

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