1998

Forces Analysis of Rotary Vane Compressor for Automobile Air Conditioning System

L. Li  
Xi'an Jiaotong University

J. Hu  
Xi'an Jiaotong University

B. Guo  
Xi'an Jiaotong University

P. Shu  
Xi'an Jiaotong University

Follow this and additional works at: https://docs.lib.purdue.edu/icec

https://docs.lib.purdue.edu/icec/1274

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information. Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
FORCES ANALYSIS OF ROTARY VANE COMPRESSOR FOR AUTOMOBILE AIR CONDITIONING SYSTEM

Li Liansheng  Hu Jianhuan  Guo Bei  Shu Pengcheng

National Engineering Research Center of Fluid Machinery & Compressor Xi'an Jiaotong University Xi'an 710049, P. R. China Email Lils @ xjtu. edu. cn

ABSTRACT

Based on the rotary vane compressor for automobile air conditioning system, the numerical model for force analysis is built up and the corresponding computer programs is developed. The forces acting on vane and rotor are mainly analyzed in this paper. The acting forces varies with vane number are also discussed.

NOMENCLATURE

$F_{nc}$—Inertia force of implication
$F_{nc}$—Inertia force of centrifugation
$F_{nb}$—Cologorial inertia force
$F_r$—Gas pressure difference at two side of blade
$F_{sb}$—Back gas pressure at slide groove (slot)
$F_{sc}$—Contact Force between cylinder inner wall and slide head
$F_{sb}$—Contact Force between blade and rotor at back pressure groove.
$F_{sn}$—Contact force between blade and rotor at compression chamber
$F_{fc}$—Friction force correspondins to $F_{sc}$
$F_{fb}$—Friction force corresponding to $F_{sb}$
$F_{fa}$—Friction force corresponding to $F_{sn}$
$n$—Blade number
$L$—Axial length of rotor
Blade length at compression chamber
Sliding friction coefficient.

INTRODUCTION

For about half century, reciprocating compressor, including wobble structure, is used in the automobile air conditioning system. This type of compressor has larger size, more complicated structure, lower efficiency, and nosmooth starting.

In recent years, the rotary vane compressor is applied to the air conditioning system successly. See Fig 1, its structure features are summarized as follows:

1. The inner wall curve of cylinder is the continuous and smooth curve, which is made up of circle, trigonometric function, and so on.
2. Central line of rotor and cylinder should be coincide in theory.
3. Rotor and cylinder separate the high pressure district from the low pressure district at short axis direction.
4. Blade inclined improves its acting force and movement.
5. Lubricating oil, flowing at the pressure differential, has a better seal and lubrication effect.

The acting forces on the rotor and blade are analyzed in the paper. the effect of important parameter, blade number, on rotary vane compressor are also discussed. To analyze the complicated geometrical relations and dynamics feature, a common computering program is compiled.

FORCES ANALYSIS AND MATHEMATICS MODEL

Taking a blade as an object of study, as the compressor works, it rotates with the rotor. the blade slides at the pushment of the centrifugal force and the gas pressure at the back of the slide groove. Considering the acting force on two sides and tip of blade, the force model of blade is taken as a cantilever beam, seeing Fig 2.

If the weight of blade, the pressure varying at suction and discharge processes, and the effect of oil film on friction coefficient are negligible, the force equation of blade can be written in matrix.

\[
\begin{bmatrix}
\mu \cdot \cos \theta_{fsc} & \mu \cdot \cos \theta_{fma} & \cos \theta_{fsc} + \mu \cdot \cos \theta_{fma} \\
\text{sign}(F_{ab}) & \text{sign}(F_{ma}) & \sin \theta_{fsc} + \mu \cdot \sin \theta_{fma} \\
\text{sign}(F_{ab}) \cdot h & \text{sign}(F_{ma}) \cdot l & 0
\end{bmatrix}
\begin{bmatrix}
F_{ab}/\text{sign}(F_{ab}) \\
F_{ma}/\text{sign}(F_{ma}) \\
F_{sc}
\end{bmatrix}
= \begin{bmatrix}
b1 \\
b2 \\
b3
\end{bmatrix}
\]

where
\[ b_1 = - (F_{m_x} \cos \theta_{f_{m_x}} + F_{m_y} + F_{bb}) \]
\[ b_2 = - (F_{m_x} \sin \theta_{f_{m_x}} + F_{mb} + F_{r}) \]
\[ b_3 = - 0.5 (F_{m_x} \sin \theta_{f_{m_x}} \cdot \dot{h} + F_{mb} \cdot \dot{h} + F_{r} \cdot \dot{l}) \]

The symbol function is given a definition

\[
\text{sign}(x) = \begin{cases} 
  +1.0 & x \geq 0 \\
  -1.0 & x < 0 
\end{cases}
\]  

The solutions of equation (1) are the forces acting on slide blade. Because all the forces are the complex functions of crank angle \( \theta \), it is difficult to express each force in a simple relation. Here, each force is written as \( F(\theta) \), i.e. the function of \( \theta \). The force and moment acting on rotor can be got, provided the forces of blade and gas forces of rotor surface are done vector sum and taken the moment.

Radial force acting on rotor

\[
\vec{F}_r(\theta) = \sum_{i=1}^{s} \left[ \vec{F}_{ab}(\theta_i) + \vec{F}_{am}(\theta_i) + \vec{F}_{f}(\theta_i) + \vec{F}_{fn}(\theta_i) + \vec{F}_{bh}(\theta_i) \right] \\
+ \sum_{i=1}^{s} [p(\theta) L d \theta] 
\]  

where \( \theta_i \) is the force phase angle \( \theta_i = \theta + (i-1) \beta \), \( \alpha_n \) is the integration upper limit \( \alpha_n = \theta + i \cdot \beta - 0.5 \theta_c \), \( \beta \) is the stretch angle, \( \alpha_n \) is the lower limit of integration \( \alpha_n = \theta + (i - 1) \beta + 0.5 \theta_c \). \( p(\theta) \) is the pressure vector, \( \theta_c \) is the center angle of sliding blot.

Moment acting on the rotor

\[
\vec{M}(\theta) = \sum_{i=1}^{s} \left[ \vec{F}_{ab}(\theta_i) \times \vec{r}_{f_{ab}} + \vec{F}_{am}(\theta_i) \times \vec{r}_{f_{am}} + \vec{F}_{f}(\theta_i) \times \vec{r}_{f} + \vec{F}_{fn}(\theta_i) \times \vec{r}_{f_{fn}} \right] \\
+ \vec{F}_{fn}(\theta_i) \times \vec{r}_{f_{fn}} + \vec{F}_{bh}(\theta_i) \times \vec{r}_{f_{bh}} 
\]  

where \( \vec{r} \) is the force vector responding to the footnote.

**VARY REGULARITY OF FORCES IN THE COMPRESSION PROCESS**

Based on the above analysis, the forces of rotary vane compressor are calculated. The discharge capacity of compressor is 96ml/rev, the working condition being: compressor rotating speed 1800r.p.m, suction pressure 0.3MPa, discharge pressure 1.45MPa, back gas pressure of slide slot 1.16MPa, R134a, blade number 5. The calculating results of inertia force and contact force are separately shown in Fig 3 and Fig. 4.
Fig 3 shows that the inertia force of centrifugation of blade changes a little more at the starting and ending point as the curve of cylinder inner wall has turning points at these two place.

Fig. 4 shows that the contact force $F_{oc}$ between blade and rotor at compression chamber gets a greatest value near the discharge starting, that the contact force $F_{oc}$ between cylinder inner wall and the slide head may be less than nought. If the latter situation occurs, the blade will depart from the cylinder wall and the compressed gas may leak from the high pressure district to the lower pressure district.

**EFFECT OF BLADE NUMBER ON THE FORCES ACTING ON THE BLADE AND ROTOR**

Blade number is a important structure parameter of rotary vane compressor. The vary of acting forces and moment with blade number are shown in Fig 5~Fig 8. It is shown in Fig 5 that the gas force $F_1$ of blade sides at compression process will decrease as the blade number increases.

The contact force $F_{oc}$ of blade head will vary with blade number only after the discharge process starts. When the blade number gets greater, the gas pressure difference of blade at the adjacent basic volume will lessen. It can be clearly seen in Fig 6 that if the blade number $n=6$, the contact force $F_{oc} \leq 0$ may appear at the second half of the discharge process, resulting in gas leakage.

If the blade number $n$ is the odd number, the vary cycle of moment is equal to $(\pi/n)$, while that of the radial force of rotor is equal to $(2\pi/n)$, see Fig 7. Although the radial force and moment acting on rotor vary with the crank angle $\theta$, the fluctuation is little. Less moment fluctuation is beneficial to driving mechanism.

If the blade number $n$ is even number, the vary cycle of moment is $(2\pi/n)$, while that of radial force is a constant owing to the symmetry of structure and compression process. Fig 8 shows that the fluctuation of moment is greater as the blade number is even than is odd.

**CONCLUSIONS**

To express the forces and moment acting on rotary vane compressor in a simple function is very difficult. The contact force between blade head and cylinder inner wall is an important force. The less force may result in the gas leakage at compression pro-
cess, while the greater force needs more friction power consume. The bearing load and driving force are influenced by the vary of the radial force and moment of rotor.

Analysis above shows that the forces and moment relate to the blade number. What blade number is selected should consider concrete design desire. Generally speaking, the blade number $n=3\sim6$ is appropriate.

REFERENCES

2. Deng Dinguo. Shu Pengcheng. Rotary type of compressor, Machinery industry publishing house, 1982, P. R. China

Fig 1 Scheme of rotary vane compressor

Fig 2 Scheme of acting forces

Fig 3 Inertia force Vs the crank angle

Fig 4 Contact force Vs the crank angle
Fig 5  Gas pressure Vs the crank angle

Fig 6  Contact force of blade head Vs the crank angle

Fig 7  Radial force acting on rotor Vs the crank angle

Fig 8  Moment acting on rotor Vs the crank angle