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by

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ABSTRACT

The present study examines the dynamic behavior of a variable displacement wobble plate compressor which makes it possible to control the cooling capacity continuously. The continuous cooling capacity control is achieved by the complicated motion of the piston, the piston rod, the wobble plate, the rotating journal and the drive shaft. In order to analyze the complicated motion of each moving element, the inertial forces and moments of the moving elements are calculated first. Subsequently, the constraint force at each pair of compressor elements and the frictional forces given by Coulomb's law of friction are calculated. The unknown frictional coefficient was determined on the basis of experimental data on the cylinder pressure and the driving power. It can be concluded that the inertial and frictional forces have a significant effect upon the feasibility to control the cooling capacity, and upon the quiet operation and durability of compressor parts at high operating speeds.

INTRODUCTION

Variable displacement wobble plate compressors have brought on a revolution in automobile air conditioning since commercial production [see Ref. 1]. The refrigerant flow rate discharged from the compressor is controlled by continuously changing the piston displacement, thus resulting in wide-range continuous control of the cooling capacity and in high efficiency. The variable displacement wobble plate compressors are now widely considered to be the best choice for automobile air conditioning with its stringent requirements for comfort and reduced energy consumption.

The continuous control of the cooling capacity is achieved, however, by the complicated motion of many compressor elements, such as the piston, the piston rod, the wobble plate, the rotating journal and the drive shaft. Subsequently, questions concerning the quiet operation and durability of these parts at high operating speeds arise. The complicated motion of variable displacement wobble plate compressors and the dynamic balancing of the rotating shaft have been treated in several previous studies [2, 3] from the kinematics viewpoint. However, the inertial forces and moments of the moving elements and also the frictional force at each pair of compressor elements have not been taken into account. As the operating speed of the compressor increases, the effects of the inertial and frictional forces become increasingly significant.

The present study examines the dynamic behavior of a 6-cylinder variable displacement wobble plate compressor. The complicated motion of each moving element of the compressor is precisely analyzed, from both the kinematics and dynamics viewpoints, and thus the inertial forces and moments of the moving elements and the frictional forces acting on the element pairs are calculated. Based on the calculated results, the effects of the inertial and frictional forces upon the feasibility to control the cooling capacity, and upon the quiet operation and durability of compressor parts at high operating speeds, are examined. Furthermore, the power losses due to friction are examined and design criteria for quiet operation and high speed durability of compressor parts are obtained.

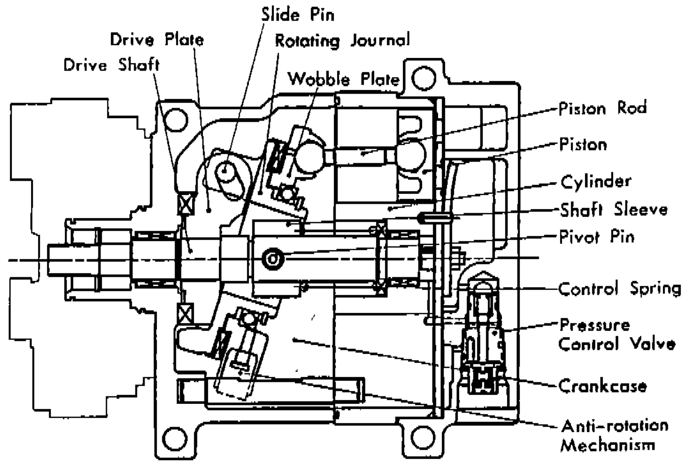


Fig. 1 Mechanism of variable displacement compressor

MECHANISM FOR VARIABLE PISTON DISPLACEMENT

The vertical cross-sectional view of a 6-cylinder variable displacement wobble plate compressor is shown in Fig. 1. The compression mechanism is composed of many compressor elements, such as the drive shaft, the drive plate, the rotating journal, the wobble plate, the piston rods and the pistons. The rotating journal driven by the drive plate rotates but the wobble plate on the rotating journal undergoes a wobbling motion, since its rotation around the drive shaft is constrained by an anti-rotation mechanism with five degrees of freedom. Thus, the piston connected to the wobble plate through the double-ended ball piston rod reciprocates in the cylinder. The variable piston displacement is achieved by the kinematics of the rotating journal. The rotating journal can rotate about the pivot pin fixed on the shaft sleeve, but its nutation angle and position are controlled by a slide pin which is located in the guide slot. As the nutation angle decreases, the shaft sleeve slides up along the drive shaft to keep the piston head clearance at a constant value for any nutation angle, and thus the piston displacement is decreased. It is a matter of course that the nutation angle of the rotating journal is determined by the equilibrium of the moments around the pivot pin, which act on the rotating journal. The major factor which determines the nutation angle is the gas force acting on each piston. Therefore, the piston displacement can be controlled by adjusting the gas pressure in the crankcase. A pressure control valve is attached between the discharge chamber and the crankcase. At full load, the pressure control valve closes and the crankcase pressure becomes equal to the suction pressure. Thus, the maximum nutation angle of the rotating journal is maintained. At partial load, the pressure control valve opens and the crankcase pressure is increased. Thus, the nutation angle is decreased. In this way, the pressure control valve regulates the crankcase pressure, resulting in the continuous control of the piston displacement. Thus, the wide-range continuous control of the cooling capacity as well as high efficiency can be obtained.

KINEMATICS OF VARIABLE DISPLACEMENT COMPRESSOR

Before the dynamic behavior of the variable displacement compressor can be examined, the kinematics of each moving element must first be analyzed. For this purpose, the rectangular coordinates and variables shown in Fig. 2 are introduced. The static coordinates are represented by x, y and z, where the z axis coincides with the drive shaft center. The x_m axis represents the pivot pin center of the shaft sleeve. Therefore, the x_m-y_m coordinates rotate with the drive shaft. The rotating angle of the drive shaft is represented by θ. When rotating the x_m-y_m-z coordinates around the x_m-axis by the nutation angle α, the x_m-y_α-z_α coordinates fixed on the rotating journal are obtained. For example, the point P_0 on the x-y plane moves to the point P on the x_m-y_m plane. The point P has to move to the point S, since the rotation of the wobble plate is controlled by the anti-rotation mechanism.

The piston rod TS is shown in Fig. 3. The position of a cylinder is represented by the radius r_p and the angle θ_n from the x-axis. The subscript n takes the integers from 1

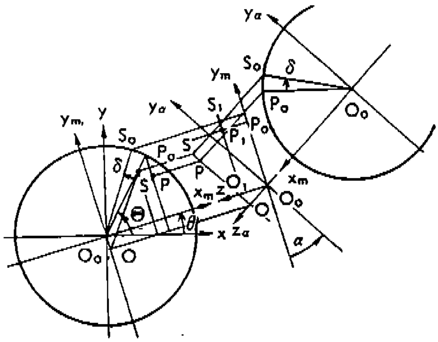


Fig.2 Coodinate and variables

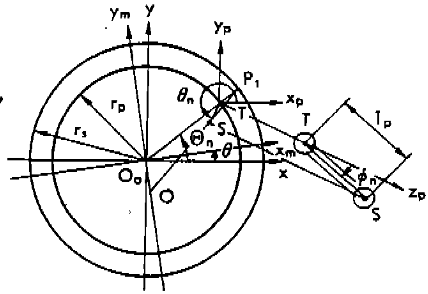


Fig.3 Variables for the piston rod

to 6. The position of the piston rod center S_n on the wobble plate side can be analyzed on the basis of the coordinate transformations shown in Fig.2, resulting in the following x-y-z coordinates:

$$\begin{bmatrix} x_{Sn} \\ y_{Sn} \\ z_{Sn} \end{bmatrix} = A(\theta) \begin{bmatrix} r_s \cos(\Theta_n - \theta + \delta) \\ r_s \sin(\Theta_n - \theta + \delta)\cos\alpha - h_s \sin\alpha \\ r_s \sin(\Theta_n - \theta + \delta)\sin\alpha + h_s \cos\alpha \end{bmatrix} \tag{1}$$

where r_s and h_s represent the radius from the origin O_0 and the height from the x_m-y_α plane, respectively, of the piston rod center S_n. $A(\theta)$ is a coordinate transformation matrix and δ is a constraint angle correcting the rotation of the wobble plate controlled by the anti-rotation mechanism, given by the following expressions:

$$A(\theta) = \begin{bmatrix} \cos\alpha & -\sin\alpha & 0 \\ \sin\alpha & \cos\alpha & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{2}$$

$$\delta = \tan^{-1}\left[\frac{\tan(\Theta_B - \theta)}{\cos\alpha} + \frac{h_B \tan\alpha}{r_B \cos(\Theta_B - \theta)}\right] - \Theta_B + \theta \tag{3}$$

The piston rod center T_n on the piston side is represented by the following x-y-z coordinates:

$$x_{Tn} = r_p \cos\Theta_n, \quad y_{Tn} = r_p \sin\Theta_n, \quad z_{Tn} = z_{Sn} + l_p \cos\phi_n \tag{4}$$

where ϕ_n is the rotation angle of the piston rod from the z axis and is given by the following expression:

$$\phi_n = \sin^{-1}\left[\{(x_{Sn} - x_{Tn})^2 + (y_{Sn} - y_{Tn})^2\}^{0.5} / l_p\right] \tag{5}$$

INERTIAL FORCES AND MOMENTS

The inertial forces and moments of the moving elements are derived as follows.

Piston Rod

Assuming a point in the piston rod represented by the (x_n, y_n, z_n) coordinates, the inertial moments M_{pzn} around the z_p axis and $M_{\theta\phi n}$ around the orthogonal axis of the piston rod are given by the following integral expressions:

$$M_{pzn} = \int\{\ddot{x}_n (y_n - y_{Tn}) - (x_n - x_{Tn})\ddot{y}_n\} dm$$
$$M_{\theta\phi n} = \int r(\ddot{x}_n \cos\theta_n \cos\phi_n + \ddot{y}_n \sin\theta_n \cos\phi_n + \ddot{z}_n \sin\phi_n) dm \tag{6}$$

where dm is a small mass in the piston rod, r the distance from the piston rod center T_n, and θ_n the rotation angle of the piston rod from the x_p axis. The integral sign represents a definite integral for the whole piston rod. θ_n is given by the following expression:

$$\theta_n = \tan^{-1}\{(y_{Sn} - y_{Tn}) / (x_{Sn} - x_{Tn})\} \tag{7}$$

Wobble Plate

Assuming a point in the wobble plate represented by the (x,y,z) coordinate, the inertial forces in the x, y and z directions are given by the following integral expressions, respectively:

$$F_{Wx} = -\int \ddot{x}\, dm, \quad F_{Wy} = -\int \ddot{y}\, dm, \quad F_{Wz} = -\int \ddot{z}\, dm \tag{8}$$

The inertial moments around the x, y and z axes are given by the following integral expressions, respectively:

$$M_{Wx} = \int(\ddot{y}\, z - y\, \ddot{z})dm, \quad M_{Wy} = \int(\ddot{z}\, x - z\, \ddot{x})dm, \quad M_{Wz} = \int(\ddot{x}\, y - x\ddot{y})dm \tag{9}$$

Rotating Journal

Assuming a point in the rotating journal represented by the (x,y,z) coordinates, the inertial forces and moments of the rotating journal are given by the same integral form of the expressions in (8) and (9):

$$F_{Tx} = -\int \ddot{x}\, dm, \quad F_{Ty} = -\int \ddot{y}\, dm, \quad F_{Tz} = -\int \ddot{z}\, dm \tag{10}$$

$$M_{Tx} = \int \ddot{y}\, z\, dm, \quad M_{Ty} = -\int z\, \ddot{x}\, dm, \quad M_{Tz} = \int(\ddot{x}y - x\ddot{y})dm \tag{11}$$

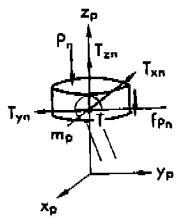


Fig.4 Forces on the piston

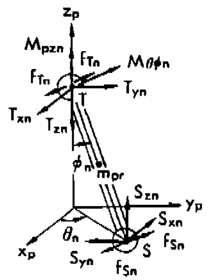


Fig.5 Forces and moments on the piston rod

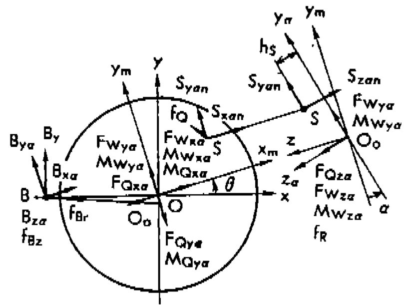


Fig.6 Forces and moments on the wobble plate

EQUATIONS OF MOTION

The equations of motion of the moving elements are derived from the equilibrium of forces and moments acting on each moving element.

Piston and Piston Rod

Fig.4 shows the forces acting on the piston. The reaction forces from the piston rod are represented by T_{xn}, T_{yn} and T_{zn} , and the resultant gas force by P_n. The reaction forces T_{xn} and T_{yn} balance with the reaction forces from the cylinder. Assuming Coulomb's law of friction, the frictional force f_{Pn} between the piston and the cylinder wall is represented by the following expression:

$$f_{Pn} = \mu_P\ (T_{xn}^2 + T_{yn}^2)^{0.5} \tag{12}$$

The equation of motion of the piston is given by the following expression:

$$-m_P \ddot{z}_{Tn} - P_n - sgn(\dot{z}_{Tn}) f_{Pn} + T_{zn} = 0 \tag{13}$$

Fig.5 shows the forces and moments acting on the piston rod. The reaction forces from the piston are represented by T_{xn} and T_{yn}, and those from the wobble plate by S_{xn}, S_{yn} and S_{zn}. The frictional moments represented by f_{Tn} and f_{Sn} arise at the ball joints T and S, respectively. The inertial moments M_{Pzn} and $M_{\theta\phi n}$ are given by (6). When these forces and moments are at equilibrium, the 5 equations of motion with respect to x_p, y_p, z_p, θ_n and ϕ_n can be derived.

Wobble Plate

Fig.6 shows the forces acting on the wobble plate. The reaction forces from the piston rod are represented by $S_{x\alpha n}$, $S_{y\alpha n}$ and $S_{z\alpha n}$, and the reaction forces and moments

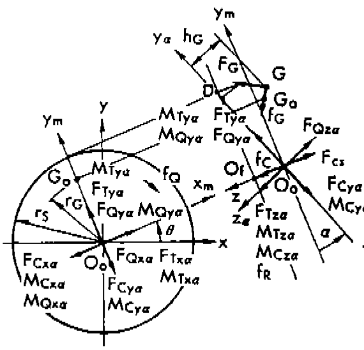


Fig.7 Forces and moments on the rotating journal

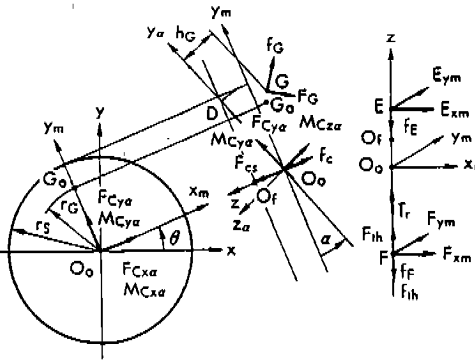


Fig.8 Forces and moments on the drive shaft

from the rotating journal by $F_{Qx\alpha}$, $F_{Qy\alpha}$, $F_{Qz\alpha}$, $M_{Qx\alpha}$ and $M_{Qy\alpha}$, respectively. The inertial forces and moments are represented by $F_{Wx\alpha}$, $F_{Wy\alpha}$, $F_{Wz\alpha}$, $M_{Wx\alpha}$, $M_{Wy\alpha}$ and $M_{Wz\alpha}$, and can be derived by applying the coordinate transformations to (8) and (9). The point B represents the joint of the wobble plate and the anti-rotation mechanism. The frictional forces from the anti-rotation mechanism are represented by f_{Bz} in the z direction and f_{Br} in the z_α direction. The frictional moment from the thrust bearing is represented by f_Q and that from the radial bearing by f_R. When these forces and moments are at equilibrium, the 6 equations of motion in the x_m, y_α, z_α directions and around each axis can be derived.

Rotating Journal

Fig.7 shows the forces and moments acting on the rotating journal. The nutation of the rotating journal is controlled by the slide pin (G) and the shaft sleeve pin (x_m-axis). The reaction force and frictional force at the point G from the drive shaft are represented by F_G and f_G, respectively. The reaction forces and moments from the wobble plate are represented by $F_{Qx\alpha}$, $F_{Qy\alpha}$, $F_{Qz\alpha}$, $M_{Qx\alpha}$, $M_{Qy\alpha}$ and those from the drive shaft by $F_{Cx\alpha}$, $F_{Cy\alpha}$, $M_{Cx\alpha}$, $M_{Cy\alpha}$ and $M_{Cz\alpha}$. The inertial forces and moments are represented by $F_{Tx\alpha}$, $F_{Ty\alpha}$, $F_{Tz\alpha}$, $M_{Tx\alpha}$, $M_{Ty\alpha}$ and $M_{Tz\alpha}$, and can be derived by applying the coordinate transformations to (10) and (11). f_C is the frictional force between the shaft sleeve and the drive shaft, and F_{cs} the reaction force of the spring as it presses the shaft sleeve down. When these forces and moments are at equilibrium, the 6 equations of motion in the x_m, y_α, z_α directions and around each axis can be derived. For instance, the equilibrium equation of the moment around the x_m axis is given by the following expression:

$$M_{Tx\alpha} - M_{Qx\alpha} \cdot M_{Cx\alpha} + (F_{Gy\alpha} - f_{Gy\alpha}) + (F_{Gy\alpha} + f_{Gy\alpha}) r_G = 0 \qquad (14)$$

where the moment $M_{Cx\alpha}$ is introduced when the nutation angle reaches to a maximum value.

Drive Shaft

Fig.8 shows the forces and moments acting on the drive shaft. The drive shaft is supported by the two radial bearings denoted by E and F. The reaction forces from the bearing E are represented by E_{xm} and E_{ym}, and those from the bearing F by F_{xm} and F_{ym}. The frictional moments from the two radial bearings are represented by f_E and f_F, respectively. The reaction force and frictional moment from the thrust bearing are represented by F_{th} and f_{th}, respectively. Therefore, one can obtain the equilibrium equations of the forces in the x_m, y_m, z directions and the moments around the x_m, y_m, z axes exerted on the drive shaft. When these forces and moments are at equilibrium, the equations of motion of the drive shaft can be derived. The rotation of the drive shaft is subject to the following equation of motion:

$$T_r = I_c \ddot{\theta} - M_{Cy\alpha} \sin\alpha + M_{Cz\alpha} \cos\alpha + f_E + f_F + f_{th} \qquad (15)$$

where T_r represents the driving torque and I_c the moment of inertial of the drive shaft.

CALCULATED RESULTS

In order to evaluate the total dynamic behavior of the variable displacement compressor, the equation of motion (15) of the rotating drive shaft must first be calculated. However, from the basic standpoint of examining the effects of the inertial forces upon the reaction forces, the reaction moments and the frictional forces, they were calculated for various constant operating speeds of the drive shaft. For given data concerning the mechanical constants, the dimensions and the operating parameters, the variables representing the motion of the moving elements were calculated first, and subsequently, the time-differential terms of the variables were calculated. Thus, the inertial forces and the moments of inertial of the moving elements were calculated. By using the calculated results, the reaction forces and moments acting on the compressor elements were calculated, and finally, the driving torque and the gas pressure necessary for cooling capacity control were examined. The unknown frictional coefficient was determined on the basis of experimental data on the cylinder pressure and the driving power.

Motion of the Piston Rod

Fig.9 shows the calculated results for the constraint angle δ given by (3). The angle δ fluctuates twice for one revolution of the drive shaft. This means that any point on the wobble plate follows two elliptical orbits in each revolution. The calculated results for the rotation angles θ_n and ϕ_n (see Figs.3 and 5) of the 6 piston rods are shown in Figs.10 and 11, respectively. Their motions are very complicated and, moreover, each motion is different. As a result of the complicated motion of the piston rod, the moments of inertial (M_{Pzn} and $M_{\theta\phi n}$) of the piston rod, shown in Figs.12 and 13, arise. Depending upon the position of the piston rod, the moments of inertial of the piston rods are fairly different in amplitude and mode.

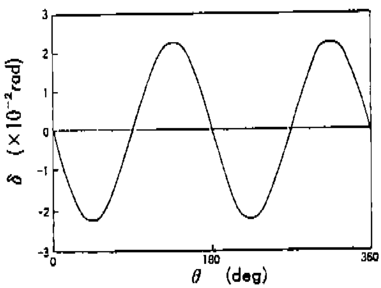


Fig.9 Constraint angle δ

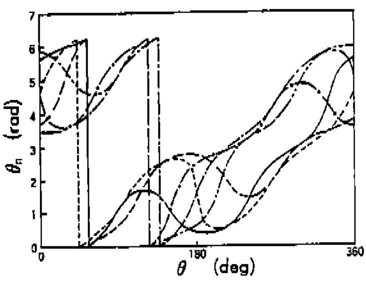
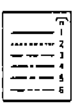


Fig.10 Rotating angle θ



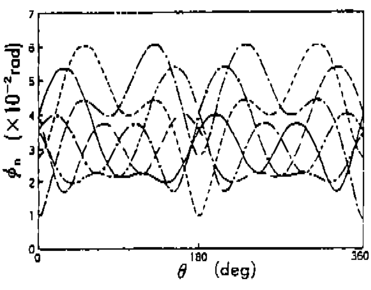


Fig.11 Rotating angle ϕ

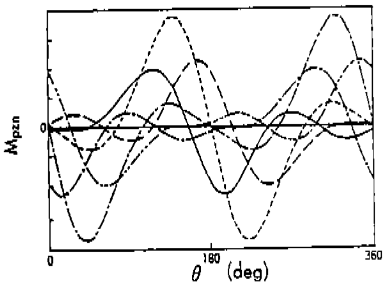
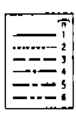


Fig.12 Inertial moment M_{pzn}



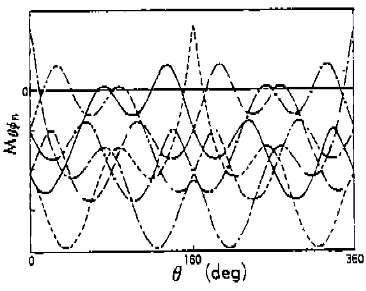


Fig.13 Inertial moment $M_{\theta\phi n}$

Reaction Force on the Piston Rod

The reaction force T_{zn} (see Fig.5) acting on the piston rod is shown in Fig.14. The solid line represents the reaction force and the dotted line the reaction force component due to the gas force. Therefore, the difference between the solid and dotted lines suggests the magnitude of the effect of the inertial and frictional forces upon the reaction force. Due to the inertial effects, a negative reaction force appears during the suction process. As the operating speed of the the drive shaft increases, the magnitude of this negative reaction force increases. It can be concluded that alternating compression and tensile forces act on the piston rod and thus durability will be severely compromised at the piston-rod ball joints, especially at high operating speeds.

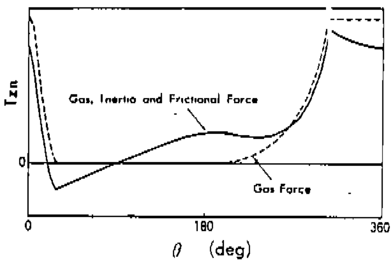


Fig.14 Reaction force on the piston rod

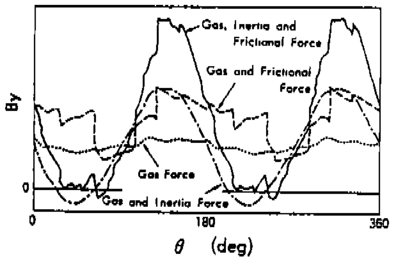


Fig.15 Reaction force on the anti-rotation mechanism

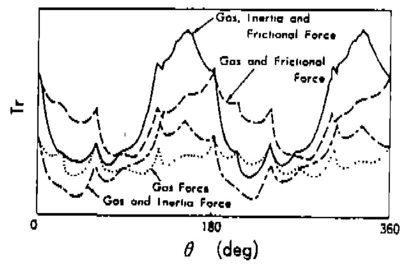


Fig.16 Drive torque

Constraint Force on the Anti-rotation Mechanism.

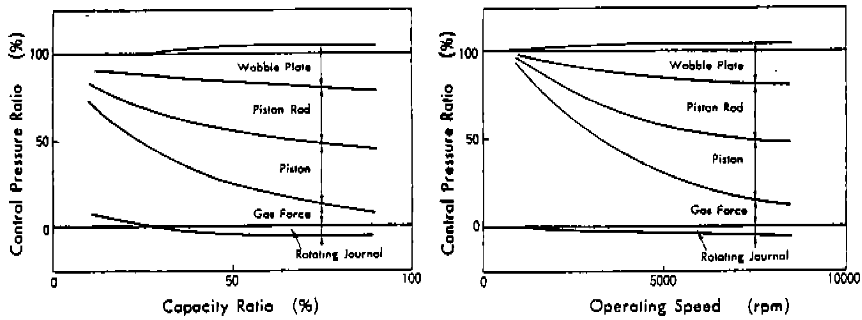
The reaction force B_y acting on the anti-rotation mechanism is shown by the solid line in Fig.15. In order to examine the effects of the inertial and frictional forces upon the reaction force, the reaction force component due to the gas force is shown by the dotted line, the component due to the frictional and gas forces by the broken line and the component due to the inertial and gas forces by the chain line. The broken line represents a step curve, due to a rapid change in direction of the frictional forces. By comparing the calculated results in this way, it can be seen that the effects of the inertial and frictional forces upon the resultant reaction force at the anti-rotation mechanism is fairly large and that the gas force has only a slight effect.

Driving Torque

The calculated results of the driving torque T_r given by (15) are shown by the solid line in Fig.16. In order to examine the effects of the inertial and frictional forces, the driving torque components are shown in the same way as in Fig.15. This figure also shows that the effects of the inertial and frictional forces upon the driving torque are great.

Control Pressure

In order to control the piston displacement for various operating speeds of the compressor, the gas pressure difference (control pressure) between the crankcase and the suction chamber can be used for regulating the crankcase pressure. The control pressure can be determined from (14). This means that the control pressure depends not only upon the gas force but upon the inertial and frictional forces. It is interesting enough to examine the effects of the inertial and frictional forces upon the control pressure. Here, the effects of the inertial forces of the moving elements were calculated, as shown in Figs.17a and b. It can be concluded from these figures that as the cooling capacity or the operating speed increases, the inertial forces of the moving elements have a greater effect upon the feasibility of controlling the cooling capacity. However, the impact of these inertial effects upon the control pressure can be decreased by the careful design of each moving element.



(a) Control pressure ratio vs capacity ratio

(b) Control pressure ratio vs operating speed

Fig. 17 Effect of inertial forces upon the control pressure

CONCLUSIONS

This study examined the dynamic behavior of a 6-cylinder variable displacement wobble plate compressor. The complicated motion of each moving element of the compressor was precisely analyzed. By taking into account the inertial forces and moments of the moving elements, the equation of motion of each element is derived, and the motions of the moving elements, and the forces and moments acting on the compressor elements are calculated. The conclusions can be summarized as follows:

(1) When the wobble plate is constrained by the anti-rotation mechanism, each of the six piston rods has a complicated motion. As a result, the inertial forces and moments become complicated. Moreover, as the direction of the relative motion at each pair of elements changes, the direction of the frictional force changes rapidly. Mainly as a result of these two factors, the constraint forces at each element and the driving torque also fluctuate complicated manner of a step. This phenomenon causes the control pressure for the cooling capacity to be unstable;

(2) As the cooling capacity or the operating speed increases,the inertial effects of the moving elements upon the constraint forces become increasingly important. Especially the inertial forces of the piston and the piston rod during high speed operation have a significant effect upon the feasibility to control the cooling capacity, upon the increase in driving power and the driving variation;

(3) By analyzing the dynamic behavior of each moving element, design criteria were obtained for quiet operation and durability of the compressor parts at high operating speeds.

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APPEMDIX: NOMENCLATURE

l_p : piston rod length
B_y : reaction force on anti-rotation mechanism
E_{xm}, E_{ym}, F_{xm}, F_{ym} : constraint forces on drive shaft
f_{Bz}, f_{Br} : frictional forces on anti-rotation mechanism
fc : frictional force on rotating journal and shaft
f_E, f_F : frictional moments at radial bearings of drive shaft
f_G : frictional force on slide pin
f_P : frictional force on piston
f_Q, f_R : frictional moments at thrust and radial bearings between wobble plate rotating journal
f_{th} : frictional moment at thrust bearing of drive shaft
f_{Tn}, f_{Sn} : frictional moments on piston rod joints
F_{cs} : spring force
$F_{Cx\alpha}$, $F_{Cy\alpha}$, $M_{Cy\alpha}$, $M_{Cz\alpha}$: reaction forces and moments on rotating journal and drive shaft
F_G : reaction force on slide pin
$F_{Qx\alpha}$, $F_{Qy\alpha}$, $F_{Qz\alpha}$, $M_{Qx\alpha}$, $M_{Qy\alpha}$: reaction forces and moments on wobble plate and rotating journal
F_{th} : reaction force on drive shaft from thrust bearing
F_{Tx}, F_{Ty}, F_{Tz}, M_{Tx}, M_{Tz} : inertial forces and moments of rotating journal
F_{Wx}, F_{Wy}, F_{Wz}, M_{Wy}, M_{Wz} : inertial forces and moments of wobble plate
I_c : moment of inertial of drive shaft
m_p : mass of piston
m_{pr} : mass of piston rod
M_{Pzn}, $M_{\theta\phi n}$: inertial moments piston rod
n : cylinder number
P_n : gas force on piston
r_P : arranged radius of piston center
r_S, h_S : arranged radius and height of wobble plate/piston rod joint
r_B, h_B : arranged radius and height of anti-rotation mechanism center
r_G, h_G : arranged radius and height of slide pin center
S_{xn}, S_{yn}, S_{zn} : reaction forces on piston rod and wobble plate
T_r : driving torque
T_{xn}, T_{yn}, T_{zn}: reaction forces on piston
x, y, z : orthogonal coordinate
x_m, y_m, z: rotating orthogonal coordinate
x_m, y_α, z_α: rotating orthogonal coordinate fixed on rotating journal
x_P, y_P, z_P: orthogonal coordinate fixed on each cylinder
α : nutation angle of rotating journal
δ : constraint angle of wobble plate
θ : rotating angle of drive shaft
θ_n : rotating angle of piston rod form x_P axis
Θ_n : angular position of each cylinder bore center
Θ_B : angular position of anti-rotation mechanism
μ_P : frictional coefficient at piston
ϕ_n : rotating angle of piston rod from z_P axis