Application of Phase-Angled Counterweights for a Scroll Compressor*

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This paper discusses the capability of phase angled counterweights to improve the dynamic characteristics of a radially compliant scroll compressors. Based on force and moment balances on crankshaft and compressor frame, mathematical formulation of shaft bearing loads, unbalanced force and moment acting on the frame has been made in terms of counterweights design parameters such as masses and phase angles of the upper and lower counterweights. Any two of crankshaft main bearing load, sub bearing load, unbalanced body force, and overturning moment can be controlled by the counterweights design parameters.

Key Words: Scroll Compressor, Counterweights, Bearing Load, Unbalance Force and Moment

1. Introduction

Due to increasing concern for energy-saving and comfortableness, the compressor of high efficiency and low noise/vibration has been demanded. Furthermore, to improve the annual performance factor of heat-pump air conditioners, capacity control by means of motor rotational speed variation is becoming widespread. The scroll compressor satisfying these requirements has been highlighted. The vertical cross-sectional view of a scroll compressors shown in Fig. 1.

Scroll compressor is consisted of a motor, a shaft, fixed and orbiting scrolls, and main/lower frames that support the shaft. Low noise/vibration level and high efficiency at all rotational speed are important consideration for scroll compressors. Previous studies⁽¹⁾⁻⁽³⁾ have discussed the dynamic behavior of constant and variable speed scroll compressor. To dynamically

balance a scroll, two counterweights are typically included on the drive shaft as shown in Fig. 1. Conventional counterweights design for scroll compressors typically employs two counterweights: upper one is circumferentially positioned 180 deg. from the crank pin and lower one on the same side of the pin. Sizing of the counterweights is made to satisfy the balance of centrifugal forces of moving members and the moment balance due to these forces.

Nieter and DeBlois⁽⁴⁾ suggested an optional method of designing counterweights which theoretically

Fig. 1 Compressor cross section

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Fig. 3 Forces acting on the oldham ring

Fig. 4 Forces acting on the slide bush

$$(F_{rg} + F_{rs})\left(\delta + \frac{h}{2}\right) - F_{osc}l_c + F_a \frac{r_s}{2} - F_d r_x - F_{sbr} \frac{l}{2} - \frac{b}{2} \sin \theta (F_2 - F_1) = 0 \quad (4)$$

$$\frac{r_s}{2} F_{tg} + L_{sb} + F_1(-(R_{ox} - r_s \cos \theta)) + F_2(-(R_{ox} + r_s \cos \theta)) = 0 \quad (5)$$

A model of the oldham ring which performs reciprocating motion is shown in Fig. 3. The motion of the oldham ring is described as:

$$F_3 - F_4 = 0 \quad (6)$$

$$F_1 - F_2 - m_o \ddot{y} = 0 \quad (7)$$

$$R_{ox}F_1 + R_{ox}F_2 - R_{oy}F_3 - R_{oy}F_4 = 0 \quad (8)$$

Figure 4 shows the forces on the slider bush which is inserted between the orbiting scroll hub and the crank pin for radial compliance. The equations of motion for the slider bush are given by the followings:

$$F_{sbr} - F_{bl} \sin \alpha + F_{sbc} = 0 \quad (9)$$

$$F_{sbt} - F_{bl} \cos \alpha + F_{sbi} = 0 \quad (10)$$

$$-L_{sb} - F_{bl}(r_s \cos \alpha - d) = 0 \quad (11)$$

Various forces and their acting points on the crankshaft are shown in Fig. 5. The equations of equilibrium in forces and moments are obtained as follows:

$$(F_{cpr} + F_{cpc}) - F_{mjr} + F_{sjr} - F_{uwr} - F_{csc} + F_{dwr} = 0 \quad (12)$$

$$-F_{cpt} + F_{mjt} - F_{sjt} + F_{uwt} - F_{dwt} = 0 \quad (13)$$

$$-F_{cpt}l_{cp} + F_{sjt}l_{sj} - F_{uwt}l_{uw} + F_{dwt}l_{dw} = 0 \quad (14)$$

(a) Radial forces (b) Tangential forces

Fig. 5 Forces acting on crankshaft

(a) Radial forces (b) Tangential forces

Fig. 6 Forces acting on compressor frame

$$-(F_{cpr} + F_{cpc})l_{cp} + F_{sjr}l_{sj} - F_{uwr}l_{uw} - l_{cs}F_{csc} + F_{dwr}l_{dw} = 0 \quad (15)$$

Figure 6 shows reactions from various members to the frame. F_{rg} , F_{rs} and F_{tg} are transmitted from orbiting scroll to the frame via fixed scroll which is rigidly attached to the frame, and F_d is to the thrust surface.

The Oldham ring key forces are given by: $F_{3r} = F_3 \cos \theta$, $F_{4r} = F_4 \cos \theta$, $F_{3t} = F_3 \sin \theta$, $F_{4t} = F_4 \sin \theta$. Resultant of these forces is the unbalanced force acting on the frame, F_{Body} . These unbalanced forces are decomposed into the radial and tangential components as in the Eqs. (13) and (14), respectively.

$$F_{rB} = F_{rg} + F_{rs} + F_{3r} - F_{4r} + F_{mjr} - F_{sjr} \quad (16)$$

$$F_{tB} = F_{tg} - F_{3t} + F_{4t} - F_{mjt} + F_{sjt} \quad (17)$$

The overturning moment takes place due to the unbalanced forces. With the moment center at the sub bearing position, components of overturning moment acting on the frame are obtained by the Eqs. (15) and (16).

$$M_{tB} = -(l_{sj} + l_h)(F_{rs} + F_{rg}) + (r_x - r_s)F_a - l_{sj}F_{mjr} - l_{or}(F_{3r} - F_{4r}) \quad (18)$$

$$M_{rB} = -(l_{sj} + l_h)F_{tg} + r_y F_a + l_{sj}F_{mjt} - l_{or}(F_{3t} - F_{4t}) \quad (19)$$

2.2 Counterweights design

Figure 7 shows a schematic of the phase-angled

Fig. 8 (a) Effects of upper counterweight phase angle on main bearing load components

Fig. 8 (b) Effects of upper counterweight phase angle on lower counterweight phase angle

Fig. 8 (c) Effects of upper counterweight phase angle on masses of counterweights

Minima of both F_{sj} and M_{Body} take place at about the same β_1 , the locations of which are marked with diamond marks in the figures. Figure 10 shows that decrease in F_{mj} is accompanied by increase in M_{Body} , and that there is an optimum value of F_{mj} for minimum F_{sj} .

Shaft bearing loads, unbalanced body force, and overturning moment of the present method and those of the conventional method are compared at various compressor speeds in Figs. 11(a), (b), (c), respec-

Fig. 9 (a) Effects of upper counterweight phase angle on sub bearing load at $F_{mi}=2800$ N

Fig. 9 (b) Effects of upper counterweight on overturning moment at $F_{mi}=2800$ N

Fig. 10 Variations of sub bearing load and overturning moment with main bearing load

tively. In the figures, 'A' and 'B' represent the conditions specified in Fig. 10, and 'C' stands for the conventional method. The trend of monotonic increase in all of the items with increasing speed is general, except F_{sj} of the condition B. For F_{Body} , the conventional method gives a little larger value.

Furthermore, as shown in Fig. 12, considerable variation in F_{Body} with the crank angle exists for the conventional method, while no such variation for the present method. Fairly steadiness in F_{Body} of the present method results from inclusion of the centripu-

Fig. 11 (a) Effects of compressor speed on counterweight performance: shaft bearing loads

Fig. 11 (b) Effects of compressor speed on counterweight performance: unbalanced body force

Fig. 11 (c) Effects of compressor speed on counterweight performance: overturning moment

gal force of oldham ring in the radial force balance on the compressor frame.

4. Conclusions

From the application of phase-angled counterweights to a radially compliant scroll compressor, the

Fig. 12 Variation of unbalanced body force with crank angle

following conclusions can be drawn:

1. By circumferential positioning of counterweights in addition to sizing, any two of the following items can be controlled to desired values: shaft main bearing load, sub bearing load, unbalanced body force, and overturning moment of the body.
2. With minimum unbalanced body force, reduction in the shaft main bearing load can be obtained at the expense of increase in the overturning moment.
3. The unbalanced body force can not be totally eliminated, mainly because the centrifugal force of the oldham ring changes with the crank angle. Consideration of its mean value into the radial body force balance results in the removal of variation in the unbalanced body force with the crank angle.

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