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ABSTRACT

This paper describes a new optional method for counterweighting a scroll compressor. The technique involves sizing and positioning counterweights on the drive shaft such that the resultant load imparted by both the compression process and by the inertia of the moving scroll element is nearly cancelled at the drive shaft support bearings. Consequently, expensive bearings could be replaced with lower cost bearings while maintaining or improving reliability and life of the compressor. Some increase in vibration of the compressor frame also can be expected using this method. Thus, a tradeoff must be made between reduced bearing loads and increased frame vibration before an advantage can be gained using this technique in scroll compressor design for certain applications.

NOMENCLATURE

a	Base circle radius of scroll involute
F_{cwl}	Inertia force of lower counterweight
F_{cwu}	Inertia force of upper counterweight
F_{icr}	Inertia force of crank offset (eccentric)
F_{isc}	Radial inertia (centrifugal) force of orbiting scroll
F_{it}	Tangential inertia force of orbiting scroll
F_{pa}	Axial pressure force due to orbiting scroll
F_{pr}	Radial pressure force due to orbiting scroll
F_{pt}	Tangential pressure force due to orbiting scroll
F_{sbc}	Total resultant force acting on crank due to orbiting scroll
h	Height of scroll wrap
i	Index for pair of scroll compression pockets
m_{sc}	Mass of orbiting scroll
N	Number of pairs of compression pockets at start of closed compression
$p_1,\cdots,p_N$	Pressure in scroll compression pockets
p_{sc}	Pressure in suction chamber surrounding orbiting scroll
r_{sc}	Radius from crankshaft axis, f, to orbiting scroll axis, m
z_1, z_2, z_3, z_4	Lengths defined in Fig. 1
ϕ	Relative angle between crank offset and F_{sbc}
θ	Crank angle
ψ_{cwl}	Phase angle of lower counterweight
ψ_{cwu}	Phase angle of upper counterweight
ω	Angular velocity of crankshaft
$\dot{\omega}$	Angular acceleration of crankshaft

INTRODUCTION

Scroll vapor compressors are becoming increasingly popular for air conditioning applications in the 1 to 10 ton capacity range due mostly to their high efficiency, fewer parts, low noise, and low vibration relative to reciprocating compressors.

The principal components in a typical scroll compressor are shown schematically in Fig. 1 and include the fixed and moving (orbiting) scroll elements, drive shaft with eccentric crank, and drive motor. Vapor compression is achieved as the orbiting scroll is driven by the eccentric crankshaft and, as a result, loads are transmitted to the drive shaft and reacted by the support bearings. To dynamically balance a scroll compressor, and thereby reduce the loads at the bearings, two counterweights are typically included on the drive shaft as shown. The balancing of existing scroll compressors follows the practice used with most rotating machinery wherein only inertial forces are considered in sizing and positioning counterweights on the drive shaft; typically, the upper counterweight is positioned close to the crank in the axial direction and 180 deg. from the crank in the circumferential direction while the lower counterweight is aligned circumferentially with the crank. This approach is appropriate for minimizing inertia forces transmitted through the shaft bearings to the frame.

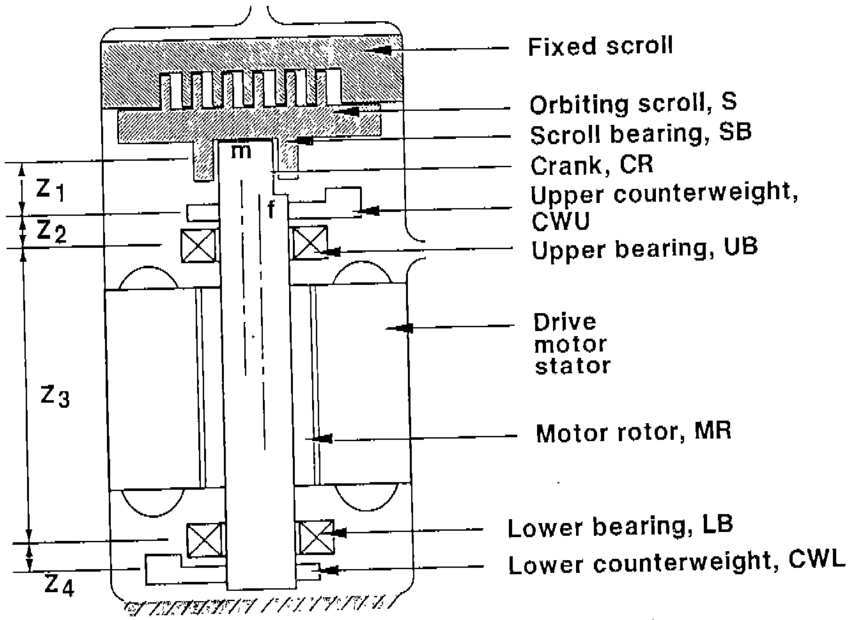


Figure 1 Scroll compressor in low-side shell

An optional method for counterweighting scroll compressors is described herein such that the size and position of the counterweights on the drive shaft are determined by compensating not only the inertia of the orbiting scroll and eccentric crank, but also the pressure forces which act on the orbiting scroll. This technique has the potential for reducing the drive shaft bearing loads, but at the expense of some increase in frame vibration. A discussion of the analysis for this technique is presented along with example cases showing shaft bearing load reductions.

KINEMATIC ANALYSIS OF COUNTERWEIGHTING TECHNIQUE

Previous papers [1-4] have analyzed the loads acting on the orbiting scroll which are developed during steady-state operation. These loads are shown in Fig. 2 and include radial, tangential, and axial pressure loads as well as radial and tangential inertia loads. The primed variables in the lower part of Fig. 2 represent the components in the X-Z plane of the corresponding unprimed variables shown in the upper part of Fig. 2. The pressure forces acting on the fixed and moving scroll elements are generated during the compression process as the moving scroll orbits relative to the fixed scroll. The following equations were used to compute the radial and tangential pressure loads, respectively [1].

$$F_{pr}(\theta) = 2ha(\ p_1 - p_{sc}\ ) \tag{1}$$

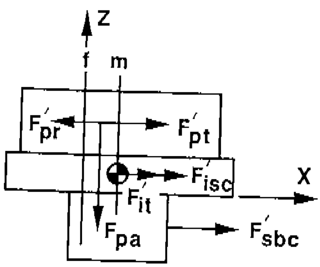
$$F_{pt}(\theta) = 2ha\left\{ \sum_{i=1}^{N-1} (2\pi i - \theta)[p_i - p_{i+1}] + (2\pi N - \theta)[p_N - p_{sc}] \right\} \tag{2}$$

The radial and tangential inertia loads are due to centripetal and angular acceleration, respectively, of the orbiting scroll and are computed as follows:

$$F_{isc} = m_{sc} r_{sc} \omega^2 \tag{3}$$

$$F_{it} = m_{sc} r_{sc} \dot{\omega} \tag{4}$$

Both analysis and measurement show that the speed of a scroll compressor fluctuates little at a given steady-state operating condition. Therefore the angular acceleration is normally very low and F_{it} can be neglected.



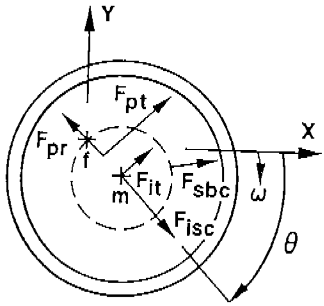


Figure 2 Forces acting on orbiting scroll

Reaction forces to these loads are produced at the orbiting scroll bearing and thrust surface as shown in Fig. 2. A detailed analysis of the forces acting on the orbiting scroll in a typical scroll compressor indicates that the total resultant force vector F_{sbc} which acts on the crank is relatively constant in magnitude with peak-to-peak fluctuations of only about 25% of the average. Even more importantly, the relative angle ϕ between the crank and F_{sbc} is nearly constant. (The vector direction rotates with crank angle at a nearly constant relative angle to the crank). This characteristic of the scroll compressor is shown in Fig. 3 where F_{sbc} and ϕ are plotted as a function of the crank angle. It is apparent from this figure

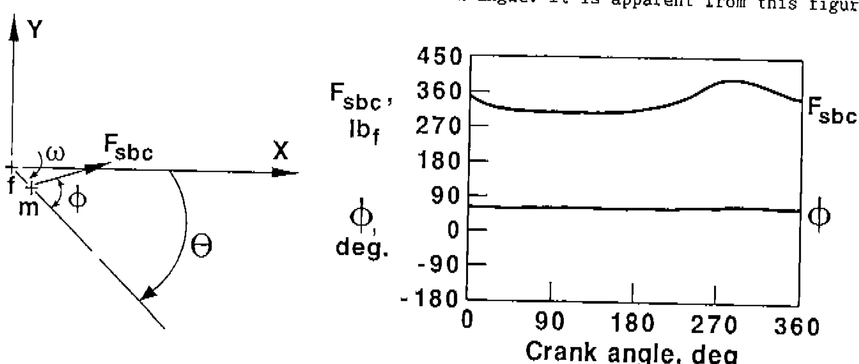


Figure 3 Resultant force acting on crank

that the magnitude of the resultant force does not vary much during one revolution of the crank, and the direction of the resultant force relative to the crank position is nearly constant. Therefore, the total resultant force acting on the crank behaves in a manner similar to the purely inertial forces and, accordingly, the size and location of the counterweights for a scroll compressor can be determined based on this total resultant force.

An analysis of the loads acting on the crankshaft of a scroll compressor will include the resultant force F_{sbc} generated by the orbiting scroll. A schematic of the forces acting on the crankshaft is shown in Fig. 4. The schematic uses center-of-mass symbols to indicate positions of lumped mass for the crank (CR), upper counterweight (CWU), lower counterweight (CWL), and motor rotor (MR). For the purposes of this analysis, upper (UB) and lower (LB) shaft bearings are assumed to be located as shown and indicated by the symbol 'X'. No forces are shown at the shaft bearings in Fig. 4 so that the inertia forces at CWU and CWL can be determined which will produce zero reaction forces at these bearings. Force and moment balances on the crankshaft produce the desired relationship between the loads acting

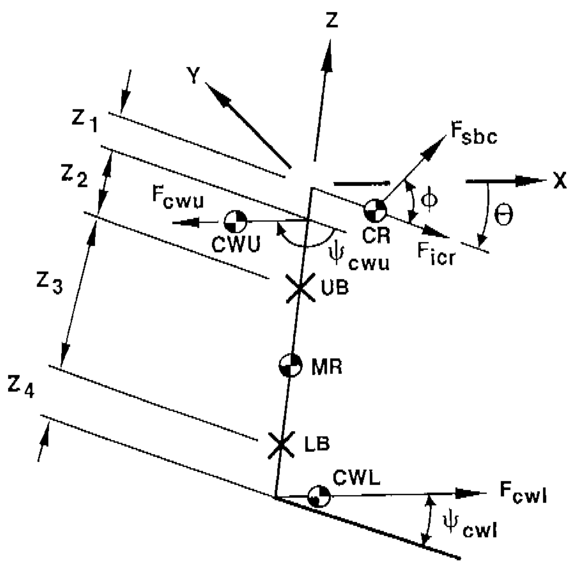


Figure 4 Forces acting on crankshaft

on the crankshaft and the inertia force magnitude and circumferential location for counterweights which will produce zero shaft bearing reaction forces. The inertia force due to motor rotor eccentricity is normally quite small and, therefore, will be neglected. For sizing and positioning counterweights using the conventional method of considering only the inertia forces and ignoring pressure forces, the following equations are commonly used:

$$C = \frac{z_1 + z_2 + z_3 + z_4}{z_2 + z_3 + z_4} \tag{5}$$

$$\psi_{cwu} = \pi \tag{6}$$

$$F_{cwu} = (F_{icr} + F_{isc}) C \tag{7}$$

$$\psi_{cwl} = 0 \tag{8}$$

$$F_{cwl} = (F_{icr} + F_{isc})(C - 1) \tag{9}$$

However, if one chooses the option to size and position counterweights so as to compensate not only inertia, but also pressure forces, then the total resultant force F_{sbc} acting on the crank must be used in the force and moment balances, not just the inertia forces. The following equations determine the inertia force magnitude and location for counterweights which will theoretically elliminate any shaft bearing reaction forces from being transmitted to the frame:

$$\tan(\psi_{cwu}) = \frac{-F_{sbc}\sin\phi}{F_{icr} + F_{sbc}\cos\phi} \tag{10}$$

$$F_{cwu} = F_{sbc} \frac{\sin\phi}{\sin\psi_{cwu}} C \tag{11}$$

$$\psi_{cwl} = \psi_{cwu} + \pi \tag{12}$$

$$F_{cwl} = F_{sbc} \frac{\sin\phi}{\sin\psi_{cwl}} (C - 1) \tag{13}$$

An example of the results obtained using this method are shown in Fig. 5 where the inertia force and location relative to the crank are plotted as a function of crank angle for upper and lower counterweights which produce zero reaction forces at the shaft bearings. If upper and lower counterweights could be constructed having inertia force and center-of-mass location characteristics as shown in Fig. 5, then reaction forces at the shaft bearings would be zero at all crank angles. Such counterweight designs are impractical (if not impossible) to build, however Fig. 5 clearly shows that the magnitudes of the inertia forces vary only moderately while the angular locations relative to the crank are nearly constant. Therefore, the average value of inertia force and angular location could be used to define the upper and lower counterweights so that their basic structural design would be no different than conventional counterweights; only their size and location would be changed as shown schematically in Fig. 6. Likewise, for variable speed applications, passive variable geometry counterweights could be designed that would also effectively maintain zero bearing loads.

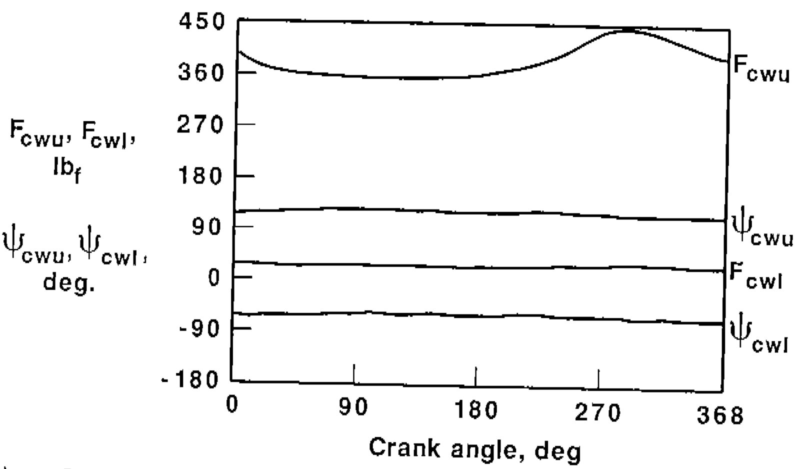


Figure 5 Counterweight inertia forces and locations which produce zero shaft bearing loads

Figure 7 compares reaction forces at the shaft bearings predicted by this method to those predicted by the conventional method for a scroll compressor at 3600 rpm. It is obvious from this comparison that a very significant reduction in shaft bearing forces can be realized using this technique. The predicted reaction forces

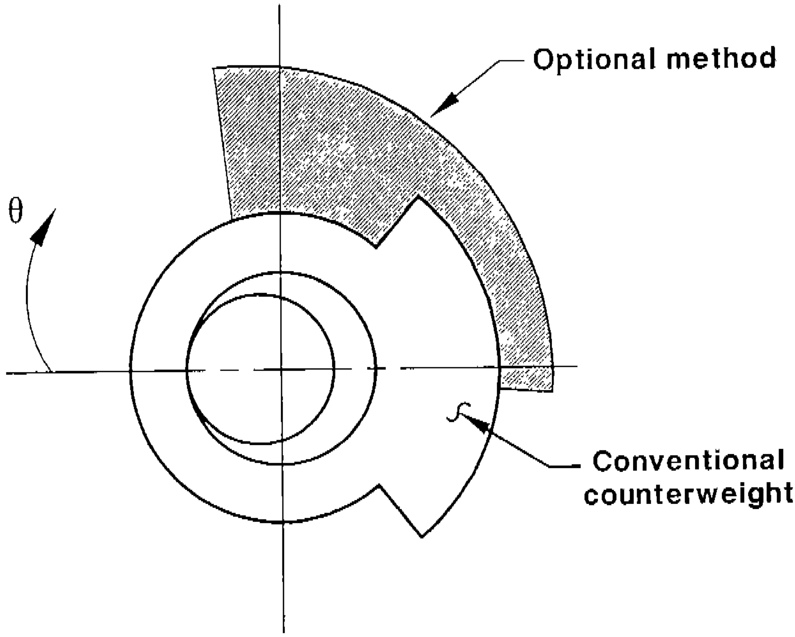


Figure 6 Comparison of conventional and optional upper counterweight

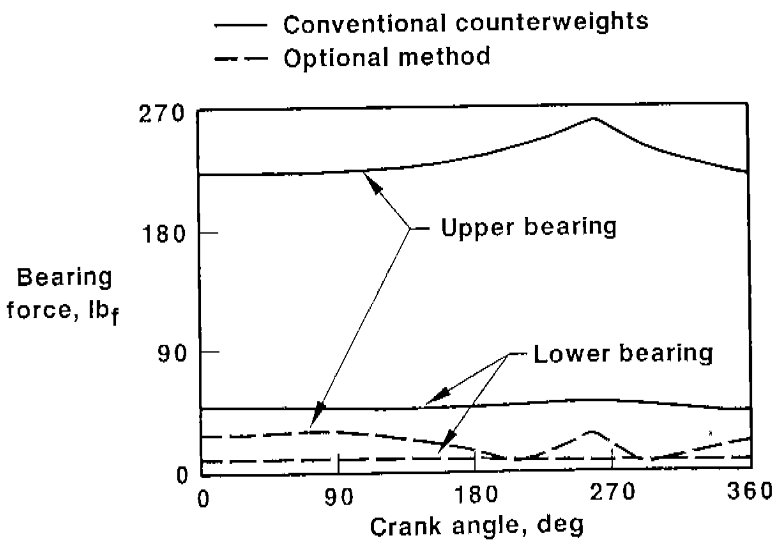


Figure 7 Comparison of shaft bearing reaction forces which result using conventional and optional counterweights

using this method are not zero in Fig. 7 simply because average values for inertia force and angular location were used as proposed previously. The ability to significantly reduce shaft bearing loads translates directly into increased reliability and life of the scroll compressor. Consequently, expensive bearings (such as the roller or ball bearings used in some scroll compressors) could be replaced with lower cost bearings while maintaining or improving compressor reliability and life.

It should be noted that there is a negative aspect of this new technique for counterweighting the crankshaft of a scroll compressor. The difference in the inertia force of counterweights defined by this method and those defined by the conventional method goes directly into increasing the vibration of the frame member which is rigidly attached to both the fixed scroll element and shaft bearings. In many hermetic scroll compressors, the fixed scroll is attached rigidly to the compressor shell, as is the crankcase containing the shaft bearings. For this design, the shell will experience an increase in vibration. However, if the fixed scroll and crankcase are isolated from the shell, or if the shell is isolated from the environment, then the increased vibration which results using this method may be tolerated.

CONCLUSIONS

A new method for counterweighting scroll compressors to minimize shaft bearing loads has been discussed. The technique has the potential for reducing the costs of crankshaft support bearings while increasing life and reliability. However, implementation of these methods can increase the vibration of the compressor. For certain installations, the vibration could be acceptable; in other cases isolation provisions may be required. For instance, in an application where weight is not critical, the weight of the frame or shell structure could be increased to reduce vibration while maintaining the other desirable features of this technique. Thus, trade-offs arise between bearing costs and durability, and acceptable vibration or isolation costs. In any event, the methods presented offer the scroll compressor designer an additional tool that can be used where applicable.

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