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Analysis of journal bearings in a scroll compressor considering deflections and dynamics of the crankshaft

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ABSTRACT

Engineers are trying to design a scroll compressor which covers wide range of speed to realize an efficient and compact air-conditioning system. Recently, a motor with inverter technology has been driven from 15 to 150 Hz in an advanced scroll compressor. The importance of design for a crankshaft and journal bearings has been emphasized in this trend because the lubrication characteristics are greatly affected by the operating speed. In this study, the lubrication characteristics of journal bearings in a scroll compressor are analyzed considering deflections and dynamics of a crankshaft. The Reynolds equation is solved using the finite difference method to determine the pressure distribution of journal bearings, and then deflections of the crankshaft and its four-degrees-of-freedom motions are calculated by the finite element method and the fourth-order Runge–Kutta method, respectively. It is confirmed from the results that the crankshaft deflections significantly affect the minimum clearance of journal bearings.

Analyse des paliers lisses dans un compresseur à spirale en tenant compte des déviations et de la dynamique du vilebrequin

Mots-clés: Compresseur à spirale; Palier lisse; Équation de Reynolds; Déviations du vilebrequin; Dynamique du vilebrequin

1. Introduction

Most of the positive displacement compressors ran at constant operating speed up to approximately twenty years ago. However, compressors have been changed to variable speed type with the advance of motor technology. The variable speed compressors are particularly spot-lighted on the refrigerant compressor because of energy saving regulations. In particular, one of the fast-changing compressors along this trend is a scroll compressor used in a premium air-conditioning system such as a variable refrigerant flow system. The system has various loads conditions because it is used not only for both cooling and heating but also for multiple indoor units. So the scroll compressor used in this system should be able to control the flow rate of the refrigerant through adjusting the motor speed. The scroll compressor for the system is driven in the range from 15 to 150 Hz these days.

The compression part of a scroll compressor is composed of an orbiting scroll and a fixed scroll. Two scrolls are designed by involute curves, and the orbiting scroll moves in a revolutionary motion for compression. The crankshaft of the scroll compressor has three-journal bearings. An orbiting journal bearing (OJB) which is located at the top of the crankshaft transfers motor torque to the orbiting scroll. And, a main journal bearing (MJB) and a sub journal bearing (SJB) are located above and below the motor, respectively. They support two kinds of loads. One is the gas force which acts on the OJB to drive an orbiting scroll. The gas force is transferred to the MJB and SJB with leverage effect. The other is the centrifugal force due to the rotation of a main balance weight (MBW) and a sub balance weight (SBW) which are designed to balance with the motion of the orbiting scroll. The basic configuration of balance weights and journal bearings was proposed by Bush and Elson (1988a,b). They focused on minimizing journal bearing loads from a static point of view. Since then, studies on the crankshaft dynamics and journal bearings in a scroll compressor have been continuously carried out

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Nomenclature

Roman symbols

B	direction of centrifugal force for MBW
c	radial clearance of bearing at concentric status
D	bearing diameter
E	elastic modulus of crankshaft
G	direction of gas force
h	bearing clearance
I	second moment of inertia
I_{xx}, I_{yy}	moment of inertia of crankshaft assembly in θ_x- and θ_y-direction, respectively
L	length of element
m_{CA}	mass of crankshaft assembly
p	oil-film pressure
R	bearing radius
S	distance between lower end of bearing and supporting point
t	time
u	linear displacement of crankshaft
w	deflection of crankshaft

Greek symbols

α, β	angular displacement of crankshaft in θ_x- and θ_y-direction, respectively
γ	deflection angle of crankshaft
μ	viscosity of oil
ϕ	angle between x- and G-directions
ω	angular velocity

Subscripts

1, 5	lower end of sub and MJB, respectively
2, 4	mass center of SBW and MBW, respectively
3	mass center of crankshaft assembly
6	center of OJB
gas	gas force

(Hayano et al., 1988; Meter and DeBlois, 1988; Narumiya et al., 1992; Ahn et al., 2016; Liu et al., 2010). However, the studies have been conducted under the assumption of a rigid crankshaft. Studies on effects of a flexible crankshaft have been done for a rotary vane compressor (Hattori and Kawashima, 1990; Kitsunai et al., 2010; Wang et al., 2013; Zhang et al., 2014). The reason is that an overhang type motor is usually applied to the rotary vane compressor while the motor of a scroll compressor is mounted between two journal bearings. However, the variation of bearing clearance by the crankshaft deflections and the supporting point of a crankshaft has not been strictly considered in the studies.

Dynamics and deflections of a crankshaft are an important issue in compressors as well as in most systems involving a crankshaft and journal bearings. The misalignment, in general, is the concept that encompasses both dynamics and deflections of a crankshaft. The study of misaligned journal bearings in general systems has been carried out until recently. Bouyer and Fillon (2002) analyzed the misaligned effect in a journal bearing experimentally. They confirmed that the minimum clearance is noticeably affected by the misalignment. Guha (2000) analyzed the misaligned hydrodynamic journal bearings with isotropic roughness effect numerically. Furthermore, Sun and Gui (2004) introduced the concept of misalignment caused by crankshaft deflections. In the study, the linear crankshaft deflections were assumed along journal bearings, and the supporting points of a crankshaft were fixed at the center of bearing length. Furthermore, Gui et al. (2017) analyzed the mechanical behaviors of a crankshaft-bearing system in a vehicle. They explained that the consideration of misalignment caused

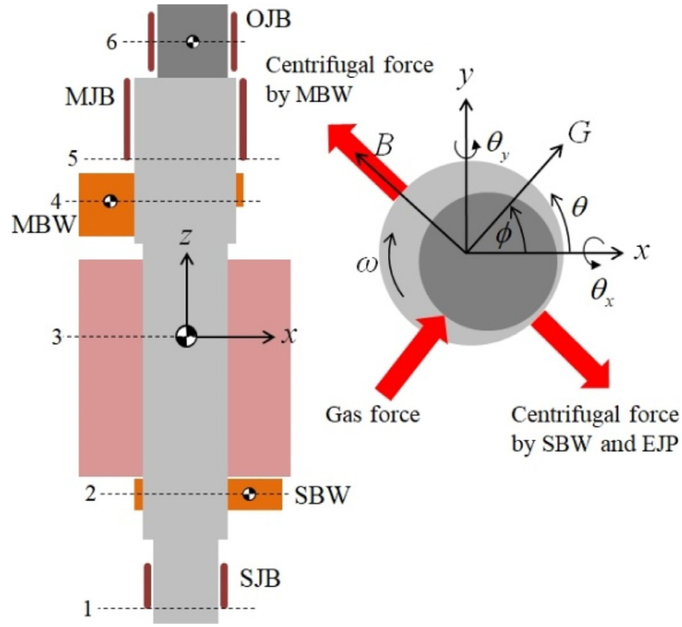

Fig. 1. Analysis model and coordinate systems.

the variation of analysis results for the dynamic behavior and the stress of the crankshaft.

In this study, the lubrication characteristics of journal bearings supporting a crankshaft in a scroll compressor are analyzed considering deflections and dynamics of a crankshaft. In other words, the oil-film pressure of the journal bearings is calculated considering the variation of clearance due to deflections and dynamics of a crankshaft. The Reynolds equation is solved using the finite difference method to determine the pressure distribution of journal bearings. Furthermore, the four-degrees-of-freedom motion of a crankshaft and its deflections are calculated by the fourth-order Runge–Kutta method and the finite element method with one-dimensional beam model, respectively. The supporting points of a crankshaft in the beam model are determined as the acting points of the resultant force of oil-film pressure.

2. Analysis

2.1. Analysis model and coordinate systems

This study is carried out with an analysis model based on a high-side shell scroll compressor used in the variable refrigerant flow system. The analysis model and coordinate systems are represented in Fig. 1. The fixed Cartesian coordinate system (x, y, z) and cylindrical coordinate system (r, θ, z) are used for the dynamic equations and the Reynolds equation, respectively. Furthermore, the rotating Cartesian coordinate system (G, B) is employed to calculate the crankshaft deflections. The G-direction is always parallel to the direction of the gas force, and the B-direction coincides with the direction of the centrifugal forces of an MBW. The rotating Cartesian coordinate system is quite useful because two force vectors are always perpendicular. The angular position of the rotating Cartesian coordinate system, ϕ, can be expressed as follows:

$$\phi = \frac{\pi}{2} + \omega t \tag{1}$$

where, t is the time and ω is the angular velocity of the crankshaft. At $t = 0$, the y-axis coincides with the G-axis. Numbers in Fig. 1 indicate the specific points of the crankshaft. Especially, the point 3 is the mass center of the crankshaft assembly and the point 6 is