Energy Conversion and Management 51 (2010) 277–287

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Transient analysis of a variable speed rotary compressor

Youn Cheol Park *

Department of Mechanical Engineering, Cheju National University, Ara-Dong, Jeju-City, Jeju-Do 690-756, Republic of Korea

ARTICLE INFO

Article history:
Received 8 September 2007
Received in revised form 10 February 2009
Accepted 22 September 2009
Available online 10 November 2009

Keywords:
Variable speed compressor
Rotary compressor
Transient
Dynamic of roller
Dynamic of crankshaft

ABSTRACT

A transient simulation model of a rolling piston type rotary compressor is developed to predict the dynamic characteristics of a variable speed compressor. The model is based on the principles of conservation, real gas equations, kinematics of the crankshaft and roller, mass flow loss due to leakage, and heat transfer. For the computer simulation of the compressor, the experimental data were obtained from motor performance tests at various operating frequencies.

Using the developed model, re-expansion loss, friction loss, mass flow loss and heat transfer loss is estimated as a function of the crankshaft speed in a variable speed compressor. In addition, the compressor efficiency and energy losses are predicted at various compressor–operating frequencies. Since the transient state of the compressor strongly depends on the system, the developed model is combined with a transient system simulation program to get transient variations of the compression process in the system.

Motor efficiency, mechanical efficiency, motor torque and volumetric efficiency are calculated with respect to variation of the driving frequency in a rotary compressor.

1. Introduction

The variable speed compressor is designed to change the motor speed compared to a constant speed compressor. The efficiency of a constant speed motor is dependent on the load of the motor. However, the efficiency of variable speed motor is changed with not only the load of the motor but also the driving frequency. Therefore, the efficiency of the compressor with a variable speed motor in the compression and suction process will be continuously changed with driving frequency and load.

The rolling piston type rotary compressor shown in Fig. 1 is widely used for controlling the speed of the compressor due to the simple mechanism. The compressor consisted with a variable speed motor, crankshaft, roller, cylinder, and vane. Refrigerant enters through a suction side accumulator and is then forced into the suction chamber directly, where there is small amount of suction gas heating. The drawn refrigerant is compressed by the rolling motion of the roller against the cylinder caused by an eccentricity between roller and cylinder center. The vane sustains the roller by spring force and it divides the suction and compression volumes in inside of the cylinder. The compressed refrigerant, which is discharged from the cylinder, resides for a short time in the compressor shell where it undergoes heat transfer and leaves the compressor and enters the condenser through the discharge port of the compressor.

Research of rotary compressors can be divided into categories of constant speed compressors and variable speed compressors. In order to study losses in the constant speed rotary compressor, vibration of the compressor has been studied through analysis of forces and velocity of moving parts using the kinematics of the roller, vane, and crankshaft [1]. The study found that the major source of the vibration is the variation of angular velocity of the crankshaft rather than the velocity variation of the roller. Refrigerant losses through the axial clearance between the outer wall of the roller and the inner wall of the cylinder, as well as the radial clearance between the roller and cylinder surface have been studied in series of four papers [2–5].

For variable speed rotary compressors, comparisons of the performance of rotary and reciprocating compressors were conducted through calorimetric experiments with variation of operating frequency of the compressors [6]. The results showed that the volumetric efficiency of the rotary compressor increases with the operating frequency. In contrary, for the reciprocation compressor, mechanical, compression and volumetric efficiency decrease with operating frequency. Another result of the study shows that the capacity of the rotary compressor is proportional to the operating frequency. For the reciprocating compressor, however, capacity of the compressor linearly varies with operating frequency only in the low frequency range and there is no fluctuation of the capacity in the high frequency range. From the above results, it is possible to note that the rotary compressor might be more suitable as a capacity regulation compressor.

* Tel.: +82 64 754 3626; fax: +82 64 756 3886.
E-mail address: ycpark@cheju.ac.kr

0196-8904/$ - see front matter © 2009 Elsevier Ltd. All rights reserved.
doi:10.1016/j.enconman.2009.09.023

Nomenclature

a	length of vane (m)
A	area (m^2)
A_c	cylinder inside area (m^2)
A_h	cylinder head area (m^2)
A_r	roller outside area (m^2)
A_{sp}	suction port area (m^2)
A_v	vane side wall area (m^2)
b	ratio of roller radius to cylinder radius (=R_r/R_c)
d	diameter of suction port (m)
e	eccentricity (m)
F	vector summation of forces acting on roller (N)
F_d	gas force acting on root of vane (N)
F_h	gas force acting on vane (N)
F_i	inertia force of vane (N)
F_k	spring force acting on vane (N)
F_p	pressure force (N)
F_{vn}	normal component of friction force at vane tip (N)
g	gravity (m/s^2)
h	cylinder height (m) or heat transfer coefficient (W/m^2 K)
h_j	height of journal bearing (m)
i	enthalpy (kJ/kg)
I_s	inertia moment of the crankshaft (N m s^2)
k	adiabatic coefficient (−)
l	length of suction port (m)
l_r	length between center of cylinder and exerting point of total force (m)
l_{vs}	length of vane inside vane slot (m)
m	mass of fluid (kg)
$\dot{m}$	mass flow rate (kg/s)
$\dot{m}_{loss}$	rate of mass flow loss (kg/s)
$\dot{m}_{cano}$	discharge mass flow rate from compressor (kg/s)
M_f	moment action on crankshaft by roller (N m)
M_j	moment of journal bearing (N m)
M_k	moment of friction force between crank fin and inside of roller (N m)
M_r	moment of friction force between roller and cylinder head (N m)
M_t	moment of thrust bearing (N m)
N	rotational speed of motor (rpm)
P	pressure (Pa)
P_d	discharge pressure (Pa)
P_n	discharge pressure (Pa)
P_s	suction pressure (Pa)
P_u	upstream pressure (Pa)
Q_{can}	heat transfer from compressor case (kJ)
Q_{cyc}	heat transfer from cylinder (kJ)
Q_{mot}	heat transfer from motor (kJ)
R_1, R_2	reacting forces on vane (N)
R_c	radius of cylinder (m)
R_e	radius of crank fin (m)
R_j	radius of journal bearing (m)
R_r	radius of roller (m)
R_s	radius of shaft (m)
R_v	radius of vane tip (m)
t	time (sec) or vane thickness (m)
T	temperature (K)
T_{cyl}	cylinder temperature(K)
v	specific volume (m^3/kg)
V	volume (m^3)
W	relative humidity
W_r	relative humidity at room
$\dot{W}$	compressor power (kW)
x	vane displacement (m)

Greek symbols

α	offset angle of eccentric center (rad)
β	angle of suction port (rad)
δ	clearance (m)
ϕ	angle from vane axis (rad)
η_{inv}	inverter efficiency
μ_t	coefficient of friction at thrust bearing
μ_j	coefficient of friction at journal bearing
μ_{vn}	coefficient of friction at vane tip
μ_{vs}	coefficient of friction at vane side slot
θ	rotational angle of crankshaft (rad)
θ_f	angle of total force (rad)
ρ	density(kg/m^3)
T_L	load torque (N m)
T_M	motor torque (N m)
ω	angular velocity (rad/s)
ω_e	relative angular velocity of between crank pin and roller (rad/s)
ω_p	angular velocity of roller (rad/s)
ω_s	angular velocity of crankshaft (rad/s)

A loss analysis on rotary compressors conducted experimentally with a pressure–volume diagram [7]. The results show that mechanical losses at low frequency are the most important among all losses in the compressor even though the absolute value of the loss is smaller than those seen at high frequency operation. Generally air conditioners run, for the most part, at low frequency conditions; in order to increase energy saving of air conditioning systems, the mechanical losses have to be reduced not only at high frequency operation but also at low frequency operation. Loss analyses of variable speed rotary compressors also can be found in some literature, including a paper about high-speed compressors that run at up to 180 Hz frequency [8].

The effect of suction gas heating, mass flow losses, re-expansion of compressed refrigerant, and over-charging of the refrigerant on the compressor performance has been studied [9]. This paper suggests that over-charging of the refrigerant increases compressor capacity with a small degradation of the efficiency. Also, how the compressor performance is affected by refrigerant leakage is the most important parameter at low frequency operation; while on the other hand, the re-expansion of the compressed refrigerant is important at high frequency operation.

In the past, most compressor dynamic analysis was conducted with constant motor efficiency or without consideration of motor performance, even studies on variable speed compressors. The variable speed motor is the heart of the compressor, which drives the crankshaft and changes the speed of the compressor. In this study, the motor performance with variation of the driving frequency was experimentally determined. This study is concerned with developing a dynamic program to predict friction loss, mass flow loss, re-expansion loss, and heat transfer loss as a function of the variation of driving frequency in a variable speed compressor. Several computational models for the compression process are based on the polytropic process. In this study, the model is developed with conservation equations, a real gas equation and kinematics of the crankshaft and roller.

The present study can be characterized as follows:

(1) The model in this study is a dynamic analysis model based on the principles of the mass and energy conservation equations. The variations of temperature and pressure of the refrigerant can be calculated with the rotation of the crankshaft.