INTERNATIONAL JOURNAL OF REFRIGERATION 36 (2013) 1701–1715

Available online at www.sciencedirect.com

SciVerse ScienceDirect

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

Experimental study on stepless capacity regulation for reciprocating compressor based on novel rotary control valve

Dacheng Li [a,*], *Haiqi Wu* [b], *Jinji Gao* [b]

[a] *State Key Laboratory of Multi-Phase Complex Systems, Institute of Process Engineering, Chinese Academy of Sciences, P.O. Box 20, No. 1 Zhongguancun North Second Street, Beijing 100190, PR China*

[b] *Engineering Research Center of Chemical Technology Safety of the Ministry of Education, Beijing University of Chemical Technology, Beijing 100029, PR China*

ARTICLE INFO

Article history:
Received 27 March 2012
Received in revised form 25 January 2013
Accepted 4 April 2013
Available online 18 April 2013

Keywords:
Reciprocating compressor
Capacity reduction
Rotation
Valve
Rotor
Energy saving

ABSTRACT

A capacity-regulation system based on a novel rotary control valve for reciprocating refrigeration compressor is proposed and designed for the first time. The regulation system is mainly composed of a rotary control valve and an adaptive regulation system. The structure and working principle of the rotary control valve is described in detail, and the control process of the adaptive regulation system for the valve is studied together with the program design. In addition, the parameters for the design and control of the rotary control valve are theoretically determined. To verify the feasibility and effectiveness of the proposed system, a three-cylinder reciprocating compressor was adopted as a test device. Experimental results showed that the technology was able to realize continuous stepless capacity regulation for the compressor within the range of (0)10–100%, and power consumption decreased correspondingly with the load reduction.

Etude expérimentale sur la régulation de la puissance en une seule étape d’un compresseur à piston muni d’une vanne rotative innovante

Mots clés : compresseur à piston ; réduction de la puissance ; rotation ; vanne ; rotor ; économies d’énergie

1. Introduction

Reciprocating refrigeration compressors characterized by high energy consumption are widely used in the industries. Under normal working conditions, the displacement of the compressor does not change because of its volume structure. However, in practical applications, consumption of the refrigerating output often changes with the requirement of

* *Corresponding author*. Tel.: +86 010 82544917, +86 13693080880 (mobile).
E-mail addresses: drdanlee@live.cn, lidacheng1983@hotmail.com (D. Li).

http://dx.doi.org/10.1016/j.ijrefrig.2013.04.002

Nomenclature

Roman

a_R	flow efficiency of the scallop hole
a_V	flow efficiency of the original valve clearance
A_{out}	flow area of the valve seat outlet (m^2)
A_R	geometric area of the scallop hole (m^2)
A_V	geometric area of the original valve clearance (m^2)
H	maximum valve lift (m)
H_0	precompression length of spring (m)
I	current for the gas circulation in the modified cylinder under work pressure (A)
k	adiabatic exponent
l	center-to-center spacing of the two ends of connecting rod (m)
m	expansion exponent
M	Mach number of the gas flow in the valve clearance
N	number of the preset pulses
N_{close}	number of pulses outputted by PLC in the closed section
N_{open}	number of pulses outputted by PLC in the open section
N_{p1}	pulse number of acceleration in curve AB
N_{p2}	pulse number of constant speed in curve AB
N_{p3}	pulse number of deceleration in curve AB
N_{p4}	pulse number of acceleration in curve CE
N_{p5}	pulse number of constant speed in curve CE
N_{p6}	pulse number of deceleration in curve CE
N_R	number of the scallop hole
p_a	gas pressure in the cylinder after the suction (Pa)
p_b	gas pressure in the cylinder after the backflow (Pa)
p_d	discharge pressure (Pa)
p_s	suction pressure (Pa)
P	power consumption (kW)
Q	displacement of the compressor ($m^3\ min^{-1}$)
Q_{max}	maximum displacement to be compressed ($m^3\ min^{-1}$)
r	crank radius (m)
r_R	internal radius of the scallop hole (m)
R	gas constant ($m^2\ s^{-2}\ K^{-1}$)
R_R	external radius of the scallop hole (m)
S_P	sectional area of the piston (m^2)
S_R	effective flow area of the scallop hole (m^2)
t	rotation time of the crankshaft (s)
t_{count}	time needing to count the preset signal pulse (s)
t_d	time length required for the tank pressure to rise from p_1 to p_2 under different loads (s)
t_{min}	time length t_d required under the maximum displacement (s)
t_{open}	open time of rotary control valve (s)
t_{p1}	initial cycle time of acceleration in curve AB (s)
t_{p2}	initial cycle time of constant speed in curve AB (s)
t_{p4}	initial cycle time of acceleration in curve CE (s)
t_{p5}	initial cycle time of constant speed in curve CE (s)
t_{R1}	time length for rotor rotates to the maximum openness degree from the closed section (s)
$\overline{t_{R1}}$	average cycle time of the pulses during t_{R1} (s)
t_{R2}	time length for rotor rotates from the maximum openness degree to the closed section (s)
$\overline{t_{R2}}$	average cycle time of the pulses during t_{R2} (s)
t_{R3}	time length for rotary control valve remains in the maximum openness degree (s)
t_{R4}	time length for rotor rotates in the closed section (s)
$\overline{t_{R4}}$	average cycle time of the pulses during t_{R4} (s)
t_1	time length for original valve sheet moves to the lift stop from the valve seat at the beginning of suction stroke (s)
t_2	time length for original valve sheet returns to the valve seat from the lift stop at the end of suction stroke (s)
Δt	preset time for the timer interrupt (s)
T_b	gas temperature in the cylinder after the backflow (K)
T_s	suction temperature (K)
(p_1, T_1)	pressure and temperature of the tank in first record (MPa, K)
(p_2, T_2)	pressure and temperature of the tank in second record (MPa, K)
u_b	backflow velocity of the gas through the scallop hole ($m\ s^{-1}$)
u_P	instantaneous velocity of the piston ($m\ s^{-1}$)
U_{prs}	standard uncertainty for pressure (s)
U_{re1}	standard uncertainty for t_{min} (s)
U_{re2}	standard uncertainty for t_d (s)
U_{timing}	standard uncertainty for timing (s)
U_{T_1}	standard uncertainty for T_1 fixing (s)
U_{T_2}	standard uncertainty for T_2 fixing (s)
U_η	measurement uncertainty of the cooling load (%)
U_1	uncertainty component caused by the measurement repeatability of time (%)
U_2	uncertainty component caused by the timing error (%)
U_3	uncertainty component caused by the pressure measurement (%)
U_4	uncertainty component caused by T_1 fixing under different loads (%)
U_5	uncertainty component caused by T_2 fixing under different loads (%)
V	actual volume to be compressed under the regulation system (m^3)
V_b	suction gas volume after the backflow (m^3)
V_{bs}	stroke volume after the backflow (m^3)
V_c	volume of the cylinder (m^3)
V_{max}	maximum volume could be compressed under the regulation system (m^3)
V_0	clearance volume of the cylinder (m^3)
ΔV_b	volume reduction under p_b by expansion of the high-pressure gas (m^3)
Z	unit spring force ($N\ m^{-1}$)

Greek

α	relative clearance volume
β_t	thrust exponent