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Dynamic model of a hermetic reciprocating compressor in on–off cycling operation (Abbreviation: Compressor dynamic model)

Demba Ndiaye, Michel Bernier *

Département de Génie Mécanique, École Polytechnique de Montréal, C.P. 6079, Succursale Centre-Ville, Montréal, Québec, Canada H3C 3A7

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ABSTRACT

This paper presents a dynamic model of a hermetic reciprocating compressor in on–off cycling operation. The proposed model is detailed enough to account for the important phenomena influencing the suction and discharge mass flow rates and the electrical power drawn by the compressor, but simple enough to be usable for different reciprocating compressor designs with readily available data from manufacturers. The experimental validation of the model under steady-state and transient conditions in both heating and cooling modes confirms the good performance of the model.

1. Introduction

The compressor is often the most complex component in refrigeration systems. The compressor models found in the literature may be grouped into either steady-state or dynamic models, each of various degrees of complexity. The models of Cavallini et al. [1] and Navarro et al. [2] are two examples of a simplified steady-state model. More complex steady-state models will require very specific proprietary data only available from manufacturers. Furthermore, due to the complex mass, pressure and thermal fluctuations inside the compressor, models will be specific to a particular compressor design. Steady-state compressor models cannot capture the dynamic behavior of refrigeration systems. For example, the suction and discharge mass flow rates may be different during transient operations. Equally important is to have a model that can account for the off-operation of the compressor to fully capture the on–off cycling of the refrigeration system. In this case, the evolution of the refrigerant in the shell is important with regards to the interaction of the compressor with the evaporator, the evolution of temperatures in the compressor, and compressor heat losses during shutdown.

In simplified dynamic modeling of compressors, the shell is usually treated as a single lumped element and the discharge mass flow rate is derived from correlations [3,4] or from a thermodynamic analysis of the compression-expansion process [5–7]. None of these models addresses the off-operation of the compressor. Further, the model of Rossi and Braun [4] assumes equal suction and discharge mass flow rates at all times. This assumption is only valid in steady-state operation.

This paper presents a dynamic model of a hermetic reciprocating compressor in on–off cycling operation. The proposed model is detailed enough to account for the important phenomena influencing the suction and discharge mass flow rates and the electrical power used by the compressor, but simple enough to be usable for different reciprocating compressor designs with readily available manufacturer data.

In the subsequent sections of this paper, the behavior of the refrigerant in the shell and the resulting temperature evolution of the different components such as the motor are investigated. The discharge flow rate and the work of compression are determined via a thermodynamic analysis of the compression and expansion processes, and the electrical power used by the compressor is evaluated on the basis of manufacturer's performance data. Finally, a discussion of the experimental validation of the model in both steady-state and transient conditions follows the presentation of the numerical solution technique.

2. Suction and discharge

Fig. 1 presents a schematic view of a hermetic reciprocating compressor. The refrigerant enters into the compressor through the suction tube. The refrigerant cools the compressor by absorbing heat coming from the motor, the mechanical parts, the oil, and other metallic parts including the discharge tubing and the wall of the shell. A pressure drop occurs when the

* Corresponding author. Tel.: +1 514 340 4711x4381; fax: +1 514 340 5917.
E-mail address: michel.bernier@polymtl.ca (M. Bernier).

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Nomenclature

a	regression coefficient
A	area (m^2)
C	clearance ratio
C	friction factor
Cp	specific heat ($J\ kg^{-1}\ K^{-1}$)
Cst	constant
D	diameter (m)
h	enthalpy ($J\ kg^{-1}$)
h	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
L	length (m)
m	mass (kg)
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
n	polytropic exponent
N	frequency (Hz)
P	pressure (Pa)
$\dot{Q}$	heat transfer rate (W)
T	temperature (°C)
t	time (s)
V	volume (m^3)
w	specific work ($J\ kg^{-1}$)
$\dot{W}$	power (W)
W	work (J)

Greek letters

Δ	differential
μ	viscosity ($kg\ m^{-1}\ s^{-1}$)
η	efficiency
ρ	density ($kg\ m^{-3}$)

Subscripts

1	suction side of the compressor
2	upstream piston suction valve
3	downstream piston suction valve
4	upstream piston discharge valve
5	downstream piston discharge valve
6	discharge side of the compressor
a	state of the refrigerant in the cylinder after suction and before compression
b	state of the refrigerant in the cylinder after compression and before discharge
c	compressor
c	refrigerant in the shell of the compressor
c	state of the refrigerant in the cylinder dead volume before expansion
C	convective
ch	crankcase heater
d	state of the refrigerant in the cylinder dead volume after expansion
dis	discharge
dt	discharge tubing
dv	discharge valve
env	environment
f	free volume in the shell
f	friction
Hz	hertz
m	compressor motor
o	lubricating oil
r	refrigerant
rc	refrigerant to metallic parts of the compressor
R	radiative
suc	suction
sv	suction valve
sw	swept
th	theoretical
v	volumetric
w	wall

Superscript

0	initial

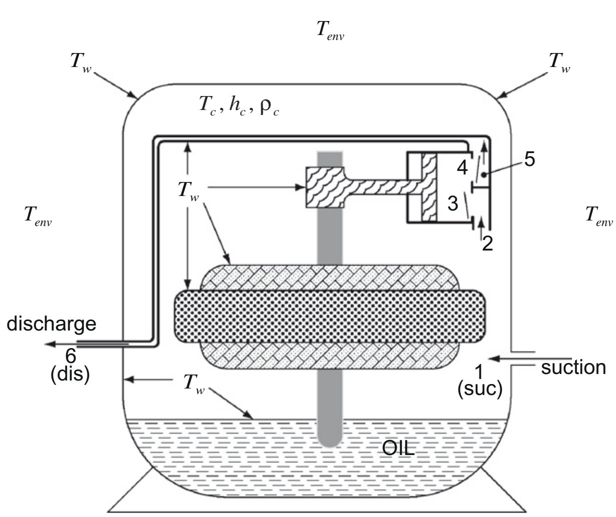

Fig. 1. Schematic representation of a hermetic reciprocating compressor.

refrigerant passes through the suction valve. A pressure drop may also occur in the suction tubing. After compression in the cylinders, the refrigerant is discharged through the discharge valve and tubing, where it experiences other pressure drops. The refrigerant in the discharge tubing will lose part of its heat to the compressor.

The lubricating oil can mix with the refrigerant and be entrained into the system. Given that there is no viable procedure to model the oil-refrigerant interaction and considering that refrigeration systems are conceived to minimize oil entrainment in the system, the effects of the oil will be neglected in the modeling of the compressor.

2.1. Suction side

The analysis on the suction side is based on a few assumptions. First, it is assumed that any refrigerant entering into the compressor will first mix with the refrigerant already present in the shell before entering in the cylinders. It is also assumed that this mixture is thermally homogeneous. Finally, the effects of suction and discharge mufflers on the heat exchange and pressure drop are not accounted for.

With these assumptions, the following mass and energy conservations equations apply for the refrigerant in the shell:

$$\frac{d\rho_c}{dt}V_f = \dot{m}_{suc} - \dot{m}_2 \tag{1}$$

$$\frac{d(\rho_c h_c)}{dt}V_f = \dot{m}_{suc}h_{suc} - \dot{m}_2 h_2 + \dot{Q}_c + \dot{Q}_{ch} + \frac{dP_{suc}}{dt}V_f \tag{2}$$