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Development of a control algorithm employing data generated by a white box mathematical model

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HIGHLIGHTS

► The model was validated in steady-state and dynamic conditions using experimental data.
► A variable maximum error was utilized to address the problem caused by the change in the heat transfer coefficient.
► The heat transfer coefficient of the liquid deficient region was estimated using a third-order polynomial.
► A full multivariable adaptive controller was developed using the data generated by the proposed model.

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ABSTRACT

The optimization of vapor compression refrigeration systems to improve their energetic efficiency has become an important goal of the field of thermal engineering. To this end, a refrigeration capacity control that operates through the continuous adjustment of the compressor speed and the opening of the expansion valve has been utilized. To develop the algorithm responsible for these adjustments, it is necessary to gather information about the refrigeration system's dynamics. This information is generally obtained from experimental data, which is not always available. This study presents the development of a concentric tube evaporator mathematical model. After being validated with experimental data, this mathematical model was utilized to generate the information on the system dynamics that is necessary to project an adaptive multivariable controller. The obtained results showed that the proposed model can be used to describe refrigeration machine dynamics and that this information can be used in the controller design.

1. Introduction

Refrigeration systems are great energy consumers, and their operations are still considered inefficient. The capacity control method employed in most of these systems consists of a thermostat used to turn the compressor on and off to maintain the temperature between the desired levels. However, it is well known that this method of capacity control is responsible for efficiency losses as a result of the start-up and shut-down transients [1–4]. In this process, during the compressor's off cycle, refrigerant can migrate from the condenser and the liquid line to the evaporator until the system pressure equalizes. This phenomenon increases the evaporator temperature and makes a refrigerant redistribution during the on cycle necessary, which reduces the system's efficiency [5]. In addition to the efficiency losses related to the migration of refrigerant fluid, during the compressor start-up, the electrical motor current can reach very high levels of approximately 8–10 times higher than the nominal current. This strong current peak during the compressor start-up also contributes to the efficiency loss [6].

The introduction of variable speed compressor and electronic expansion valves allowed more efficient control methods of capacity regulation to be employed. By adjusting the compressor speed and the expansion valve opening continuously in relation to the system's thermal load, it is possible to optimize the set point tracking and promote an improvement in these systems' energy efficiency [4,5,7–10]. The magnitudes of these adjustments are defined by a control algorithm.

To design such a control algorithm, knowledge about the refrigeration system dynamic model is necessary. The most

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Nomenclature

Latin symbols

$\dot{Q}$	refrigeration capacity
$\dot{V}$	volumetric flow rate
A	section (m^2)
C	controller transfer function matrix
c	specific heat (J kg^{-1} K^{-1})
c_r	compressor clearance ratio
G	mass flux (kg m^{-2} s^{-1})
G	transfer function
h	enthalpy (J kg^{-1})
K	gain (K h kg^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
N	compressor speed (r.p.s.)
n	number of tubes
p	perimeter (m)
P	pressure (Pa)
s	Laplace operator
T	temperature (°C or K)
t	time (s)
t	time constant (s)
V	Piston displacement volume (m^3)
v	specific volume (m^3 kg^{-1})
W	pre-compensator transfer function matrix
x	vapor quality
z	axis (m)

Greek symbols

α	heat transfer coefficient (W m^{-2} K^{-1})
λ	void fraction
Δ	property variation
η	efficiency
θ	heat exchanger pipes inclination angle (rad)
ρ	density (kg m^{-3})

Subscripts

1	evaporator inlet
2	compressor inlet
3	compressor outlet
a	secondary fluid
d	diagonal
def	liquid deficient
critical	critical property
eb	evaporation
F	friction
f	refrigerant fluid
i	integral
l	liquid
p	constant pressure
w	pipe wall
p	proportional
v	constant volume
v	vapor
v	superheating region

Superscripts

0	preceding moment of time

common approach to obtaining this mathematical model, especially for the purposes of controller design, consists in performing experimental measurements of the system's response to some defined inputs and trying to determine a mathematical relation between them without delving into the details of what is actually happening inside the system (black box model) [11]. Gathering the experimental data necessary to this process can be difficult because the physical system may not be available for experimental tests, or it may not have the required instrumentation to provide measurements of all the variables of interest.

A new strategy consists of using a mathematical model based on governing physical laws (the white box model) to generate these data through computer simulations. This approach may be advantageous, especially when it is not possible to perform experimental tests or when the tests' costs are relatively high.

This work presents a mathematical model of a concentric tube evaporator. The information about the system's dynamics obtained from the computer simulations using this mathematical model was employed to design an adaptive multivariable controller. The controller's effectiveness was evaluated through simulations, and all the results are presented in this study.

2. Experimental apparatus

The experimental device used is presented in Fig. 1. This device consists of a vapor compression refrigerating system that uses R134a as the refrigeration fluid and, as a secondary fluid, pure water in the evaporator and condenser. The system comprises a reciprocating compressor, a condenser, a sub-cooler, an evaporator, three expansion valves and systems to perform measurements and data acquisition. The compressor is of the alternative type and has a piston displacement of 157 cm^3. A three-phase electrical motor is employed to drive the compressor. This electrical motor is powered by a frequency inverter that enables variations in the revolution speed of the motor-compressor assemblage in a wide range of operations. The operating points employed in this study correspond to 48 Hz (650 rpm) and 56 Hz (750 rpm). A shell and tube type condenser with a 6 kW capacity is used. The secondary fluid temperature in the condenser is adjusted by mixing warm water that comes from the condenser itself with room-temperature water coming from the feeding system. A coaxial type sub-cooler made of an envelope tube and an internal tube in a "U" shape is used. A multiple-tube type coaxial evaporator is also used, composed of an envelope tube of PVC and of three inner copper tubes through which the refrigeration fluid flows. The water flows in a counterflow in the annular space between the PVC and copper tubes. The evaporator was projected to provide a maximum refrigeration capacity of 3 kW. In the evaporator, the secondary fluid temperature is maintained within the desired limits by an electrical heating system. The experimental bench has three expansion valves placed in parallel (manual, thermostatic and electronic types). A blockage valve permits the isolated operation of each expansion device. In this study, only a manual expansion valve was used. Eleven T-type thermocouples were installed inside the tubes at the inlet and outlet of each system component. Two piezoresistive pressure sensors were installed at the inlets and outlets of the expansion devices. At its entrance, the sensor has a range of measurement from 0 to 2.4 MPa, while at its exit, its range is from 0.1 to 0.9 MPa. The refrigerant mass flow was measured with a Coriolis flow meter whose range of measurement is from 15 to 200 kg h^{-1}. The accuracy provided by the manufacturer for both pressure sensors is 0.2% and 0.7% for the mass flow meter. To determine the mass flow of the secondary fluid in the evaporator, a test tube (with subdivisions of 20 ml) and a digital chronometer were used. All the signals