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Research Paper

On identifying steady-state parameters of an experimental mechanical-compression refrigeration plant

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HIGHLIGHTS

- Thermodynamic modelling of a vapour mechanical compression refrigeration system.
- Steady-state identification method focused on the heat exchangers.
- Overall heat transfer coefficients identified using experimental data.
- Order of magnitude of heat transfer coefficients from the literature are considered.
- Joint validation using an experimental plant, achieving low estimation errors.

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ABSTRACT

This paper describes the thermodynamic model of a vapour compression refrigeration system, focusing on the performance of the heat exchangers. A novel steady-state identification method is proposed, which is based on inaccessible refrigerant phase-change zones assuming that a constant overall heat transfer coefficient can be identified for each zone. The proposed method has been applied to a configurable two-stage, two-load-demand vapour compression refrigeration plant. Considering steady-state experimental data and taking into account orders of magnitude of the global heat transfer coefficients found in the literature, some relationships between them and the heat exchanger zone lengths are suggested. Then coherent values are obtained for all heat transfer coefficients, which confirms the validity of the hypothesis. Additionally the remaining elements of the refrigeration system have been identified using well-known models. Experiments have been carried out in order to validate this method, using a one-stage, two-load-demand configuration. Although all components are separately identified, a global validation method is implemented, of which results show good agreement with experimental data, with relative estimation errors around 10% in the key variables.

1. Introduction

The most extensive method for cooling generation is the vapour compression refrigeration cycle, regardless of its application, as it can be industrial, domestic, and commercial refrigeration, as well as air conditioning [1]. A high percentage of energy consumption is due to the widespread use of refrigeration processes, as supported by several studies in the literature, which detail their environmental and economic impact. Baxter states that supermarkets are one of the highest consumers in the area of energy [2]. An average-sized department store consumes around 3 million kWh yearly, and approximately 50% of this energy is used for refrigeration purposes. In the case of office buildings, Lombard et al. stated that the consumption due to HVAC (Heating, Ventilating, and Air Conditioning) systems is around 20–40% of total energy consumption of developed countries [3].

The increasing concern about shortage of natural energy sources combined with a more efficient energy management offer new opportunities for the development of new design and control strategies expected to improve energy efficiency and reduce environmental impact of current refrigeration systems. These improvements involve:

- Redesign of some elements, mainly the heat exchangers, pursuing a significant improvement in performance.
- Integration of electronics and control systems, in order to operate with floating pressures.

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Nomenclature

Latin symbols

A	heat exchanger cross section at the refrigerant side (m^2)
A_{trnsf}	heat transfer area per heat exchanger length unit (m)
A_v	expansion valve opening (%)
b	compressor parameter
C	thermal capacity ratio
COP	Coefficient of Performance
c_{eev}	electronic expansion valve coefficient (m^2)
c_p	specific heat at constant pressure ($J\,kg^{-1}\,K^{-1}$)
c_{puv}	pressure upholding valve coefficient (m^2)
c_v	specific heat at constant volume ($J\,kg^{-1}\,K^{-1}$)
c	compressor parameter
EEV	electronic expansion valve
g_{he}	heat exchanger characteristic function
h	specific enthalpy ($J\,kg^{-1}$)
k_{drop}	pressure drop factor ($kg^{-1}\,m^{-1}$)
L	heat exchanger length (m)
$\dot{m}$	mass flow ($kg\,s^{-1}$)
N	compressor speed (Hz)
NTU	Number of Transfer Units
OP	operating point
P	pressure (Pa)
$\dot{Q}$	cooling power (W)
R	thermal resistance ($m^2\,K\,W^{-1}$)
RE	relative error (%)
S_t	compressor parameter (m^3)
s	specific entropy ($J\,kg^{-1}\,K^{-1}$)
T	temperature (K)
T_{SC}	subcooling degree (K)
T_{SH}	superheating degree (K)
t	time (s)
UA	global heat transfer coefficient ($W\,K^{-1}$)
υ	specific volume ($m^3\,kg^{-1}$)
$\dot{W}$	compressor power (W)

Greek symbols

α	overall heat transfer coefficient between the refrigerant and the secondary flux ($W\,m^2\,K^{-1}$)
$\boldsymbol{\Gamma}$	vector of experimental outputs in the identification process
$\bar{\gamma}$	mean void fraction
ε	heat exchanger efficiency
ζ	heat exchanger zone length
$\boldsymbol{\theta}$	parameter vector in the identification process
λ	convection heat transfer coefficient ($W\,m^2\,K^{-1}$)
$\boldsymbol{\xi}$	residue vector in the identification process
ρ	density ($kg\,m^{-3}$)
σ	standard deviation
$\boldsymbol{\varphi}$	vector of experimental inputs in the identification process

Subscripts

c	condenser
$comp$	compressor
$cond$	conduction
dis	discharge
e	evaporator
$e5$	evaporator at 5 °C
$e20$	evaporator at −20 °C
f	saturated liquid
g	saturated vapour
in	inlet
is	isentropic
max	maximum
mid	middle
min	minimum
out	outlet
sat	saturation
sc	subcooled liquid zone
sec	secondary flux
sh	superheated vapour zone
suc	suction
$surr$	surroundings
tp	two-phase zone

- Development of control systems capable of automatic detection of anomalies and loss of service.
- Instigation of techniques which allow system parameters to continuously adapt to operating conditions, seeking high efficiency, in terms of Coefficient of Performance (*COP*), both in steady state and in transient states.
- Use of environment-friendly refrigerants harmless in their impact on the ozone layer.

These improvements require a deep knowledge of the process. Use of multivariable control techniques plays a central role. Furthermore, testing such control techniques at a purpose-built breadboard refrigeration plant would greatly help to achieve real improvements in terms of energy efficiency and controller performance. Some examples of refrigeration experimental plants can be found in the literature but these are usually one-stage refrigeration cycles, with one or more cooling levels [4–6].

However, it is typical in consumers which make use of refrigeration systems that different service specifications coexist. For instance freezing, refrigeration and air conditioning are typical applications, whose temperature requirements and load demands are diverse. In this context, the availability of multi-stage refrigeration cycles is useful, whose optimal energy efficiency, is intended.

For instance, a two-stage, two-load-demand refrigeration cycle is made up of two variable-speed compressors, two electronic expansion valves (*EEV*), and three heat exchangers (two evaporators and one condenser), as depicted in Fig. 1. Heat is removed at the evaporators (specifically from their secondary fluxes) by evaporating the refrigerant at low pressure and temperature. The temperature and pressure of the refrigerant are increased at the compression stage. Then, heat is transferred to the secondary flux at the condenser by condensing and subsequently subcooling the initially superheated refrigerant. Pressures and temperatures are usually quite different at the cooling and freezing chamber, in such a way that the booster compressor must raise the refrigerant pressure from the freezing evaporator output, and an additional valve is required in order to uphold the pressure difference between Evaporator 1 output and the discharge of the booster compressor, as shown in Fig. 1. The expansion valves allow the cycle to be closed by maintaining the pressure differential between the condenser and each evaporator.

In accordance with the research aims, a refrigeration plant, which can be configured to work with up to two compression stages and up to two load demands in addition to two alternative condensers, is located at the Department of Systems Engineering and Automation of University of Seville [7]. Optimization and