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Research Paper

Suboptimal hierarchical control strategy to improve energy efficiency of vapour-compression refrigeration systems

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HIGHLIGHTS

- Optimal control of vapour-compression refrigeration systems is intended.
- It is a one-degree-of-underactuation control problem.
- Nonlinear pointwise controllability analysis based on the phase portrait method.
- There is no full controllability and only a subspace can explored by any controller.
- A suboptimal hierarchical control strategy is proposed to approach the optimum.

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ABSTRACT

This paper deals with optimal control of vapour-compression refrigeration systems, where only the compressor speed and the expansion valve opening are considered as manipulable inputs. Any given cycle is completely defined by a three-variable set, thus it is an underactuated control problem. The controllability analysis presented by the authors in a previous work applying linear theory is extended to a pointwise nonlinear analysis based on the phase portrait method. It is concluded that there is no full controllability and only a two-dimensional subspace of the three-dimensional solution space can be explored. A suboptimal hierarchical control strategy is proposed, where an online optimizer explores the two-dimensional controllable subspace to generate the reference on the degree of superheating, which, along with the cooling demand set point and the uncontrolled state, defines the cycle in steady state. The uncontrolled state is, by definition, not manipulable and defines the maximum achievable efficiency. Some simulation results comparing the proposed control architecture with other strategies studied in the literature are included, regarding the energy efficiency achieved in steady state and the dynamic behaviour of the controlled variables, where the controllability issues are highlighted.

1. Introduction

Refrigeration systems are widely used, not only in applications for private consumers, but also in industrial facilities. Most of these systems, including air conditioning equipments and refrigerators, work the same way: they utilise the inverse Rankine cycle to remove heat from a cold reservoir (i.e. a cold storage room) and transfer it to a hot reservoir, normally the surroundings. Although air conditioning systems are technologically very different from refrigerators, as well as the disturbances affecting the system, the refrigerant thermodynamic cycle is essentially the same. This work is focused on food refrigerators and freezers, although the conclusions and the developed optimisation and control technique is expected to have an application to other refrigeration systems with minor modifications.

A great deal of energy is used by these systems, which negatively impacts energy and economical balances. For instance, nowadays almost every household uses at least one refrigeration system for domestic food preservation. Based on statistical data, in 2008, there were on average 1.27 and 0.54 refrigerators and freezers per household in Canada, respectively [1]. This indicates that a minimum of about 24 million household refrigerators and freezers are currently in use in Canada [2], which confirms that a huge amount of energy consumption is required for their operation. The most recent Residential Energy Consumption Survey (RECS) shows that refrigeration systems (including air conditioners) represent 28% of home energy consumption in the United States [3]. Moreover, supermarkets and department stores are also

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Nomenclature

Latin symbols

A_v	expansion valve opening [%]
$\boldsymbol{B}$	linear model input matrix ($\boldsymbol{B} \in \mathbb{R}^{3\times 2}$)
b_{ij}	element {i,j} in matrix $\boldsymbol{B}$
COP	Coefficient of Performance
$\boldsymbol{D}$	linear reduced model input vector ($\boldsymbol{D} \in \mathbb{R}^2$)
d_i	element {i} in vector $\boldsymbol{D}$
E	mechanical energy [W h]
EER	Energy Efficiency Ratio
$\boldsymbol{e}$	error vector ($\boldsymbol{e} \in \mathbb{R}^2$)
$\boldsymbol{f}$	force function ($\boldsymbol{f} \in \mathbb{R}^3$)
f_i	element {i} in force function $\boldsymbol{f}$
$\boldsymbol{G}$	Jacobian matrix
h	specific enthalpy [J kg^{-1}]
IP	initial point
J	cost function
k	discrete time step
$\dot{m}$	mass flow [kg s^{-1}]
NLF	nonlinear function
N	compressor speed [Hz]
N_c	control horizon
N_p	prediction horizon
N_u	number of manipulable inputs
N_y	number of controlled outputs
P	pressure [bar]
$\dot{Q}$	cooling power [W]
$\boldsymbol{Q}$	tracking error weighting matrix ($\boldsymbol{Q} \in \mathbb{R}^{2\times 2}$)
q	vapour quality
$\boldsymbol{R}$	control effort weighting matrix ($\boldsymbol{R} \in \mathbb{R}^{2\times 2}$)
$\mathbb{S}$	controllable subspace
T	temperature [°C]
T_{SH}	degree of superheating [°C]
t	time
$\boldsymbol{u}$	input vector ($\boldsymbol{u} \in \mathbb{R}^2$)
V_R	heat exchanger internal volume [m^3]
$\dot{W}$	mechanical power [W]
$\boldsymbol{w}$	external and internal input vector ($\boldsymbol{w} \in \mathbb{R}^4$)
$\boldsymbol{x}$	state vector ($\boldsymbol{x} \in \mathbb{R}^3$)
$\boldsymbol{y}$	output vector ($\boldsymbol{y} \in \mathbb{R}^2$)
$\boldsymbol{Z}$	coefficient matrix ($\boldsymbol{Z} \in \mathbb{R}^{3\times 3}$)
z_{ij}	element {i,j} in coefficient matrix $\boldsymbol{Z}$

Greek symbols

$\bar{\gamma}$	mean void fraction
δ	distance to the controllable subspace
ζ	fraction of the total heat exchanger length
$\boldsymbol{\Lambda}$	projection matrix ($\boldsymbol{\Lambda} \in \mathbb{R}^{2\times 3}$)
ρ	density [kg m^{-3}]
$\boldsymbol{\phi}$	measurable variable set ($\boldsymbol{\phi} \in \mathbb{R}^3$)
$\boldsymbol{\chi}$	optimisation variable set ($\boldsymbol{\chi} \in \mathbb{R}^3$)
$\boldsymbol{\psi}$	deviation with respect to the desired state ($\boldsymbol{\psi} \in \mathbb{R}^3$)
ψ_i	element {i} in vector $\boldsymbol{\psi}$

Subscripts and superscripts

c	condenser
ctrl	control
comp	compressor
cycle	vapour-compression cycle
e	evaporator
FB	feedback
FF	feedforward
f	saturated liquid
forced	forced response
free	free response
g	saturated vapour
in	inlet
inv	inverse
it	iterative
*m*1	condenser *mode* 1
*m*2	condenser *mode* 2
max	maximum
min	minimum
out	outlet
PNMPC	Practical Nonlinear MPC
past	past and current values
predict	predicted values
ref	reference
sc	subcooled liquid zone
sec	secondary flux
settle	settling
sh	superheated vapour zone
start	control start
tp	two-phase zone

great energy consumers, since official reports estimate that the average intensity for grocery stores is around 500 kW h/m^2 a year in USA, which means more than twice the energy consumed by a hotel or an office building per square metre [4]. In Denmark, around 4500 supermarkets, whose installed cooling capacity ranges from 10 to 200 kW, depending on the supermarket size, consume more than 550,000 MW h per year. This corresponds roughly to 2% of the entire electricity consumption in the country.

Furthermore, environmental issues related to refrigerants and energy waste merit consideration. The pressing concern for shortage of fossil energy sources, as well as the slow development of renewable energy alternatives, has caused in recent years increasing efforts to increment energy efficiency, while reducing the environmental impact of current vapour-compression systems. Refrigerants play an important role in food-transport refrigeration systems, among others. Improvement of the quality of life is resulting in more refrigerated food transport, which means more refrigerants used and more CO_2 emitted due to energy consumption. According to some statistical reviews in 2002, there were over 1,000,000 refrigerated food trucks and over 400,000 refrigerated food containers in the world [5]. This refrigerated food transport results in large amounts of CO_2 emissions. Thus, even small performance enhancement of these appliances brings huge amounts of energy saving and reduction of CO_2 emissions. In this context, system control and optimisation have potential to improve the performance of current refrigeration systems and they are expected to contribute to energy efficiency enhancement.

Fig. 1 shows a canonical one-compression-stage, one-load-demand refrigeration cycle. In this paper a particular application is considered, where the cycle is expected to provide a certain cooling power to a continuous flow entering the evaporator as secondary flux. Neither the mass flow nor the inlet temperature of such secondary flux are to be controlled by the refrigeration system, since they are managed by another high-level controller. Therefore, the cooling demand can be expressed as a reference on the outlet temperature of the evaporator secondary flux, where the mass flow and inlet temperature behave as measurable disturbances to the refrigeration system. The difference with respect to the conventional case analysed in the literature relies on considering the secondary mass flow as non-manipulable, since conventionally the cooling demand is merely a certain thermal power to be provided to the secondary flux for any mass flow, which is used