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Modeling, simulation and optimization of a vapor compression refrigeration system dynamic and steady state response

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HIGHLIGHTS

- We introduce a transient vapor compression refrigeration (VCR) system model.
- Dimensionless groups are identified, and results presented in normalized charts.
- Model adjustment and experimental validation are conducted.
- A system performance comparison is done for refrigerants R12, R134a, and R1234yf.
- Optimal heat transfer area allocation is found for maximum system performance.

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ABSTRACT

This paper introduces a dimensionless simplified mathematical model of a vapor compression refrigeration (VCR) system, in order to optimize the system dynamic response. The model combines principles of thermodynamics, heat and mass transfer applied to the system components with empirical correlations, assigning thermodynamic control volumes to each component, which yield a system of ordinary differential equations with respect to time that is integrated explicitly and accurately with low computational time. Appropriate dimensionless groups are identified, and the results are presented in the form of normalized charts for general application to similar systems. Experiments were conducted using an industrial chiller that was instrumented for real time data acquisition in order to determine model adjustment parameters through the solution of the inverse problem of parameters estimation (IPPE) for a 300 W thermal load. The model was then experimentally validated using the adjusted parameters by direct comparison of simulation results to the system measured thermal response for a 1200 W thermal load with good agreement. After experimental validation, the study addressed the optimization of the heat exchangers heat transfer area inventory for minimum pull down time. A system performance comparison is conducted for three refrigerant fluids: (i) a banned refrigerant (R12); (ii) its original ozone depletion harmless substitute, but with a high global warming potential, GWP (R134a), and (iii) one of the current substitutes of R134a, i.e., R1234yf. The dynamic results show that, for a system originally designed for R12, substitution with R1234yf depicts a closer performance to R12 than R134a. The optimal configuration that leads to steady state maximum system first (or coefficient of performance, COP), and second law efficiencies is also pursued. The normalized results for refrigerants R12, R134a, and R1234yf show that an optimal heat transfer area distribution in both evaporator and condenser, represented by an evaporator to total system heat exchanger area ratio $x_{4,\mathrm{opt}} \cong 0.55$, leads the system to minimum pull down time and maximum system 2nd law efficiency, whereas $x_{4,\mathrm{opt}} \cong 0.4$ to maximum COP, when the heat exchangers global heat transfer coefficients are of the same magnitude. The difference in the optimum location when the objective function is the pull-down time (and second law efficiency) and COP is due to the fact that the first law analysis does not fully capture the thermodynamic losses due to changes in heat exchangers' areas, which stresses the importance of a system second law assessment for realistic results. However, changes in the optima location are observed when the ratio of heat exchangers global heat transfer coefficients departs from 1. The obtained maxima and minima are sharp in the searching interval ($0.1 \leqslant x_4 \leqslant 0.9$), showing up to a 189.6% and 93.37% variation, $\left(\frac{f_{\max}-f_{\min}}{f_{\max/\min}}\right) \times 100$,

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Nomenclature

A	area, m^2
$\tilde{A}_{set}$	dimensionless set point of expansion valve opening area corresponding to ΔT_{dsh}
a	constant for a gas in the van der Waals equation, kPa m^6 $kmol^2$
$a_1, \ldots, a_5$	polynomial coefficients, Eqs (14) and (15)
b	constant for a gas in the van der Waals equation, m^3 $kmol^{-1}$
b_1, b_2	polynomial coefficients, Eq. (39)
c_1, c_2, c_3	polynomial coefficients, Eq. (25)
c	specific heat, J kg^{-1} K^{-1}
c_0	dead volume ratio in the compressor
C_v	compressor valve coefficient
CV	control volume
E	energy, J
G	thermostatic valve adjustment constant, m^2 K^{-1}
h	specific enthalpy, J kg^{-1}
M	molecular weight, kg $kmol^{-1}$
m	mass, kg
$\dot{m}$	mass flow rate, kg s^{-1}
n	compressor polytropic coefficient
p	pressure, N m^{-2}
$\dot{Q}$	heat transfer rate, W
R	reliability coefficient in polynomial fitting
$\bar{R}$	universal gas constant, 8.314 kJ $kmol^{-1}$ K^{-1}
rps	compressor revolutions per second
s	evaporator to condenser global heat transfer ratio, Eq. (3)
t	time, s
T	temperature, K
U	global heat transfer coefficient, W m^{-2} K^{-1}
v	specific volume, m^3 kg^{-1}
v_0	specific volume of the refrigerant at (p_0, T_0)
V_c	compressor volumetric displacement, m^3
$\dot{W}$	power, W
x	heat exchanger area fraction, Eq. (4)
$\vec{x}$	vector of dimensionless system variables
y	space-averaged quality
z	dimensionless wall thermal conductance, Eq. (7)

Greek symbols

Δ	variation, difference
η	efficiency
η_v	compressor volumetric efficiency
μ	dimensionless conversion factor, Eq (44)
ξ	dimensionless conversion factor, Eq. (41)
τ	dimensionless temperature, Eq. (5)
ϕ	dimensionless compressor speed
ψ	dimensionless heat capacity rate, Eq. (5)

Subscripts

a	air
a, b, c, d	points in the VCR system cycle, Fig. 1
cp	compressor
crit	critical point
C	Carnot
dsh	desired degree of superheat
GS	global system
heat	heating mode
l	liquid
p	constant pressure
r	refrigerant
sat	saturation
sc	subcooled liquid zone in the condenser
sh	superheated vapor
set	set point
v	constant volume; vapor
w	wall
0	exterior ambient; reference conditions, initial conditions
1, 2, ..., 7	control volume number
2p	two phase zone (refrigerant liquid and vapor mixture)
I	first law of thermodynamics
II	second law of thermodynamics

Superscript

$\sim$	dimensionless variable

in which f is either the calculated system pull down time or second law efficiency, respectively, which was observed with refrigerant R1234yf, which points out their importance for actual HVAC–R (heating, ventilation, air conditioning and refrigeration) vapor compression systems design.

1. Introduction

The statistics on the total electrical energy consumption in the United States of America, USA, in 2009 [1] show that the residential, commercial and industrial sectors were responsible for 38%, 36%, and 26% of that total, respectively. The heating, ventilation, and air conditioning (HVAC) and refrigeration (R) sectors are responsible for 42% and 7% of the electrical energy consumption of the residential sector in 2008, respectively [2]. For the commercial sector, HVAC and R were responsible for 38% and 7% of the sector's electrical energy consumption in 2008, respectively [3]. Facility heating, ventilating, and air conditioning (HVAC), and refrigeration (R) systems account for about 9% of all industrial electricity use, which somehow varies according to the type of industry [4]. Using such data, it is possible to calculate that the HVAC–R systems electrical energy consumption, in the residential, commercial and industrial sectors, correspond to 18.62%, 16.20%, and 2.34% of the total electrical energy consumption in the USA, respectively, which adds up to 37.16% of the total electrical energy consumption in the country. Hence, it is demonstrated that HVAC–R systems affect considerably the USA energetic matrix, which could be reasonably extrapolated to the rest of the world, accounting for regional particular differences. As a result, any technical or scientific action that aims at the energy consumption reduction of HVAC–R systems will greatly contribute to the search of solutions to the current growing world energetic demand.

Additionally, HVAC–R systems are widely used for the thermal management of large manmade devices, which comprise the so called field of systems engineering. Modeling and simulation are tools that are currently used for such complex design [5]. Systems