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Thermoeconomic analysis and optimization of residential split-type air conditioners

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HIGHLIGHTS

► A model-driven design methodology for air conditioners is presented.
► Model predictions for cooling capacity and COP are within ±6% error bounds.
► An optimization tool was devised to size the heat exchangers and the compressor.
► The COP can be increased by 7% if the cost is held fixed.
► Cost savings of 33% were achieved when the system COP was held constant.

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ABSTRACT

A simulation-based optimization methodology for designing unitary residential air conditioners with focus on both energy performance enhancement and cost savings is presented. A steady-state system simulation model was put forward for a 2.5-ton nominal cooling capacity split-type air conditioning unit operating with R-410A as the working fluid. The model predictions for cooling capacity, sensible heat ratio (SHR) and coefficient of performance (COP) were compared with experimental data, when it was found that the model is able to predict the experimental trends within a ±6% error band. The model was then used to find out the condenser and evaporator geometries (face area, heat transfer area) that enhance the system COP for a fixed cost. On one hand, it was observed that the COP can be increased by 7% if the cost is held fixed. On the other hand, cost savings of 33% were achieved when the system COP was held constant.

1. Introduction

Albeit the HCFC working fluids have already been phased out in most of the developed countries [1], R-22 replacement for static air conditioning applications is still an up-to-date concern in emerging countries like Brazil as the phase-out deadline is approaching. Among several replacement candidates under consideration, the most prospective ones seem to be the binary quasi-azeotropic mixture R-410A and propane (R-290) [2]. The former is seen as a drop in solution for most small-capacity air conditioning units – although the working pressures are much higher than those found when R-22 is used – whereas the latter will require a new system design aimed at charge minimization because of flammability issues. As learned from the CFC phase out in the late 1980s [3], a series of component matching and performance enhancement studies will be required before new air conditioning systems running with either R-410A or R-290 come onto the Brazilian market.

In general, the component matching exercise is carried out by testing a prototype according to a standardized test procedure [4]. The experimental tests are costly and time demanding not only by themselves, but also because of the cost and time associated with prototype assembling and transporting. It has been advocated in the open literature that the development costs of vapor compression refrigeration systems may be reduced if proper simulation tools are adopted [5–7] as hundreds of design options can be evaluated within a few minutes without the need of a large number of prototypes.

Both steady-state [8–12] and transient [13–18] simulation models have been proposed in the past decades for predicting the performance of air conditioning and heat pump systems. In spite of the large number of publications in the field, optimization studies are uncommon, and the few available are focused on a single

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Nomenclature

Roman

A	area, m^2
a, b	coefficients of Eqs. (2) and (3)
C	cost, $
c	specific cost, $/kg
COP	coefficient of performance, W/W
c_p	specific heat, J/kg K
e	coefficients of Eq. (18)
f	friction factor
G	maximum air mass flux, kg/s m^2
h	specific enthalpy, J/kg
h_{lv}	latent heat of condensation, J/kg
Le	Lewis number
m	mass flow rate, kg/s
M	material mass, kg
N	number of tubes or fins
p	pressure, Pa
Q	heat transfer rate, W
SHR	sensible heat transfer ratio
t	temperature, K
V	air flow rate, m^3/s
W	power, W

Greek

α	heat transfer coefficient, W/m K
β	compressor scaling coefficient
ε	heat exchanger effectiveness
η	surface effectiveness
ρ	specific mass, kg/m^3
ω	humidity ratio, kg_s/kg_a
ϕ	relative humidity

Subscripts

a	dry air
Al	aluminum
atm	atmospheric
c	condenser
Cu	copper
duct	ductwork
e	evaporator
f	fin
face	heat exchanger face area
fan	condenser or evaporator fan
i	inlet
k	compressor
lat	latent
lo	longitudinal tubes
m	moist air
min	minimum free flow passage
o	outdoor
r	refrigerant
sat	saturation
sc	subcooling degree
sen	sensible
sh	superheating degree
t	tube
tr	transversal tubes
wet	wet surface
x	heat exchanger (condenser or evaporator)
y	air condition (dry or moist)

component [19–22]. Recently, Waltrich et al. [6] presented a design methodology for refrigeration cassettes focused on both energy performance and cost savings. The heat exchangers (condenser and evaporator) geometries and the compressor capacity were optimized, leading to simultaneous COP and cost improvements. Following a similar approach, Negrão and Hermes [7] carried out an energy and cost optimization of a household freezer taking into account not only the refrigeration system characteristics (e.g., heat exchangers, compressor), but also the compartment insulation. So far, there is no evidence in the open literature of a multi-component optimization of air conditioning equipment. This is, therefore, the main focus of the present study.

2. Simulation model

The refrigeration system under study is a unitary (split-type) residential air conditioner comprised of a hermetic scroll compressor, two fan-supplied tube–fin heat exchangers (condenser and evaporator), and a thermostatic expansion device, as depicted in Fig. 1. A 4-way valve is used to turn the system into a heat pump in case the heating mode is required. In this work, the system was modeled and tested in the cooling mode only. More details on the constructive aspects of the system under analysis can be found in [23].

In general, the system simulation model followed the approach originally developed by Negrão and Hermes [7] for refrigerators and freezers. However, in the present study the refrigerated room does not need to be modeled as the indoor and outdoor temperatures are standardized conditions [4]. On the other hand, the latent heat transfer in the evaporator coil should be taken into account.

The compressor was modeled based on mass and energy balances following a lumped approach [24]. Thus, the refrigerant enthalpy at the compressor discharge was obtained from [25]:

$$h_2 = h_1 + W_k/m_r \quad (1)$$

where m_r and W_k are the refrigerant mass flow rate and the compression power, respectively, being both calculated as bi-quadratic polynomial functions of the condensing (t_c) and evaporating (t_e) temperatures [26]:

$$m_r = \beta \sum_{i=0}^{2} \sum_{j=0}^{2} a_{ij} t_e^i t_c^j \quad (2)$$

$$W_k = \beta \sum_{i=0}^{2} \sum_{j=0}^{2} b_{ij} t_e^i t_c^j \quad (3)$$

where β is a scaling factor used during the optimization task ($\beta = 1$ for the compressor as built), whereas the coefficients a_{ij} and b_{ij} were fitted to experimental data obtained from the compressor maps through the least squares method [26].

Bearing in mind that the best heat exchanger simulation model is the one that provides the desired results with a minimum computational effort [6], some minor effects were overlooked (i.e., two-phase heat transfer coefficient, refrigerant-side pressure drop, coil circuit), so that the modeling efforts could be placed on factors that actually play important roles on the system performance, such as heat exchanger geometry (e.g., number of fins and tubes), air-side pressure drop, and fan pumping power [27]. Thus, the heat exchangers were