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Applying refrigerant mixtures with thermal glide in cold climate air-source heat pumps

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ABSTRACT

Heat pumps, as a means of achieving significant energy reductions, have attracted a great deal of attention for decades. However, the main challenge remains improving their performance in cold climates. This paper represents the first step of a larger research project for the implementation of the zeotropic refrigerant mixtures in order to increase the performance of residential air-source heat pumps in cold climates. A detailed screening heat pump model is developed and used to assess the performance of zeotropic refrigerant mixtures. A group of pure refrigerants are selected and their potential mixtures are studied. The performance of these mixtures is compared in order to find suitable zeotropic refrigerant mixtures for cold climate residential applications. The main goal of this paper is to illustrate the possibility of applying environmentally friendly zeotropic refrigerant mixtures in conventional heat pumps, with minimal changes in the components, in order to improve their performance.

1. Introduction

Heat pumps represent the only end-use heating technology that has a coefficient of performance greater than one. However, the efficiency of air-source heat pumps decreases drastically at low cold source temperature which represents a high barrier to their adoption in cold climates. The use of mixtures of refrigerants with the aim of increasing the COP and heating capacity at lower temperatures by taking advantage of the thermal glide is an option that has been little studied so far. Higher heat pump performance can be achieved by approaching the conventional heat pump cycle to the Lorenz cycle. More exactly, a performance improvement is obtained by having a temperature variation during the constant pressure condensation and evaporation, rather than a constant temperature in conventional cycles. This improvement is mainly achieved by decreasing heat loss irreversibilities in the heat exchangers, as a result of reducing the thermal pinch.

The main goal of the present study is to find a zeotropic refrigerant mixture properly selected so that it enables good heating performance (capacity, COP) at low temperatures for household applications. This mixture must require minimum changes in the system components of conventional residential heat pumps. Besides having desirable thermophysical properties, it must be non-toxic, non-flammable, oil soluble, and have low GWP. Currently, there is no acceptable pure refrigerant that satisfies all these requirements. In this paper, a methodology used for selecting appropriate mixtures, preliminary results and potential gains for cold climate heat pumps are presented.

Over the past 20 years, developments in the field of refrigerants have been limited to finding suitable substitutes for CFCs and HCFCs in response to the Montreal Protocol and re-introducing natural refrigerants (hydrocarbons, ammonia, CO_2) in response to the Kyoto Protocol.

McLinden [1] presented a simplified model to provide performance ratings of refrigerants and refrigerant mixtures operating in vapor compression cycle called Cycle 7. His model consists of seven cycle state points. This cycle was subsequently improved by adding more states through a suction-line heat exchanger and a change of the compressor model. This new model was called Cycle 11 [2]. Pannock and Didion [3] applied Cycle 11 to investigate the best possible binary zeotropic refrigerant mixture as a replacement of R-22 for residential heat pump applications. They found that the mixtures of R-32/R-134A and R-32/R-152A perform better than R-22 with respect to COP and volumetric capacity for certain composition range and using a counter-flow heat exchanger.

Rice [4] developed a simplified model called BICYCLE for simulation of the pure and mixed refrigerants in order to evaluate R-22 alternatives. Four different refrigerant mixtures from small to large temperature glide were selected for evaluation. He assumed the

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Nomenclature

A	area, m^2
B	scaling factor
Bo	boiling number
C_p	specific heat capacity, J/(kg K)
c_v	compressor clearance volume ratio
D	diameter, m
E	two-phase convection multiplier
f	friction factor
g	acceleration due to gravity, m/s^2
G	mass flux, $kg/(m^2s)$
h	heat transfer coefficient, $W/(m^2 K)$
H	enthalpy, kJ/(kg K)
h_{LV}	differential latent heat, J/kg
k	thermal conductivity, W/(mK)
L	length/fin height, m
m°	mass flow rate, kg/s
n	polytropic index
N	number of fins
Nu	Nusselt number
P	perimeter, m^2
Pr	Prandtl number
Q	heat transfer rate, W
QH	heat derived by the condenser, W
r	radius, m
R	thermal resistance, m^2 K/W
Re	Reynolds number
T	temperature, °C
U	overall heat transfer coefficient, $W/(m^2 K)$
V_{cp}	compressor inlet volume, m^3
W_{COMP}	compressor work, W
W_F	air fan work, W
x	vapor quality

Greek symbols

β_L	liquid phase mass transfer coefficient, m/s
γ	isentropic index, C_p/C_v
Δ	difference
ΔT_1	temperature difference at the inlet
ΔT_2	temperature difference at the outlet
ε	small value
η_f	fin efficiency
η_o	overall surface efficiency
η_p	polytropic efficiency
η_v	volumetric efficiency
λ	latent heat, J/kg
μ	viscosity, kg/(ms)
ρ	density, kg/m^3
ω	compressor speed (rps)

Subscripts

bub	bubble-point
c	cross sectional
CND	condenser
dew	dew-point
EVAP	evaporator
f	fin/fluid
i	segment number/inner
l	liquid
mix	mixture
o	outer
r	refrigerant
suc	compressor suction
t	total
tp	two-phase
vap	vapor

same air flow rate, UA values and sensible heat ratios as for R-22 in his simulations, and only the compressor displacement was adjusted to obtain the same design capacity. He concluded that the potential of COP gains are 1.5–2 times larger in cooling mode than in heating.

Haselden and Chen [5] developed a simulation program for mixed refrigerant air conditioning. They stated they were the first to develop a complete mixed-refrigerant simulation model. They neglected the pressure drop in the superheated vapor and subcooled liquid flow. Moreover, they did not consider the volumetric efficiency of the compressor in their simulations. The volumetric efficiency is an important factor in determining the refrigerant mass flow rate and accordingly the system performance at different working conditions.

Ragazzi and Pederen [6] studied the thermodynamic optimization of various evaporator designs with zeotropic refrigerant mixtures using an irreversibility-based function. Their main goal was to find a replacement for R-22. They showed that the zeotropic mixtures are less irreversible than the pure refrigerants with cross-counter-flow heat exchanger.

Bensafi et al. [7] developed a computational model for the detailed design of plate-fin-and-tube heat exchangers using pure and mixed refrigerants. They predicted the pressure drop with the error of the order of 30%, and heat transfer rates were predicted with discrepancies less than 3%.

Maczek et al. [8] investigated ternary zeotropic mixture of CO_2/R-32/R-134A as a possible alternative to R-22. They found that the mixture with mass fraction of 7/31/62 increases both the COP and capacity by 2.5% and 18.6% respectively. A comprehensive review on recent development in new refrigerant mixtures for heating and cooling applications is presented by Mohanraj et al. [9].

This study is the first step of a larger research project. In this phase of the study, a detailed screening heat pump model is developed and used to investigate the performance of a group of selected refrigerant mixtures in order to find a suitable refrigerant mixture for cold climate applications. In the next phase of the project, the mixtures that are found to have a desirable performance potential in the current phase will be experimentally studied.

2. Theoretical model

A numerical model for detailed simulations of air-source heat pumps compatible with both pure and mixed refrigerants is presented. In order to make the calculations simple, the following assumptions are made:

- Steady-state flow conditions.
- One-dimensional flow for the refrigerant inside tubes and air across the coils.
- Negligible heat losses to the environment.
- The flow across the expansion valve is isenthalpic.

The modeling of each component is discussed separately.