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Modeling and application of direct-expansion solar-assisted heat pump for water heating in subtropical Hong Kong

T.T. Chow [a,*], G. Pei [b], K.F. Fong [a], Z. Lin [a], A.L.S. Chan [a], M. He [a]

[a] Building Energy and Environmental Technology Research Unit, Division of Building Science and Technology, College of Science and Engineering, City University of Hong Kong, Tat Chee Avenue, Kowloon, Hong Kong SAR, China
[b] Department of Thermal Science and Energy Engineering, University of Science and Technology of China, Hefei, China

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ABSTRACT

Direct hot water production consumes about 4% of the total energy use in Hong Kong, and about 20% when considering only the domestic sector. For water heating the energy sources are mostly town gas, liquefied petroleum gas and electricity. The use of heat pump or solar water heating, particularly the solar-assisted heat pump options, is not popular. In this paper, the potential application of a unitary type direct-expansion solar-assisted heat pump (DX-SAHP) system was examined. A numerical model of the DX-SAHP system was first introduced. From the simulation results with the use of the Typical Meteorological Year (TMY) weather data of Hong Kong, the system was found achieving a year-average coefficient of performance (*COP*) of 6.46, which is much better than the conventional heat pump system performance. The potential use of DX-SAHP therefore deserves further evaluation.

1. Introduction

According to the energy end-use statistical data (1996–2006) published by the Hong Kong Special Administrative Region Government [1], direct hot water production consumed about 4% of the total energy use in the city, and this was about 20% when considering only the domestic sector. Other than the domestic sector, hot water is consumed in the commercial sector (like restaurants, hotels, education, and health premises) and the industrial sector (like textiles, food and beverages, and other industrial processes) as well. Cooking, which accounted for 20% of the energy use in the residential buildings, also had a significant indirect demand on hot water. The heat energy sources are mostly town gas, liquefied petroleum gas and electricity. The use of heat pump or solar water heating, particularly the solar-assisted heat pump (SAHP) options, is not popular.

In the North America, numerical studies on the performance of SAHP systems were performed as early as in the 1970s and 1980s, such as the works of Freeman et al. [2] and Chandrashekar et al. [3]. In Asia, the theoretical and experimental SAHP studies were performed in the 1990s in Japan [4]. In Taiwan since 1997, several generations of the system equipment have been introduced and reported [5,6]. In mainland China, Kuang et al. [7] performed analytical and experimental studies on direct-expansion solar-assisted heat pump (DX-SAHP) as applied in Shanghai. The effects of various parameters under constant compressor speed were investigated. Kuang and Wang [8] further developed a multi-functional domestic DX-SAHP system, which was able to offer multi-fold functions to residences at low costs, including space heating in winter, space cooling in summer, and hot water supply for the whole year. Exergy analysis was carried out in the experimental works of Li et al. [9] and Dikici and Akbulut [10]. The above studies, however, did not cover the energy performance of SAHP in subtropical locations, like Hong Kong. In Hong Kong, hot water is in need throughout the year. The multi-functionality mentioned above is desirable but not essential, since the space heating demand in winter is very occasional. The low utilization rate makes the option not economically attractive.

While the "indirect-expansion SAHP" system, with the schematic diagram shown in Fig. 1a, has an intermediate heat exchanger between the solar water circuit and the chilled water circuit, the DX-SAHP design in Fig. 1b uses a two-phase solar collector to act directly as the evaporator. The evaporator–collector configuration allows a reduction in the number of components in use. On the other hand, the circulation of refrigerant (instead of water) in the collector tubes prolongs the system life. This is because the corrosion and nighttime freeze-up problems, which are usually encountered in water collectors, are virtually eliminated. Through a case study, Li and Yang [11] evaluated the economic characteristics and global warming impacts of different kinds of water heating

* Corresponding author. Tel.: +852 27887622; fax: +852 27889716.
E-mail address: bsttchow@cityu.edu.hk (T.T. Chow).

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Nomenclature

A	heat exchange surface area, m^2
C	specific heat, J/kg K
COP	coefficient of performance, –
D	diameter of cylinder bore, m
d	constants, –
H	convective heat transfer coefficient, W/m^2 K
h	specific enthalpy, J/kg
I	instantaneous solar irradiance, W/m^2
k	thermal conductivity, W/m K
L	length, m; pitch, m
M	mass, kg
$\dot{m}$	mass flow rate, kg/s
W	power consumption, W
N	rotating speed, rev/s
n	number of cylinders, –
p	pressure, Pa
Q	heat flow rate, W
R	specific gas constant, kJ/kg K
Ri	Riedel number, –
S	piston stroke, m
s	specific entropy, J/kg K
T	temperature, K
t	time, s
U	U-value or heat loss coefficient, W/m^2 K
v	specific volume, m^3/kg
$\dot{v}$	volume flow rate, m^3/s
x	distance, m
Z	compressibility factor, –

Greek symbols

δ	thickness, m
λ	compressor correction coefficient, –
ρ	density, kg/m^3
$(\tau\alpha)$	effective absorptivity, –

Subscripts

a	ambient
B	sub-cooled
b	boiling; absorber plate
c	critical; convective
cc	condenser coil
com	compressor
e	evaporating
i	inside or inlet
K	condensing
L	leakage
l	loss
o	outside or outlet
p	pressure
R	refrigerant
r	reduced
s	saturated vapor
T	temperature
t	tube
tk	tank
v	volume
w	water

systems in Hong Kong. Only a simple evaluation procedure was adopted. Assuming a coefficient of performance (*COP*) of 4.0, DX-SAHP was found more economical than the indirect-expansion type.

Introduced below is a detailed numerical evaluation of the application potential of a unitary DX-SAHP system for water heating in Hong Kong, as a good representation of the subtropical region. As the SAHP component model is not readily available at the public domain system simulation software, the numerical model of a domestic-scale DX-SAHP system was first developed. Hourly variation of *COP* has been taken into account. Then based on the typical year-round hourly weather data of Hong Kong, the annual energy performance of the system was evaluated.

2. System description

The thermal processes in a DX-SAHP system adopts the vapor compression refrigeration cycle, which is shown on the *T*–*s* diagram in Fig. 2. The working fluid (refrigerant) passes through the compressor, the condenser, the capillary tube and the evaporator in turn, with the corresponding thermodynamic state points denoted 1–5. In our study, the condenser is basically a water storage tank with a submerged heat transfer coil for refrigerant flow from state points 2 to 3. Liquid refrigerant vaporizes at the capillary tube to state 4 and evaporates at the evaporator–collector, of which point 5 indicates the saturated vapor condition. Superheated refrigerant vapor at state 1 enters the reciprocating compressor and leaves at a higher temperature and pressure condition located as point 2. Table 1 gives the indicative equipment specification of the unitary system being studied, with R134a as the working fluid.

3. Numerical modeling and simulation

3.1. The vapor compression cycle

A dynamic system model has been developed for the numerical analysis. The thermodynamic properties of R134a at various process states of the vapor compression cycle were determined by the mathematical models referred by Liu et al. [12]. A brief description is given below.

3.1.1. Fluid pressure, temperature and density

3.1.1.1. Superheated vapor. Through the use of the compressibility factor (Z), the fluid pressure (p) at temperature (T) and specific volume (v) can be expressed as

$$p = \frac{RT}{v}(1000 * Z) = 1000 * RT_c T_r \rho Z \tag{1}$$

where R is the specific gas constant, T_c is the critical temperature, T_r is the reduced temperature (=T/T_c) and ρ is the fluid density. Z can be given as a function of the polynomials of ρ and $(1/T_r)$, in that [12]

$$Z = 1 + \sum_{i=1}^{3}\left(\sum_{j=0}^{3} b_{i,j}\Big/T_r^j\right)\rho^i \tag{2}$$

where $b_{i,j}$ are the fluid specific constants.

3.1.1.2. Saturated vapor. The saturated pressure (p_s) of a saturated vapor with critical pressure (p_c) is given by

$$p_s = p_c \exp[Ri \ln T_r + (Ri - 4 + P_\alpha)\varphi(T_r)] \tag{3}$$

where P_α is an empirical parameter. Ri is known as the Riedel expression [13] given by