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Performance of hybrid refrigeration system using ammonia

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HIGHLIGHTS

- A mechanical compressor was combined with up to 5 pair bed adsorption generator.
- The hybrid's refrigerating capacity ranges between 4 kW and 24 kW.
- The driving temperature varies from 100 °C to 250 °C.
- The driving temperature rises significantly at small number of adsorbent beds.
- The system preformed best at low driving temperatures with a 5 pair adsorbent bed.

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ABSTRACT

This paper investigates the performance of a hybrid refrigeration system that combines sorption–conventional vapour compression refrigeration machine driven by dual source (heat and/or electricity). The dual source makes the system highly flexible and energy efficient. The ammonia refrigerant (R717) is used in both adsorption and associated conventional refrigeration cycles. The model of thermal compressor corresponds to a multiple pair of compact adsorption generators operating out of phase with both heat and mass recovery for continuous cooling production and better efficiency. Each generator is based on a plate heat exchanger concept using the activated carbon–ammonia pair. The model of conventional vapour compressor is a reciprocating compressor from Frigopol. The hybrid refrigeration performances are presented mainly for ice making and air conditioning applications ($T_C = 40$ °C, -5 °C $< T_E <$ 20 °C). The exhaust temperature of the compressor (driving temperature for thermal compressor) varies from 90 °C to 250 °C. The results show a cooling production ranging from 4 kW to 12 kW with back-up mode (both cycles not operating simultaneously) and from 8 kW to 24 kW with complementary mode (both cycles operating simultaneously). The effective overall COP based on the total equivalent heat rate input varies from 0.24 to 0.76.

1. Introduction

According to the Climate Change Act 2008 the UK has to cut greenhouse gas emissions by 80 per cent below 1990 levels by 2050 and 34 per cent by 2020 [1]. To achieve a 34 per cent decrease by 2020 would require the reduction of 2010 total emissions (590.4 $MtCO_2e$) by 84.6 $MtCO_2e$. Carbon dioxide emissions accounted for 84 per cent of total UK greenhouse gas emissions in 2010, attributed to the energy supply sector (39%), road transport (22%), residential sector (17%) and business (15%). The total final consumption of energy was 159.1 $MtCO_2e$ in 2010 with the domestic (30.5%), transport (35%) and industrial (17.3%) sectors being the main consumers. Electricity accounted for 17.7% of the total energy consumption by final users and 21 per cent of the domestic energy consumption. Most of the electricity in the domestic sector is used for powering household appliances. In 2009 about 17% was used by cold appliances and approximately 14% for space heating. It is the main fuel used for air-conditioning of human dwellings and refrigeration.

Improvements in the efficiency of heating and cold appliances had a direct effect on energy consumption the last twenty years. The efficiency of cold appliances improved significantly since 1990 and by 2010 they consumed less electricity by an average of 52% in the UK. In fact since 1990 and by 2010 the electricity consumption by cold appliances in the domestic sector has reduced by 7.4%. Thus the improvement of the energy efficiency of cold appliances is still of primary importance in reducing their carbon footprint.

Apart from using electricity and therefore emitting indirectly carbon dioxide, the conventional refrigerating and heat pump systems (mechanical vapour compression technology) also use

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Nomenclature

A	heat transfer area (m^2)
c_p	specific heat ($J\ kg^{-1}\ K^{-1}$)
COP	coefficient of performance
D	cylinder diameter (m)
h	specific enthalpy ($J\ kg^{-1}$)
H	specific heat of sorption ($J\ kg^{-1}$)
L	length of stroke (m)
m	mass flow rate ($kg\ s^{-1}$)
M	mass (kg)
n	number of cylinders
N	rotation speed ($rad\ s^{-1}$)
P	pressure (Pa)
q	volumetric flow rate ($m^3\ s^{-1}$)
Q	capacity (W)
R	gas constant ($J\ kg^{-1}\ K^{-1}$)
t	time (s)
T	temperature (K)
U	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
v	specific volume ($m^3\ kg^{-1}$)
$\dot{w}$	specific work ($J\ kg^{-1}$)
W	mechanical power (W)
x	concentration ($kg\ kg^{-1}$)

Greek symbols

η	efficiency (%)
γ	ratio of specific heat
α	slope

Subscripts

a	adsorbate
ads	adsorption
C	condensing
c	carbon
e	refrigerating
E	evaporating
f	fluid
h	heat
in	inlet
isen	isentropic
LM	log mean
mc	mechanical compression
p	pressure
Sys	system
sat	saturated
v	volume
LM	log mean
out	outlet
p	pressure
pp	power plant
pt	power line transmission
w	wall
1–4	states in mech. vapour compression cycle

refrigerants that can be harmful on the environment causing depletion of the ozone layer and global warming phenomena. Sorption systems can be an alternative to the vapour compression ones because they are heat driven (using waste heat or solar energy) and utilise natural refrigerants (ammonia, water, alcohols) that have low or zero global warming potential (GWP) and zero ozone depletion potential (ODP).

However, sorption systems have some undesirable characteristics; they are usually big sized with high volume for given cooling capacity and have low COP. Recent advances though in adsorption technology have made possible producing more compact and lighter systems with higher efficiency [2,3]. Hybrid systems comprise sorption and vapour compression technology, putting together the advantages of both technologies and thus giving more flexibility in operation, tolerance under extreme temperature conditions and reduced cost. Research on hybrid systems that combine mechanical and thermal compressors has been focused on the concept of operating them in series [4,5]. Here we put forward the concept of parallel operation.

In this work the performance of a hybrid refrigeration system utilising ammonia (R717) is investigated. In particular, simulations of a thermal compressor (adsorption generator developed at Warwick University) and a conventional mechanical compressor (Frigopol [6], separating hood compressor 7-DLZC-1.5) are presented. The simulations were carried out at 40 °C condensing temperature and with evaporating temperatures ranging from −5 to 20 °C. Based on the results we identify how these two processes can be matched efficiently for operating under different conditions and for different applications (i.e. air conditioning and ice production).

2. Model elaboration

The adsorption model can simulate the performance of a thermal compressor which consists of multiple pairs of compact adsorption generators. The generators operate out of phase in order to produce continuous cooling while heat and mass recovery takes place to improve the efficiency. The sorption generator is of plate heat exchanger type developed by Critoph and Metcalf [7] and uses the activated carbon–ammonia pair. The model is a finite difference model created in Matlab® that has been published in literature [8,9]. However, it is worthwhile mentioning that key governing equations for the model are related to the energy balance on the wall, the thermal fluid and the generator bed:

$$M_w c_{pw}\frac{\partial T_w}{\partial t} = (UA)_{fw} T_{LM\ fw} - (UA)_{wc}(T_w - T_c) \quad \text{for the wall} \tag{1}$$

$$M_f c_{pf}\frac{\partial T_f}{\partial t} - mc_{pf}\left(T_{f\,in} - T_{f\,out}\right) = -(UA)_{fw}\Delta T_{LM\ fw} \quad \text{for the thermal fluid} \tag{2}$$

$$M_c\left(c_{pc} + x c_{pa}\right)\frac{\partial T_c}{\partial t} - M_c H\frac{\partial x}{\partial t} = (UA)_{wc}(T_w - T_c) \quad \text{for the generator bed} \tag{3}$$

Strictly c_p should be c_v, however the distinction is unimportant for a liquid or solid for which $c_p \approx c_v$ ($c_p \sim 4734\ J\ kg^{-1}\ K^{-1}$). H is the heat of sorption and is given by the following expression:

$$H = R\alpha\frac{T_c}{T_{sat}} \tag{4}$$

where: R is the gas constant at the bed pressure P and temperature T_c ($R \sim 488\ J\ kg^{-1}\ K^{-1}$); α is the slope of the saturated ammonia line on the Clapeyron diagram ($\alpha = 2823.4$ K); T_c is the carbon