Applied Thermal Engineering 110 (2017) 949–961

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

# Performance of a finned activated carbon cloth-ethanol adsorption chiller

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## HIGHLIGHTS

- An AHP with finned adsorbent heat exchanger was constructed.
- A new method of scaling up calorimetric LTJ tests on small samples to predict SCP and COP is presented.
- Dynamic losses through vessel walls are important.
- The predicted SCP is 1.1–2.0 time measured value versus a ratio of 2–6 elsewhere.
- Air mole fractions <0.1% might account for disparities in SCP and COP between prediction and measurement.

## ARTICLE INFO

*Article history:*
Received 20 May 2016
Revised 22 July 2016
Accepted 18 August 2016
Available online 20 August 2016

*Keywords:*
Adsorption heat pump
Activated carbon cloth
Finned-adsorbent
Scale-up
LTJ

## ABSTRACT

Measurements of cooling power and heat demand are presented for an adsorption heat pump (AHP) that integrated a finned adsorbent heat exchanger and a solar collector. For this study the adsorbent heat exchanger was heated/cooled with fluid at near constant temperature. Results from a bench scale, large temperature jump (LTJ) test were scaled to predict the outcome of a larger experiment (adjusting for heat losses and additional heat capacities). The AHP's measured coefficients of performance were COP ∈ [0.119, 0.236] versus COP ∈ [0.233, 0.337] expected. The factor of discrepancy in specific cooling power (predicted cooling power divided by measured cooling power) is 1.1–2.0 versus a range of 2–6 suggested elsewhere. Although the scale-up procedure accounted for additional heat capacities, unwanted air ingress (even for mole fractions < 0.1%) might have substantially reduced adsorption/evaporation rates.


## 1. Introduction

This paper concerns assessments of the thermal performance of a prototype adsorption heat pump (AHP) through direct measurement and through scale-up of small bench scale tests employing the "large temperature jump" (LTJ) [1]. Our objectives were as follows. (1) Originally funding was granted to build a chiller driven by concentrated sunlight. Owing to constraints associated largely with laboratory space and generator orientation the work does not cover a chiller designed for such operation, but a steam heated unit. (2) The operation of the chiller was to be predicted by scaling up data from bench-scale measurements, in order to accelerate machine development in future. The use of (constant) steam heating, rather than (variable) solar heating made the scale up procedure more reliable (prior to future work dealing with a variable underlying radiation intensity further complicated by the unrepeatable nature of cloud cover on sunlight).

A few companies have marketed AHPs or have near market machines (e.g., MyCon in Japan/Singapore, Valliant and Viessmann, both based in Germany). These units exploit sources of waste heat at temperatures as low as 50 °C, with coefficient of performance of roughly 0.4 and cooling power as low as 3 kW. These are used mostly for air conditioning where relatively high evaporator temperatures >10 °C are acceptable. Per unit mass of adsorbent, cooling power is restricted by the low thermal conductivity of adsorbents. To improve metal-to-sorbent heat transfer, and shorten conduction paths lengths, one can adhere, press or coat adsorbent to fins [2,3]. Informative research reviews are available in [4–6].

LTJ offers a simpler, more representative approach than a mechanistic model of the adsorption heat exchanger. Such models deal with complex, coupled physical processes and require a large dataset [7] including intra-particle heat and mass transfer, particle-to-particle thermal resistance, particle-to-heat-exchanger thermal resistance and bed permeability. LTJ simply replicates the boundary conditions of the AHP, imposing these on a representative sample of adsorbent and then measuring the rate of refrigerant uptake

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http://dx.doi.org/10.1016/j.applthermaleng.2016.08.102
1359-4311/

**Nomenclature**

| | |
|---|---|
| A | area, $m^2$ |
| $b_{plug}$ | thickness of (brass) plug, m |
| c | specific heat capacity, $J\,kg^{-1}\,K^{-1}$ |
| COP | coefficient of performance |
| $d_{HX}$ | outer diameter of heat exchanger, m |
| $e_{RMS}$ | root mean square discrepancy between measured and predicted temperature, K |
| F | geometric factor |
| h | specific enthalpy, $J\,kg^{-1}$ |
| m | mass, kg |
| Q | heat transfer, J |
| r | radial co-ordinate, m |
| SCP | specific cooling power, $W\,kg^{-1}$ |
| T | temperature, K |
| U | overall heat transfer coefficient, $W\,m^{-2}\,K^{-1}$ |
| X | adsorption loading |
| $X^*$ | adsorption capacity |
| V | volume, $m^3$ |
| y | mole fraction |
| z | vertical distance below start of condensate film, m |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | heat transfer coefficient, $W\,m^{-2}\,K^{-1}$ |
| $\lambda$ | thermal conductivity, $W\,m^{-1}\,K^{-1}$ |
| $\rho$ | density, $kg\,m^{-3}$ |
| $\sigma$ | Stefan-Boltzmann constant, $W\,m^{-2}\,K^{-4}$ |
| $\tau$ | time constant, s |
| $\tau_{adiab}$ | time constant for evaporator load (corrected), s |
| $\tau_{1/2}$ | half cycle duration, s |

*Subscripts*

| | |
|---|---|
| a | all heat passing through fin base |
| a | refers to specific enthalpy of adsorbate |
| coil | heat transfer through evaporator or condenser |
| conv | convective component of heat transfer |
| fg | refers to heat of vaporisation |
| f | refers to saturated liquid |
| g | refers to saturated vapour |
| in | refers to heat input |
| rad | radiative component of heat transfer |
| s | refers to saturation pressure |
| v | refers to superheated vapour |
| x | adsorbent |

*Superscripts*

| | |
|---|---|
| 1,2 | serial number used to indicated different estimate attempts |
| (a−l) | allows for heat losses |
| (a−lx) | allows for heat losses and sensible heat changes |

(=adsorbate intake). In such a test, the sample is held at near constant pressure and the set point temperature undergoes a step change. The uptake is determined from a small (~2 mbar) pressure change in a reservoir of adsorbate or (more recently) by direct weighing.

Early measurements concerned "constant volume variable pressure" or V-LTJ [1]. The size of the vapour holding tank constrained the vapour uptake and therefore constrained sample sizes. Nonetheless Aristov et al. [8] report sample sizes as big as 314 mg with mono- or multiple layers of Fuji silica RD grains. Aristov et al. [8] suggested that the SCP for real adsorption coolers was 2-to-6 time lower than would have been expected from LTJ. This was attributed to (1) the temperature of metal supports changed very quickly during LTJ but not so in real AHPs and (2) the tests in [8] allowed for large mass transfer surfaces and thin adsorbent layers. More recently realistic sections of AdHex have been weighted directly, hence "G-LTJ" [9,10]. Samples of up to 600 g were tackled with claimed accuracy of 0.1 g. A complementary calorimetric approach [11] tackled samples in the range of ~60 g and the data from this approach is used here in the analysis of an AHP.

Our work was motivated by the construction of a solar chiller, designed so that the adsorption heat exchanger could be illuminated directly by concentrated irradiation. As a first step in understanding the adsorption heat exchanger, we have worked with "normal" boundary conditions - both heating and cooling fluid were supplied at a nearly constant temperature. The selected pair was activated carbon cloth (ACC)-ethanol. The paper presents the construction of the AHP and summarises features of an earlier LTJ test on comparatively small samples of finned-surface-plus-adsorbent. Heat balances indicated the importance of dynamic heat losses and steady heat losses through the generator containment. In addition to the direct thermal measurements procedures to scale up cooling power from the smaller LTJ tests are presented. Trend analysis and likely errors are discussed in conjunction with results; a separate section discusses machine performance and future improvements.

## 2. Methods and material properties

The section describes the construction of the chiller (Fig. 1), operating procedures, and material properties. Each experiment was broadly in two half-cycles. Firstly, steam heating of the generator, caused refrigerant to desorb, forcing it into the condenser coil (part 11). Secondly, water cooling of the generator caused refrigerant vapour to adsorb, forcing boiling in the evaporator coil (part 14). The essential features of smaller-scale measurements (under LTJ) are presented [11].

### 2.1. Construction of chiller

In essence the chiller comprised the generator, the evaporator, the condenser, and instrumentation.

To form the generator, the fin assembly was housed in glass tube, outer diameter 121-mm, with domed end (Fig. 1, item 2). Turnbull and Scott (Engineers) Ltd., Hawick, Scotland pressed square fins (item 17) hydraulically onto a copper tube-in-tube heat exchanger, with eleven ACC layers filling each fin-to-fin gap (items 1, 3, 4, 15, 16, 17). To render one side of the generator receptive to solar radiation the last 5 mm of fin was bent to form a lip and coated with solar selective material. (The formation of 100 such lips was time consuming, requiring 150 person hours.) A brass plug sealed the open end of the glass tube and accommodated the heat exchanger (Fig. 1 items 1, 2, 4 and 4a). The glass tube was fabricated by glass blowers and the plug ground to fit snuggly therein (items 2, 4a). Under near vacuum, the assembly was leak tight to within 1 mbar of air ingress per day. However, the brass plug (mass 3.19 kg) promoted unwanted dynamic losses. The plug requires redesign in future – for example machineable ceramics [12] would reduce mass and thermal conductivity. (Their thermal conductivity is several hundred times lower than that of brass.)

The evaporator coil (item 14) was formed from a 2-m long, 22 mm-bore copper tube coiled six times, and similarly the