



EXPERIENCE IN MODELLING OF A SINGLE-STAGE SILICA GEL-WATER ADSORPTION CHILLER

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Abstract

Heat utilization for cooling capacity production is nowadays a desirable challenge in several industrial applications. There are lots of industrial processes with low parameters of heat generated as by-product, which utilization is very important to improve their total energy efficiency.

Waste heat driven chillers seem to be great competitors for mechanical chillers. Among them special attention should be paid to adsorption chillers, since they can be powered with low – temperature heat sources.

The paper presents a model of a single-stage adsorption chiller with silica gel as adsorbent and water, acting as a refrigerant. The performed model allows to predict the behaviour of the adsorption chiller, among others the main energy efficiency factors, such as coefficient of performance (COP) and cooling capacity (CC) for different working conditions.

Nomenclature

A	– area of heat exchanger, m ²
CC	– cooling capacity,
COP	– coefficient of performance, dimensionless
c _p	– specific heat capacity at constant pressure, J kg ⁻¹ K ⁻¹
dM _{chilled} /dt	– mass flow rate of chilled water, kg s ⁻¹
dM _{cooling} /dt	– mass flow rate of cooling water, kg s ⁻¹
dM _{heating} /dt	– flow of hot water, kg s ⁻¹
D _S	– diffusive surface, m ² s ⁻¹
D _{S0}	– kinetic diffusion constant for the water-silica gel system, m ² s ⁻¹

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E_a	– activation energy, kJ kmol^{-1}
h	– specific enthalpy, latent heat of evaporation, kJ kg^{-1}
HP	– heating power transferred with the hot water flow
K	– mass transfer coefficient, m s^{-1}
K_0	– Tóth constant, 1 Pa^{-1}
M	– mass, kg
n	– structural heterogeneity parameter
P	– pressure, Pa
q	– amount of vapor adsorbed in the bed, kg kg^{-1}
q^*	– maximum bed saturation under given temperature and pressure conditions, kg kg^{-1}
q_0	– monolayer maximum sorption capacity of silica gel, kg kg^{-1}
Q_{ST}	– isothermic heat of adsorption, kJ kg^{-1}
R	– universal gas constant or individual or for water vapor, $\text{kJ kmol}^{-1} \text{ K}^{-1}$
R_p	– average radius of silica-gel particles, m
t	– dimensionless Tóth constant
T	– temperature, K
T_0	– temperature at starting point for model calculation, K
t_{cycle}	– cycle time, s
T_k	– temperature of water inside of heat exchanger component, K
U_{cond}	– heat transfer coefficient during condensation, $\text{W m}^{-2} \text{ K}^{-1}$
U_{cooling}	– heat transfer coefficient during bed cooling process, $\text{W m}^{-2} \text{ K}^{-1}$
U_{evap}	– heat transfer coefficient during evaporation, $\text{W m}^{-2} \text{ K}^{-1}$
U_{heating}	– heat transfer coefficient during bed heating process, $\text{W m}^{-2} \text{ K}^{-1}$
v	– specific volume of the vapour phase under given conditions, $\text{m}^3 \text{ kg}^{-1}$
V	– volume of water flowing via heat exchanger, m^3

Greek symbols

γ	– flag coefficient
δ	– flag coefficient
θ	– flag coefficient
ρ	– density of water at a given temperature, kg m^{-3}
χ	– flag coefficient

Subscripts and superscripts

ads	– adsorber
bed	– bed
chw	– chilled water
cond	– condenser
cw	– cooling water
evap	– evaporator
f	– liquid state
g	– gas, vapour state
Hex	– heat exchanger
hw	– hot water
i	– inlet
ini	– initial state
max	– maximum
o	– outlet
ref	– refrigerant
s	– saturated state
sg	– silica gel
x	– adsorption
y	– desorption
z	– switching time

Introduction

In their vast majority, high valued energy sources, e.g. electricity and fossil fuel driven appliances are used for the heating and cooling purposes. However, there are also a wide range of low grade sources of thermal energy, e.g. sewage water, underground resources, solar heat, waste heat, which also can be used to deliver useful heating or cooling (AMMAR et al. 2012, STANEK, GAZDA 2014). Adsorption cycles have a distinct advantage over other systems in the ability to use low grade heat, especially heat of low near ambient temperature (AMMAR et al. 2012, STANEK, GAZDA 2014).

Application of adsorption processes for cooling cycle was first described in 1920s (LOH 2010). Further research in this area provided lots of experimental data for different pairs of adsorbent-adsorbate (ARISTOV et al. 2006, DEMIR et al. 2008, WANG et al. 2014). A novel design with passive evaporation was discussed by WANG et al. (2011) and most common pairs of adsorbent-adsorbate are reported by DEMIR et al. (2008). The adsorption of water on silica gel surface was examined inter alia by ARISTOV et al. (2006) and CHUA et al. (2002). Results of experiment for silica gel impregnated with CaCO_3 can be found in AKISAWA and MIYAZAKI (2010). Adsorption kinetics for active carbon-R134a and active carbon – R507A was investigated in HABIB et al. (2010).

Besides zeolite/water the silica gel-water is considered as one of the most promising working pairs best suited for the solutions using low grade sources of thermal energy (MYAT et al. 2013, SZYC, NOWAK 2014a,b. SIMILAR conclusion was reported in CHOROWSKI and PYRKA (2015) and QIAN et al. (2013).

It is due to relatively easy regeneration process of the adsorbent. These two pairs adsorbent-adsorbate mostly have found their implementation in commercialised chillers offered by suppliers on market.

There are some applications with adsorption chillers in industrial scale in Europe. Adsorption chillers can use solar energy (HABIB et al. 2013, SEKRET, TURSKI 2012 and other waste heat sources. A review of the existing installations can be found inter alia by Wu et WANG (2006).

One of the well-known method to improve the heat and mass transfer coefficient between the bed of sorbent and the immersed heating surface is the idea to apply of a fluidized bed instead of the fixed-bed of the sorbent, commonly used in the heat recovery technologies (KRZYWANSKI et al. 2012, WIN et al. 1995, YANG 2003). WANG et al. (2012) proposed a fluidized-bed adsorber/desorber for the adsorption refrigeration system. The working pair of activated carbon and R134a in a fluidized bed was applied in the paper. The authors pointed out that the fluidized adsorbent can enhance the heat and mass transfer, leading to the increase the specific cooling power. Due to the enhanced heat and mass transfer the heating transfer area can be greatly

reduced which is beneficial, extending the applications of the adsorption refrigerators (WANG et al. 2012).

CHEN et al. (2015) proposed to use a periodic operating silica gel fluidized bed system to adsorb/desorb moisture in air-conditioning systems. The authors reported that the fluidized bed system allows to increase the adsorption/desorption processes compared with the packed bed. HORIBE et al. (2013) applied fluidized bed to study the sorption and desorption characteristics of a new organic sorbent material HU720PR. The connected fluidized beds for adsorption and desorption processes are used in the unit. Two funnels inside the beds allow particles to circulate between adsorption and desorption beds to ensure continuous operating of a dehumidification air conditioning system (CHEN et al. 2015). Similar process was also carried out by HAMED (2005). The author used an inclined-fluidized bed in the adsorption and desorption operations.

The main feature of adsorption chillers is its lower sensitivity to hot water inlet temperature fluctuations. It makes adsorption chillers more suitable for the use of low grade sources of thermal energy in comparison to their main competitors, i.e. absorption chillers. The improvement of total efficiency of adsorption process is however still challenging task.

Modelling is a wide used method of data handling which can be helpful to carry out such undertaking. Several models and experimental works on adsorption cooling cycles are described in the literature. Modeling and performance analysis of an adsorption desalination cycle was described by THU (2010), whereas the distributed-parameter approach to capture both the transient and steady state behaviors of adsorption chiller was presented by CHUA et al. (2004). A dynamic model for the prediction of the optimal cyclic operation mode of the two-bed adsorption chiller was shown by GRÄBER et al. (2011).

The analysis of an advanced three-stage adsorption chiller as well as the entropy generation analysis and performance analysis of a pressurized adsorption chiller were presented in LOH et al. (2010), SAHA et al. (1995), THU et al. (2013).

A multi-bed regenerative adsorption chiller design and a novel silica gel-water adsorption chiller model with three vacuum chambers were proposed by CHUA et al. (2001) and WANG et al. (2005), respectively.

The experimental analysis of a compact adsorption chiller, the evaluation of the effect of adsorbent isobar shape on the dynamics of the chillers and the water adsorption dynamics for adsorption chillers with adsorbent beds of loose grains was discussed in ARISTOV et al. (2012), LU et al. (2013) and OKUNEV and ARISTOV (2014). The dynamic behaviour of a single effect two bed adsorption chiller employing adsorbent beds with various layers of loose grain configu-

ations and silica gel particle sizes and the effect of design parameters on the switching frequency was discussed by CHAKRABORTY et al. (2014) and ALAM et al. (2003), respectively.

Analytically and experimentally investigated silica gel-water system was described by BOELMAN et al. (1995) and SAHA et al. (1995). Three-bed and a dual mode, three stage non-regenerative and six bed regenerative silica gel – water chillers were investigated in SAHA (2003) and SAHA et al. (2003).

The work presents a model of a single-stage adsorption chiller with silica gel acting as a solid porous sorbent and water working as refrigerant. The set of thermodynamic equations used in the model of chiller operation is described in the paper. The validation of the model was successfully performed against the experimental results from the existing adsorption chiller at National University of Singapore. Such approach is undertaken since the comparison between desired and calculated data is regarded as the most difficult type of model's validation procedure (KRZYWANSKI et al. 2015a, b, c).

An object of investigations and working principle

Typical single stage-adsorption chiller consists of: an evaporator, a condenser and two fixed-beds, which are packed in separate reactors (CHUA et al. 2001, LOH 2010, THU 2010). Silica gel-water is considered during the study as a working pair of adsorbent-adsorbate. Solid porous sorbent has relatively high sorption capability with refrigerant. The heat transfer between hot and cold water and the sorbent occurs via the coils immersed inside the beds. The scheme of the considered system is given in Figure 1.

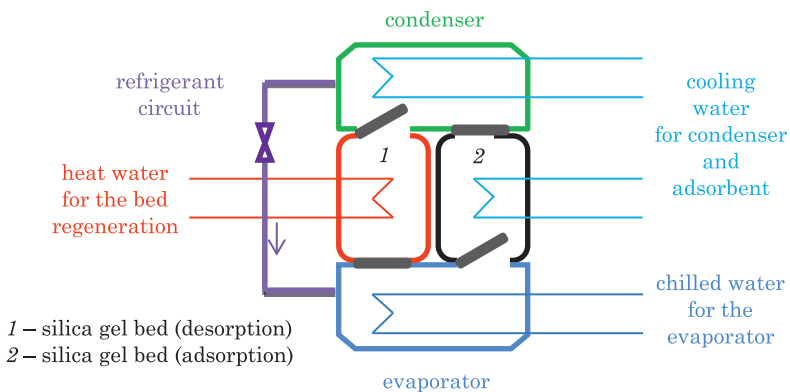


Fig. 1. Schematic diagram of an adsorption chiller

The experimental procedure for a conventional adsorption chiller is as follows. The evaporated refrigerant (water) passes from the evaporator to the fixed adsorption bed 2, which is cooled by the cold water flowing inside tubes into the bed. The adsorption processes occur until the pressure inside the adsorption bed reaches about the saturated pressure of water, corresponding to the cooling water temperature in the adsorption bed. To regenerate the bed, hot water is put into the coil and the vapour is released from the bed.

The desorbed vapour enters into the condenser (SZYC, NOWAK 2014a, b).

To improve the efficiency more than one bed are applied. Adsorption chillers usually consists of two or four beds (CHOROWSKI, PYRKA 2015, SZYC, NOWAK 2014a, b). When some of the beds work as adsorbers, the desorption process occurs in other beds at the same time, and on the contrary.

To increase the efficiency of the unit, the beds must be properly prepared for the upcoming phase. Therefore before the two main stages (i.e. adsorption and desorption) starts the beds previously must be pre-heated (before desorption) or pre-cooled (before adsorption), respectively.

During these short periods, usually called switching time periods, all valves connecting the beds with the evaporator and the condenser are closed.

By this way, four phases, i.e.: pre-cooling, adsorption, pre-heating and desorption, occur one by one during the full adsorption cooling cycle in each of the bed of an adsorption chiller.

Mathematical modelling of adsorption chiller

Fundamental assumptions

The model is based on Tóth adsorption isotherm, equation of the kinetics of the process and energy balance equations for the elements: evaporator, condenser and silica gel bed (CHUA et al. 2004, SZYC, NOWAK 2014a). The RD silica gel type is used as an adsorbent. To simplify the model, the following assumptions were taken into account:

- the values for pressure and temperature in isotherm equations during adsorption are constant, i.e. the reactor pressure is equal to the pressure in the evaporator and the reactor temperature is equal to the temperature of the cooling water bed,
- isosteric heat of adsorption is constant,
- water vapour adsorbed on the silica gel is in dry saturated conditions ($x = 1$),
- the adsorption of water vapour on silica gel is uniform,

- water vapour is adsorbed on the dry silica gel in each cycle, so the maximum amount of refrigerant is uptaken by bed under the considered pressure and temperature conditions,
- the temperature profile across the bed is uniform,
- the temperature in the different parts of the device is uniform,
- heat transfer coefficients for heat exchangers have constant values,
- the system is thermally insulated from the environment (the losses have been omitted),
- during the adsorption phase the reactor pressure is constant and has the same value as in the evaporator,
- during the desorption phase the pressure in the reactor is constant and has the same value as the condenser.

The system of balance differential equations was solved by the Runge-Kutty method of the 4th order using C++ language.

Adsorption isotherm and refrigerant uptake by bed

For estimation of the amount of vapor adsorbed on the bed surface the Tóth isotherm is used (CHUA et al. 2004):

$$q^* = \frac{K_0 \exp\left[\frac{Q_{ST}}{RT_s}\right] P_s}{\left\{1 + \left[\frac{K_0}{q_0} \exp\left(\frac{Q_{ST}}{RT_s}\right) P_s\right]^t\right\}^{\frac{1}{t}}} \quad (1)$$

The changes of bed saturation in time describes linear driving force equation (LOH et al. 2010):

$$\frac{dq(t)}{dt} = K(q^* - q) \quad (2)$$

The K coefficient is defined, as follows:

$$K = \frac{15D_s}{R_p^2} \quad (3)$$

where:

$$D_s = D_{s0} \exp\left[\frac{-E_a}{RT_s}\right] \quad (4)$$

Substituting equation (4) to (3), and then into equation (2), we have the final equation (5):

$$\frac{dq(t)}{dt} = \frac{15D_{s0} \exp\left[\frac{-E_a}{RT_s}\right] (q^* - q)}{R_p^2} \quad (5)$$

The isosteric heat adsorption can be expressed by the Dubinin-Astakov equation (LOH et al. 2010):

$$Q_{ST} = h + E_a \ln\left(\frac{q_0}{q}\right)^{1/n} + T_{v_g} \frac{dP}{dT} (P, T) \quad (6)$$

Energy balance during adsorption phase

For bed which is connected to the evaporator (index x) the energy balance can be written, as follows:

$$\left(M_{sg} cp_{sg} + M_{Hex} cp_{Hex} + M_{sg} q_{bed,x} \left[cp_g(T_{bed,x}) + Q_{ST} \left\{ \frac{1}{T_{bed,x}} - \frac{1}{v_g(T_{bed,x})} \left(\frac{\partial v_g}{\partial T_{bed,x}} \right) \right\} \right] \right) \frac{dT_{bed,x}}{dt} = M_{sg} \frac{dq_{bed,x}}{dt} [\delta \{ h_g(T_{evap}) - h_g(T_{bed,x}) + Q_{ST} \} + (1 - \delta) \{ h_g(T_{i,chw}) - h_g(T_{bed,x}) + Q_{ST} \}] - U_{cooling} A_{bed} (T_{bed,x} - T_k) \quad (7)$$

The equation (7) allows to determine bed temperature profile during adsorption.

The energy balance equation for the cooling water, which flows inside tubes immersed into the sorbent bed during the adsorption stage, is as follows:

$$\rho_f cp_f(T_k) V_{bed} \frac{dT_k}{dt} = \frac{dM_{cooling}}{dt} [h_f(T_{i,cw}) - h_f(T_k)] + U_{cooling} A_{bed} (T_{bed,x} - T_k) \quad (8)$$

where:

$$T_0(t) = T_{i,cw}, \quad T_k(0) = T_{i,cw}, \quad T_{bed,x}(0) = T_{i,cw}, \quad T_{evap}(0) = T_{i,chw}, \quad q_{bed,x}(0) = q_{bed,x}^{ini}, \\ P_{evap}(0) = P_s(T_{i,chw}).$$

This equation allows to estimate the temperature profile for cooling water which is responsible for removing heat of adsorption.

Energy balance during desorption phase

The energy balance for the bed which is connected with the condenser (index y) can be expressed by the equation:

$$\left(M_{sg}c_{p_{sg}} + M_{Hex}c_{p_{Hex}} + M_{sg}q_{bed,y} \left[c_{p_g}(T_{bed,y}) + Q_{ST} \left\{ \frac{1}{T_{bed,y}} - \frac{1}{v_g(T_{bed,y})} \left(\frac{\partial v_g}{\partial T_{bed,y}} \right) \right\} \right] \right) \frac{dT_{bed,y}}{dt} = M_{sg} \frac{dq_{bed,y}}{dt} [\theta \{h_g(T_{i,hw}) - h_g(T_{bed,y}) + Q_{ST}\} + (1 - \theta) \{h_g(T_{cond}) - h_g(T_{bed,y}) + Q_{ST}\}] - U_{heating,y} A_{bed,y} (T_{bed,y} - T_k) \tag{9}$$

Equation (9) allows to determine bed temperature changes during desorption.

The energy balance for hot water supplied for the desorption process can be written, as follows:

$$\rho_f c_{p_f}(T_k) V_{bed} \frac{dT_k}{dt} = \frac{dM_{heating}}{dt} [h_f(T_{i,hw}) - h_f(T_k)] + U_{heating,y} A_{bed,y} (T_{bed,y} - T_k) \tag{10}$$

where:

$$T_0(t) = T_{i,hw}, T_{bed,y}(0) = T_{i,hw}, q_{bed,y}(0) = q_{inibed,y}, P_{cond}(0) = P_s(T_{i,cw}), T_k(0) = T_{i,hw}.$$

Energy balance during switching time

In the period when a bed is prepared for desorption, reactor is not connected neither with the evaporator nor condenser (index z). The energy balance for the bed can be described by the equation:

$$\left(M_{sg}c_{p_{sg}} + M_{Hex}c_{p_{Hex}} + M_{sg}q_{bed,z} \left[c_{p_g}(T_{bed,z}) + Q_{ST} \left\{ \frac{1}{T_{bed,z}} - \frac{1}{v_g(T_{bed,z})} \left(\frac{\partial v_g}{\partial T_{bed,z}} \right) \right\} \right] \right) \frac{dT_{bed,z}}{dt} = - U_{heating} A_{bed} (T_{bed,z} - T_k) \tag{11}$$

Similar energy balance equations can be written for the adsorption stage (equation 12):

$$\left(M_{sg}c_{p_{sg}} + M_{Hex}c_{p_{Hex}} + M_{sg}q_{bed,z} \left[c_{p_g}(T_{bed,z}) + Q_{ST} \left\{ \frac{1}{T_{bed,z}} - \frac{1}{v_g(T_{bed,z})} \left(\frac{\partial v_g}{\partial T_{bed,z}} \right) \right\} \right] \right) \frac{dT_{bed,z}}{dt} = - U_{cooling} A_{bed} (T_{bed,z} - T_k) \quad (12)$$

For hot water flowing inside the tubes immersed in the adsorber bed we have:

$$\rho_f c_{p_f}(T_k) V_{bed} \frac{dT_k}{dt} = \frac{dM_{heating}}{dt} [h_f(T_{i,hw}) - h_f(T_k)] + U_{heating} A_{bed} (T_{bed,z} - T_k) \quad (13)$$

and the same for cooling water:

$$\rho_f c_{p_f}(T_k) V_{bed} \frac{dT_k}{dt} = \frac{dM_{cooling}}{dt} [h_f(T_{i,hw}) - h_f(T_k)] + U_{cooling} A_{bed} (T_{bed,z} - T_k) \quad (14)$$

where:

$$T_0(t) = T_{i,hw} \text{ or } T_{i,cw}, \quad T_{bed,z}(0) = T_{i,hw} \text{ or } T_{i,cw}, \quad q_{bed,z}(0) = q_{bed,z}^{ini}, \quad P_{cond}(0) = P_s(T_{i,cw}), \quad T_k(0) = T_{i,hw}.$$

Energy balance for the evaporator

The energy balance for the evaporator is described as follows:

$$\begin{aligned} & [c_{p_f}(T_{evap})M_{ref,evap} + c_{p_{evap}}M_{evap}] \frac{dT_{evap}}{dt} + h_f(T_{evap}) \frac{dM_{ref,evap}}{dt} = \\ & - \theta(1 - \gamma)h_f(T_{cond})M_{sg} \frac{dq_{bed,x}}{dt} - [\delta h_g(T_{evap}) + (1 - \delta)h_g(T_{bed,x})]M_{sg} \frac{dq_{bed,x}}{dt} - \\ & U_{evap} A_{evap} (T_{evap} - T_k) \end{aligned} \quad (15)$$

where amount of refrigerant circulated in cooling loop is calculated from:

$$\frac{dM_{ref,evap}}{dt} = - M_{sg} \sum_{z=1}^2 \chi \frac{dq_{bed}}{dt} \quad (16)$$

During the period, when a bed is connected with the evaporator (adsorption) $\chi = 1$, whilst $\chi = \theta(1 - \gamma)$ when bed is connected with condenser, during desorption, and:

$$\frac{dM_{\text{ref, evap}}}{dt} = -M_{\text{sg}} \left((1 - \delta)(1 - \gamma) \frac{dq_{\text{bed}}}{dt} \right) - \frac{dq_{\text{bed}}}{dt} \quad (17)$$

For chilled water the energy balance equation can be written as follows:

$$\rho_f c_{p_f}(T_k) V_{\text{evap}} = \frac{dM_{\text{chilled}}}{dt} [h_f(T_{i, \text{chw}}) - h_f(T_k)] + U_{\text{evap}} A_{\text{evap}} (T_{\text{evap}} - T_k) \quad (18)$$

where:

$$M_{\text{ref, evap}}(0) = M_{\text{ref, evap}}^{\text{ini}}, T_0(t) = T_{i, \text{chw}}, T_{\text{evap}}(0) = T_{i, \text{chw}}, T_{\text{bed, x}}(0) = T_{i, \text{cw}}, T_{\text{cond}}(0) = T_{i, \text{cw}}, T_k(0) = T_{i, \text{chw}}$$

Energy balance for the condenser

The energy balance for the condenser can be described by the following equation:

$$[c_{p_f}(T_{\text{cond}})M_{\text{ref, cond}} + c_{p_{\text{cond}}}M_{\text{cond}}] \frac{dT_{\text{cond}}}{dt} - \gamma h_f(T_{\text{cond}})M_{\text{sg}} \frac{dq_{\text{bed, y}}}{dt} = -[\theta h_g(T_{\text{bed, y}}) + (1 - \theta)h_g(T_{\text{cond}}) - \theta(1 - \gamma)h_f(T_{\text{cond}})]M_{\text{sg}} \frac{dq_{\text{bed, y}}}{dt} - U_{\text{cond}}A_{\text{cond}}(T_{\text{evap}} - T_k) \quad (19)$$

When the condenser collects sufficient amount of condensate $M_{\text{ref, cond}}^{\text{max}}$, refrigerant flows into the evaporator. The governing assumptions are as follows:

- when $M_{\text{ref, cond}} < M_{\text{ref, cond}}^{\text{max}}$, or $M_{\text{ref, cond}} = M_{\text{ref, cond}}^{\text{max}}$, and $\frac{dq_{\text{bed, y}}}{dt} > 0$ then $\gamma = 1$
- and
- when $M_{\text{ref, cond}} = M_{\text{ref, cond}}^{\text{max}}$ and $\frac{dq_{\text{bed, y}}}{dt} \leq 0$ then $\gamma = 0$.

The energy balance for cooling water in condenser is, as follows:

$$\rho_f c_{p_f}(T_k) V_{\text{cond}} \frac{dT_k}{dt} = \frac{dM_{\text{cond}}}{dt} [h_f(T_{i, \text{cw}}) - h_f(T_k)] + U_{\text{cond}}A_{\text{cond}}(T_{\text{cond}} - T_k) \quad (20)$$

where:

$$M_{\text{ref,cond}}(0) = M_{\text{ref,cond}}^{\text{max}}, T_0(t) = T_{i,cw}, T_{\text{cond}}(0) = T_{i,cw}, T_{\text{bed,y}}(0) = T_{i,hw}, T_k(0) = T_{i,cw}.$$

Performance parameters

The main performance parameter of a chiller is Coefficient of Performance (COP), defined as:

$$\text{COP} = \frac{\text{CC}}{\text{HP}} \quad (21)$$

where CC is cooling capacity achieved from the system and HP is heating power supplied to the system.

The CC and HP can be determine from the equations (22) and (23):

$$\text{CC} = \frac{dM_{\text{chilled}}}{dt} c_{p_{f,i,chw}}(T_{i,chw} - T_{o,chw}) \quad (22)$$

$$\text{HP} = \frac{dM_{\text{heating}}}{dt} c_{p_{f,i,hw}}(T_{i,hw} - T_{o,hw}) \quad (23)$$

Results and discussion

Validation

The above presented set of balance equations was used to create the simulation code in C++ programming language. The performed model allows to determine temperature profiles for each of the elements and further to calculate performance coefficients, like coefficient of performance (COP) and cooling capacity (CC). Operating parameters of simulated single – stage adsorption chiller are given in Table 1.

The results of simulations were compared with experimentally obtained data for two different working conditions. The experimental data which has been collected, analysed and compared with simulation results comes from the existing adsorption chiller installed at the National University of Singapore. The detailed description of the adsorption system can be found in SZYC, NOWAK 2014a,b, THU 2010). The system can also work with heat regeneration, but during experiments it operated as a single-stage device. The measured

Table 1

The operating parameters used in the study

Parameter	Temperature °C		Mass flow rate kg s ⁻¹	Other
	test 1	test 2		
Chilled water inlet (evaporator)	10	20	0.8	–
Cooling water inlet (adsorber)		29.5	0.8	–
Cooling water inlet (condenser)		29.5	2.0	–
Hot water inlet (desorber)		70–110	0.8	–
Mass of silica-gel (per bed) [kg]		–	–	36
Switching time [s]		–	–	45
Cycle time adsorption/desorption [s]		–	–	480–900

Table 2

Selected data used for simulations

Parameter	Value
q_0	0.45
Q_{ST}	2.37×10^3
K_0	2×10^{-12}
t	12
R_p	1.7×10^{-4}
D_{s0}	2.54×10^{-4}
E_a	4.2×10^4
n	1.1
$c_{p,sg}$	0.921
$M_{Hex}c_{p,Hex}$	195
A_{ads}	47.1
V_{cond}	0.1
V_{bed}	0.157
V_{evap}	0.0142
$M_{evap}c_{p,evap}$	25.4
$M_{cond}c_{p,cond}$	18.0
$M_{ref,cond}^{max}$	10

data were sampled every 3 second. Selected data used further for calculations are given in Table 2. These values together with the data from Table 1 constitute the parameters of entry conditions during simulations.

The comparison between desired and calculated by the model values of COP and CC during Tests 1 and 2 are given in Figures. 2 and 3. The calculated results are located within the range of $\pm 20\%$ compared to the desired, i.e. experimental data.

The points in the Figures 2 and 3 which were used to validate the model, correspond to different inlet temperatures of chilled water. The approach was undertaken since the comparison between desired and calculated data is regarded as the most difficult type of model's validation procedure (KRZYWANSKI et al. 2015b,c).

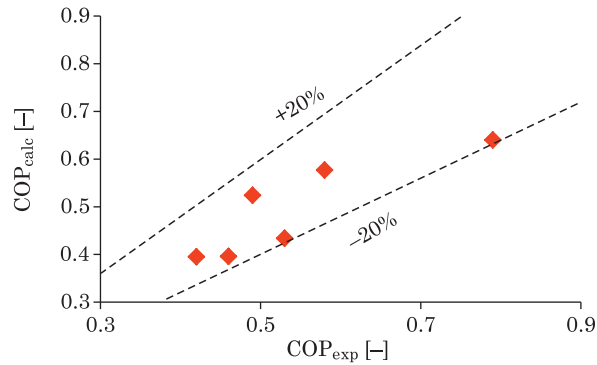


Fig. 2. Comparison between measured and calculated results for the coefficient of performance

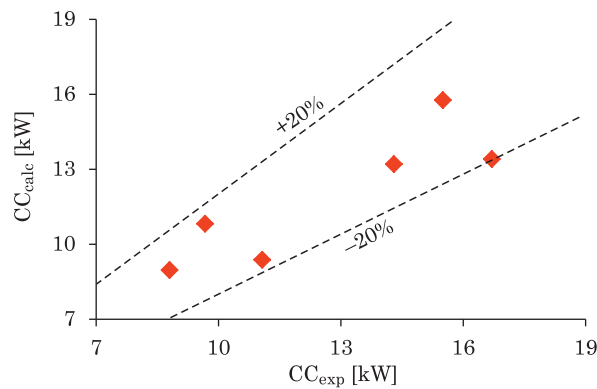


Fig. 3. Comparison between measured and calculated results for the cooling capacity

The effect of the cycle time

The dependence of the COP on the cycle time for the two considered inlet temperatures of chilled water is presented in Figure 4. As it can be seen the coefficient of performance increases with the cycle time.

Similar results were obtained by other researchers (MARLINDA et al. 2010, WANG et al. 2015). Such behaviour can be explained taking into account the differences between sorption and desorption rates and is also the result of the changes in dynamism of adsorption/desorption processes with time. Since the desorption runs faster than adsorption the heating power HP supplied to the bed may not be consumed during desorption stage when lengthening the cycle time. Therefore the instantaneous value of the heating power, constituting the denominator in equation (21), used for desorption is reduced leading to the increase in COP (MARLINDA 2010, SZYC, NOWAK 2014a).

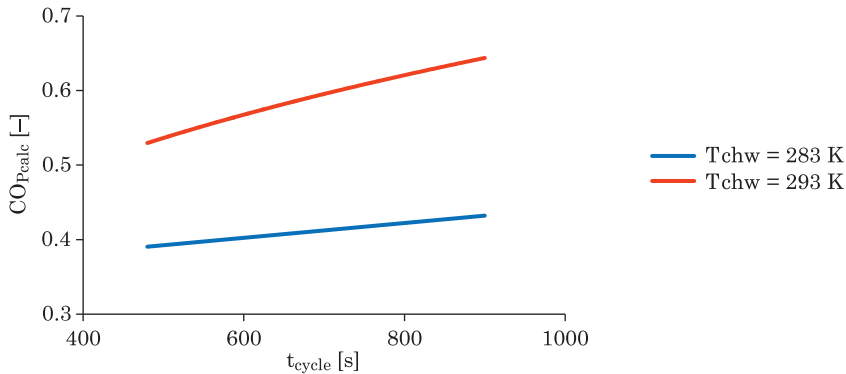


Fig. 4. The effect of the time cycle on the coefficient of performance

The effect of the cycle time on the cooling capacity is shown in Figure 5. Similar behaviour was reported by MARLINDA et al. (2010) and SAPIENZA et al. (2011). When the cycle time gets longer the water vapour-silica gel system tends to reach equilibrium and sorption processes are slowing down. This influencing the cooling capacity ratio as the coolant mass transfer between the condenser and the evaporator is also slowed down. As the result the disturbances in coolant evaporation process occur, which leads to the decrease in cooling capacity of the adsorption chiller (SZYC, NOWAK 2014a).

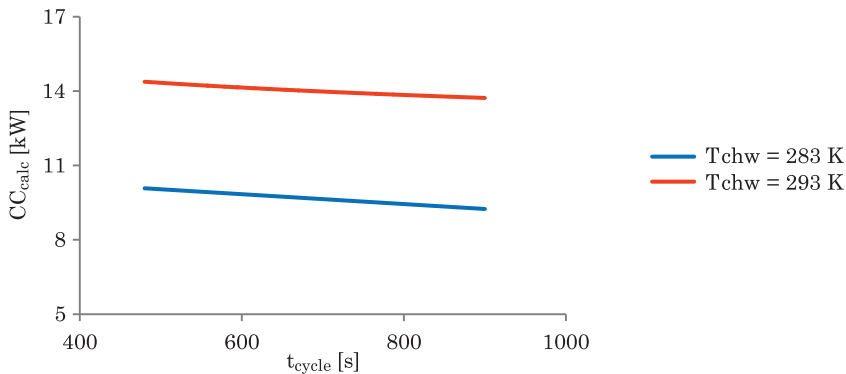


Fig. 5. The effect of the time cycle on the cooling capacity

The effect of the chilled and hot water inlet temperature

The effect of the hot water inlet temperature on the COP is given in Figure 6. The heating power (HP) supplied to the system increases with the hot water inlet temperature T_{hw} . As the HP constitutes the denominator in equation (20) the COP decreases with the increase of hot water inlet temperature.

Generally speaking, both considered parameters, i.e. COP and CC are higher for the chilled inlet water temperature of 20°C than it is for the $T_{chw} = 10^\circ\text{C}$. Similar behaviour was noted by JEON et al. (2010). Since the evaporation processes are enhanced at higher temperatures the increase in the chilled water inlet temperature cause the increase in cooling capacity CC, achieved from the system. As the result, according to equation (20), the COP tends to increase.

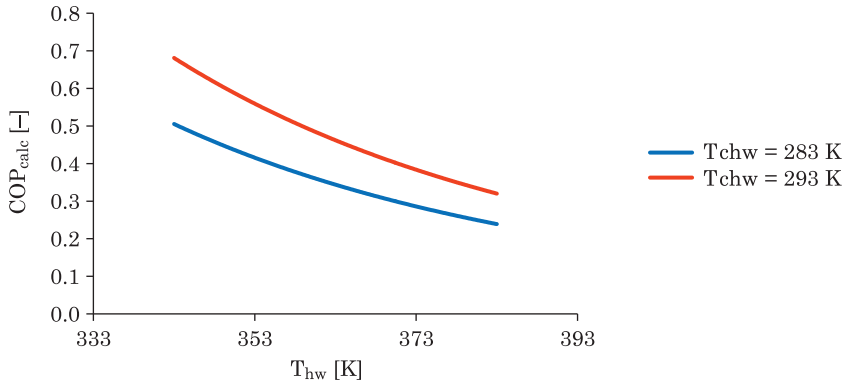


Fig. 6. The effect of the hot water inlet temperature on the coefficient of performance

The effect of the cooling water temperature and the typical profiles of beds temperatures and amounts of vapour adsorbed on the silica gel

The dependence of the cooling water temperature on the CC and COP is given in Figure 7. The main energy efficiency factors, such as coefficient of performance and cooling capacity decrease with the increase in cooling water temperature. It is the result of the decrease in the amounts of refrigerant being adsorbed for higher temperatures of cooling water flowing inside tubes into the bed. Typical profiles of the average bed temperatures as well as the amounts of vapour adsorbed on the silica gel are shown in Figure 8. As it can be seen the profiles correspond to both adsorption and desorption cycles.

Conclusions

The paper presents a model of a single-stage adsorption chiller with silica gel as adsorbent and water acting as a refrigerant. The validation of the model was successfully performed against the experimental results from the existing

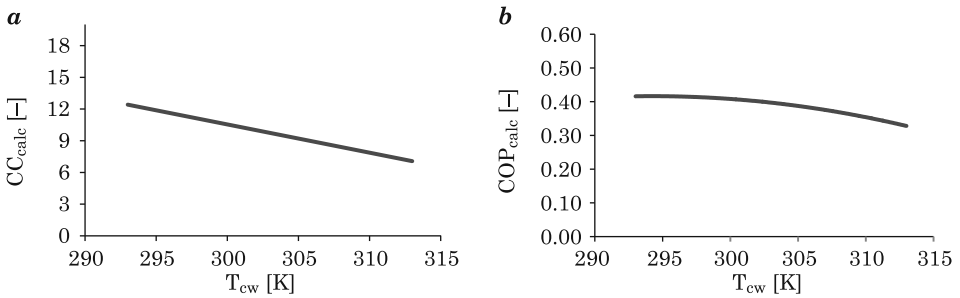


Fig. 7. The effect of the cooling water temperature on the: *a* – cooling capacity and *b* – coefficient of performance; $T_{chw} = 283 \text{ K}$

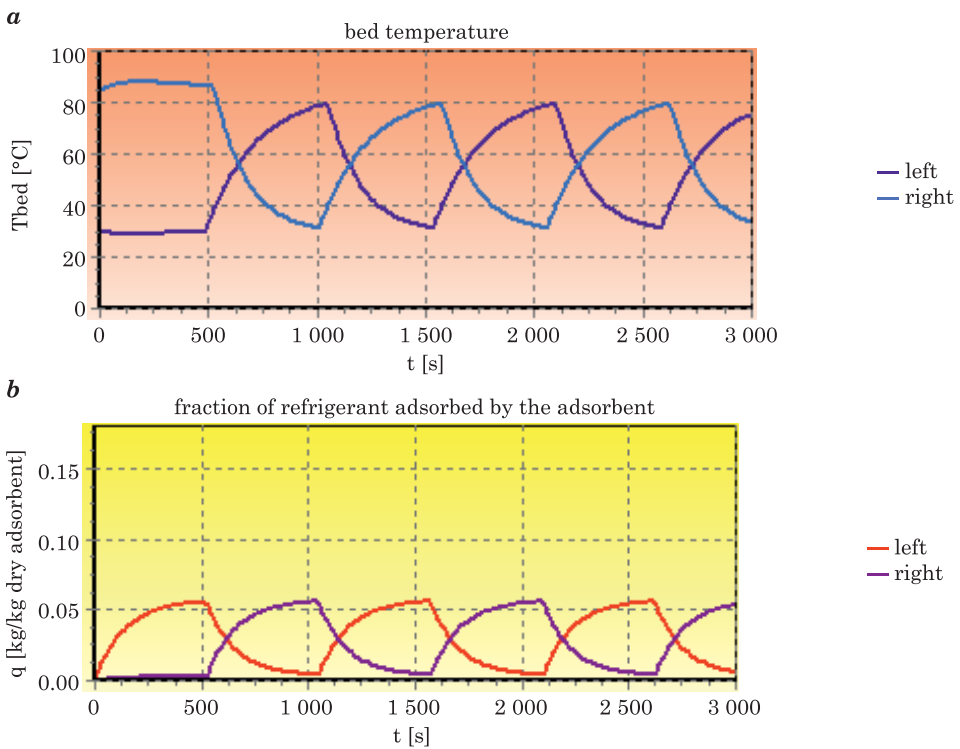


Fig. 8. The average beds temperatures (*a*) and amounts of vapour adsorbed profiles (*b*)

adsorption chiller installed at the National University of Singapore. Obtained results (calculated data) are located in the borders of $\pm 20\%$ in the relationship to experimental results.

The developed model allows to predict the behaviour of the adsorption chiller, among others the main energy efficient factors, i.e. coefficient of

performance (COP) and cooling capacity (CC) for different working conditions. The calculations carried out in the paper lead to the following conclusions:

- the coefficient of performance increases whereas cooling capacity decreases with the cycle time,
- the COP decreases with the increase of hot water inlet temperature,
- both considered parameters, i.e. COP and CC are higher for the higher chilled inlet water temperatures,
- the main energy efficiency factors (i.e. COP and CC) decrease with the increase in cooling water temperature.

The developed model can be used as an optimization tool of the examined adsorption chiller.

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