Dynamic modelling of an activated carbon–methanol adsorption refrigeration tube with considerations of interfacial convection and transient pressure process

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A R T I C L E   I N F O

Article history:
Received 26 September 2011
Received in revised form 10 February 2012
Accepted 20 February 2012
Available online 14 March 2012

Keywords:
Adsorption refrigeration tube
Interfacial convection
Transient pressure
Dynamic modelling

A B S T R A C T

In the present study, a dynamic model of the adsorption refrigeration cycle was established with the consideration of interfacial convective heat transfer within adsorbent particles. In the model, a concept and mathematical definition of a transient pressure process at the beginning of the traditionally considered isobaric adsorption process are introduced. The model was solved numerically and experimentally verified in terms of the adsorbent/adsorbate temperature development, system pressure variation, and dynamic adsorption/desorption amount. A temperature jump at the beginning of the adsorption process was experimentally identified and was successfully predicted in the numerical simulation with the introduction of a transient pressure process. Numerical results simulated with the newly introduced transient pressure process and the traditional constant pressure process were compared. The comparison shows that the introduced transient pressure process can significantly improve the accuracy of the presented model. In addition, a notable adsorbate migration phenomenon was discussed according to the abnormal temperature development in the processes of isosteric heating and cooling. The present model can be used for a valve-controlled and long cycle-time based ART and other systems with similar operating procedures.

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1. Introduction

Adsorption refrigeration has received extensive attention due to a number of advantages, such as low-grade energy requirements and low running costs. In recent years, various lab-level prototypes and a few commercialized adsorption refrigeration products have demonstrated great potential for large-scale applications [1–3].

In addition to extensive studies of complex adsorption refrigeration systems working with advanced thermal cycles (e.g., heat and mass recovery aided cycles), some researchers have studied adsorption refrigeration tubes (ARTs) that can work independently [4,5]. Through system integration, a module having multiple ARTs can achieve considerable refrigeration capacity [6–8]. An ARTs integrated system, if using the ART proposed in [4], can be a feasible solar powered and day/night cycle based adsorption refrigeration device. The feature of the physical configuration of such ARTs integrated system could overcome the vacuum maintaining problem that exists in the traditional complex adsorption refrigeration system. This is because any leak in a traditional system would destroy the whole system while that in an ARTs integrated system would only the reduce efficiency of the system because only one or a number of independent tubes are suffered.

Many studies have focused on mathematical modelling, especially for a cylindrical generator, which is the typical configuration, for both the ARTs and the basic structure of other complicated generators [5,9]. Because solid adsorption refrigeration is basically an adsorption phenomenon involving heat and mass transfer processes, the work of modelling mainly concerns adsorption performance, heat transfer in the adsorbent bed, and the coupled mass transfer processes. The present model can be used for a valve-controlled and long cycle-time based ART and other systems with similar operating procedures.
2. Physical model

Fig. 1a and b shows the cross-section of an ART and a schematic diagram of radial convective heat transfer in a generator. The generator can be heated or cooled by circulating hot water or cool water in the water jacket. A perforated tube is placed in the centre of the generator to serve as the mass transfer channel. A detailed description of the ART configuration and working principle can be found in the literature [4]. In this study, activated carbon Calgon WS-480 was selected as the adsorbent because first of all its adsorptive properties can be well characterized by the D–A equation [21]. Moreover, comparing with the tested samples Calgon 207C and 207EA [21], activated carbon WS-480 may generate higher efficiency at the low driven temperature region (85–90 °C) which is the experimental conditions in this study. The adsorbate adopted in the experiments was methanol.

Fig. 1b illustrates the convective heat transfer within the adsorbent bed, which is divided into two types, i.e., interfacial \( h_i \) and external \( h_e \) convective heat transfer. The interfacial convection represents heat transfer mainly caused by methanol vapour.
moving through the internal transport pores (i.e., macropores and mesopores [22]) of activated carbon particles, whereas the external convection denotes the heat transfer occurring at external surfaces of the activated carbon particles. However, in a mathematical model, it is difficult to precisely consider both convection mechanisms because the exact proportion of methanol vapour that transfers among the gaps of the particles and from the transport pores of the particles and that which partly travels through the transport pores and partly through the gaps are not known in practice. In order to achieve a simplified and reasonable mathematical model, the external convective heat transfer \( (h_e) \) is considered to be negligible. This simplification is justified because the convective heat transfer coefficients for the interfacial convection and the external convection are of the same order, according to the experimental data compiled by Wakao and Kaguei [23], while the effective heat convection are of the same order, according to the experimental orders of magnitude. The specific surface area of the activated carbon particles was determined using data in the literature [24].

Regarding mass transfer in the generator, the driving forces arise from two aspects, the vapour temperature difference causing pressure gradient within the bed and the external fluid-caused pressure gradient. As shown by the Clapeyron diagram for a typical adsorption refrigeration cycle in Fig. 2, during process 1–2–3, methanol vapour and solid carbon are heated from the outside (i.e., \( R_0 \)), which results in a temperature difference in the methanol vapour in the radial direction. The temperature difference results in a pressure gradient that further leads to convection. In other words, the vapour methanol moves inwards.

For the case of process 3–4–5, in addition to temperature-difference-cause pressure gradient, the external fluid-caused pressure gradient contributes to vapour movement as well. When the evaporator is at the condensing pressure, it is connected to the generator at point 4, and the generator pressure jumps (from \( P_e \) to \( P_t \)) and then takes some time to drop back to and remain at a constant \( P_o \). Before the pressure reaches the stable value \( P_g \), again, the pressure at the boundary \( Ri \) is transient, causing the vapour density at this boundary to also be transient. This transient vapour density at the inner boundary, together with the existing density field (established in process 3–4), leads to further convection.

3. Mathematical model

3.1. Modelling assumptions

When considering interfacial convective heat transfer, local thermal non-equilibrium between the solid phase and the gaseous methanol is assumed, i.e., a so-called two-temperature treatment. Methanol vapour flows through transport pores within the activated carbon particles. The effective porosity (volume fraction) of the transport pores is equal to that of the bed porosity in a loosely packed bed [25]. Other assumptions are made as follows:

1. The activated carbon and adsorbed methanol (liquid state) are in local thermal equilibrium.
2. The gaseous carbon and adsorbed methanol are assumed to be an ideal gas.
3. The thermal conductivities of the activated carbon and methanol are constant.
4. The size of the transport pores of the activated carbon particles is assumed to be uniform.
5. The adsorbent bed is packed with uniform-size and isotropic particles.
6. The thermal resistance between the housing and the heat transfer fluid is negligible [13].
7. The thermal resistance between the metal housing and the adsorbent bed is negligible [10].

3.2. Governing equations

3.2.1. Sorption rate

The linear driving force (LDF) [17] and the Dubinin–Astakhov equation (D–A) [6] are employed to represent the sorption rate; they have been used extensively to account for non-equilibrium adsorption. The equations are as follows:

\[
\frac{dx}{dt} = 15D_0R_s^2 \exp \left( -\frac{E_a}{RT_s} \right) (x_{eq} - x) \tag{1}
\]

\[
x_{eq} = x_0 \exp \left( -D \left( T_{f} \ln \frac{P_{eq}}{P} \right)^n \right) \tag{2}
\]

where \( x_{eq} \) represents the equilibrium concentration at the corresponding pressure and temperature, and \( x \) denotes the actual concentration. In Eq. (1), the coefficients 15\(D_0/\kappa_s^2\) and \(E_a/\kappa_s\) were taken from the literature [26]. In Eq. (2), \(x_0\), \(D\), and \(n\) are the coefficients of the D–A state equation, which were experimentally determined [21].

3.2.2. Energy transfer

The adsorption and desorption processes typically accompany apparent heat sinks and sources in the solid adsorbent (activated carbon). In these circumstances, the assumption of a local thermal equilibrium between the solid phase and the gaseous fluid (gaseous methanol) is not sufficient [25]. Hence, a local thermal non-equilibrium (i.e., a two-temperature treatment) was used to establish the following energy governing equations.

It is assumed that heat conduction, interfacial convection, and an internal heat source (or sink) contribute to the heat transfer within the solid phase. For the gaseous fluid, conduction and convection lead to heat transfer. Consequently, heat transfer equations for the solid phase and the gaseous fluid can be written as follows in cylindrical coordinates:

\[
(1 - \varepsilon)(\rho c_p) \frac{\partial T_s}{\partial t} - \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_s}{\partial r} \right) + A_0 h_i (T_f - T_s) + \rho h \frac{\partial x}{\partial t} \tag{3}
\]

\[
\varepsilon (\rho c_p) \frac{\partial T_f}{\partial t} + \varepsilon (\rho c_p) \frac{\partial (\rho \kappa_s)}{\partial T_f} = \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_f}{\partial r} \right) + A_0 h_i (T_s - T_f) \tag{4}
\]

where \(A_0\) is the specific surface area of the transport pores, as estimated from experimental data in the literature [24], \(h_i\) is the interfacial convective heat transfer coefficient, which was estimated from the correlation in [23], \(v\) is the velocity of methanol vapour in the transport pores, \(T_s\) and \(T_f\) are the temperatures of the solid phase and the fluid phase, respectively, and \(H\) is the sorption heat.

![Fig. 2. P–T–x diagram of the basic adsorption cycle, redrawn from Critoph [16].](image)
3.2.3. Mass transfer

During the heating-desorption process (process 1–2–3), methanol vapour moves towards the mass transfer channel from outer positions due to the temperature-difference-caused pressure gradient (i.e., the gas density gradient). For the cooling-adsorption process (process 3–4–1), the mechanism of diffusion is different; both the temperature-difference-caused density gradient and the external fluid (from the evaporator) result in convection. The latter factor, i.e., the external pressure gradient, is represented through a transient boundary condition, which will be introduced in the following section on boundary conditions.

The diffusion process within the bed can be mathematically described as:

\[ \frac{\partial C}{\partial t} + (1 - \varepsilon) \cdot \rho_s \cdot \frac{\partial (\rho_x M)}{\partial t} = \frac{1}{\chi} \frac{\partial}{\partial r} \left( D_{eq} \frac{\partial C}{\partial r} \right) \]  

(5)

where \( C \) is the local density of the gaseous methanol; \( x \) is the concentration of the adsorbed methanol (in micropores of the carbon); \( M \) is the molar mass of methanol; \( \varepsilon \) is the porosity of the carbon bed; and \( D_{eq} \) is the effective diffusion coefficient. The value of \( D_{eq} \) was assumed to be constant in the calculation because the variation of its value is negligible for a narrow range of system pressures and temperatures, according to a previous adsorption kinetics study [27].

3.2.4. Supplemental equations

Darcy’s law and a specifically modified permeability for a packed bed with porosity between 0.35 and 0.67 are employed to describe the vapour velocity, which can be written as:

\[ \mathbf{v} = -K \frac{\partial P}{\partial r} \]  

(6)

\[ K = \frac{\varepsilon^5 S^2}{56} d_h^2 \]  

(7)

Notably, \( d_h \) is not the physical diameter of the transport pores, but the hydraulic diameter, which can be approximately calculated by \( d_h = 6/A_0 \) [25].

In addition, a correlation of the methanol saturation pressure and temperature, following the ideal gas law, is cited as follows [28].

\[ \ln P = 18.587 - 3626.55/(T - 34.29) \]  

(8)

\[ C = P/RT \]  

(9)

3.2.5. Initial conditions

For thermal process 1–2–3 (see Fig. 2):

\[ T_i(t) = T_f(t) = T_{in1} = T_1 \]  

(10)

\[ C(r) = C_{in1} = C_1 \]  

(11)

For thermal process 3–4–5:

\[ T_i(t) = T_f(t) = T_{in2} = T_3 \]  

(12)

\[ C(r) = C_{in2} = C_3 \]  

(13)

3.2.6. Boundary conditions

For thermal process 1–2–3:

\[ T_i|_{r=0} = T_{in1} = T_{in2} = T_{hw} \]  

(14)

\[ \frac{\partial T}{\partial r} |_{r=0} = 0 \]  

(15)

\[ \frac{\partial C}{\partial r} |_{r=0} = 0 \]  

(16)

\[ C_i|_{r=0} = C_i \]  

(17)

\[ C_i|_{r=R_0} = C_i \]  

(18)

For thermal process 3–4–5:

\[ T_i|_{r=R_0} = T_{in3} = T_{in4} = T_{cw} \]  

(19)

\[ \frac{\partial T}{\partial T} |_{r=R_0} = 0 \]  

(20)

\[ \frac{\partial C}{\partial T} |_{r=R_0} = 0 \]  

(21)

\[ C_i|_{r=R_0} = C_i \]  

(22)

\[ C_i|_{r=R_0} = C_i \]  

(23)

\[ C_i|_{r=R_0} = C_i \]  

(24)

\[ C_i|_{r=R_0} = C_i \]  

(25)

In the above boundary conditions, Eqs. (22) and (25) represent the transient temperature and the concentration of methanol from the evaporator after it is connected to the generator. This transient process is neglected in almost all previously proposed models.

The temperature and density of the methanol vapour from the evaporator are not constant at the moment when the evaporator is connected to the generator (at point 4). The pressure stabilises from \( P_t \) to \( P_e \), which results in a transient condition of vapour temperature and density. After this stage, the temperature and density of methanol vapour from the evaporator reach and remain at stable values. A mathematical description of the transient pressure process has been introduced to a model that was based on an assumption of local thermal equilibrium (see reference [4]). In the present model that considers local thermal non-equilibrium, the same concept of the transient pressure process is employed but is computed in a different way in solving governing equations, due to the two temperature treatment of phase and gaseous fluid. For coherent illustration, the definition of the transient pressure (\( P_{tr} \)) is presented here once again, which is defined as:

\[ P_{tr} = \frac{P_t - P_e}{1 - \varepsilon t} + P_e \]  

(26)

where \( P_t \) and \( P_e \) represent the pressures in the evaporator corresponding to status points 3’ and 4’ (see Fig. 2); \( t \) is time; and \( \varepsilon \) is a coefficient that reflects the time required for the pressure to drop to \( P_t \) from \( P_e \). The value of \( \varepsilon \) is determined by experiments associated with fitting techniques.

Substituting Eq. (26) into Eq. (9), the correlation of the transient vapour temperature, density and pressure can be expressed as:

\[ C_{t-tr} = C_{t-tr} \]  

(27)

where \( C_{t-tr} \) and \( C_{t-tr} \) denote the transient vapour temperature and transient vapour density.

3.3. Numerical solution

The operating conditions \( T_{cw} = 25 \, ^\circ C \), \( T_{inw} = 90 \, ^\circ C \), \( P_t = 7.14 \, kPa \) (corresponding to \( T_t = 10 \, ^\circ C \)) and \( P_e = 16.36 \, kPa \) were chosen for the numerical study. The primary data used in numerical calculations are given in Table 1.
The set of governing equations (i.e., Eq. (1)–(9)) associated with the corresponding initial and boundary conditions were solved using COMSOL Multiphysics 3.5a, and the numerical results were post-processed in a Matlab environment. A set of partial differential equations (PDEs), i.e., four PDEs, was established using two transient heat transfer modules, a transient diffusion module and a self-defined general-form PDE for the LDF equation (Eq. (1)). The transient boundary condition for the gas density, i.e., Eq. (27), was used in the governing equations through a self-defined function.

Grid independence tests were conducted on three one-dimensional uniform grid systems (100, 250 and 500) and the time-step independence tests were conducted with three dimensional time-steps of 0.25 s, 0.1 s and 0.05 s. The present study adopted the grid system 250 and time-step 0.1 s in consideration of the computation accuracy and the computing time. After solving these governing equations, the results, such as the temperature, concentration and pressure of each elementary layer, were processed and plotted in a Matlab environment.

### 4. Experimental apparatus and methods

A schematic diagram of the test rig and a detailed view of the generator are shown in Figs. 3 and 4. The configuration of the experimental prototype of the ART was modified for testing, and included spindled evaporator, an enlarged generator and a non-compact connection. Water baths were used as the simulated heating source and cooling source. The temperature was measured by a directly inserted thermocouple with an accuracy of 0.1 °C. The thermocouples and the data logger were separately calibrated by Fluke calibrators. The volume of the evaporator was calibrated using a volume calibration syringe. The accuracy of the volume measurement for desorption/adsorption amount is of 0.5 ml. The pressure was measured by a pressure gauge with an accuracy of 0.4 kPa.

The experiments were carried out to simulate the real operating procedures of the ART. In the real cycle of an ART, the evaporator also functions as a receiver during the condensing/generation process, and its temperature (and pressure) therefore is the same as the condensing temperature (and pressure) at the end of condensing process. In the evaporating/adsorbing process, the evaporator is connected to the generator until the pressure of the generator reaches the designed evaporation pressure, which results in the pressure jump process. In other words, the pressure jumps to the condensing pressure from evaporation pressure.

During the experiments, the experimental procedures were organized as below to simulate the real cycle stated above. Detailed experimental procedures are:

1. Heat and degas the carbon bed simultaneously for 12 h and then isolate the generator by closing valve V2.
2. Fill the evaporator with methanol and maintain the evaporator at $T_0$. The amount of methanol in the evaporator is slightly higher than the amount $x_{\text{max}} \times M_{\text{cb}}$ to compensate for the loss incurred during the vacuum process.
3. Charge methanol into the carbon bed. The amount charged is $x_{\text{min}} \times M_{\text{cb}}$.
4. Set the status of the generator to point 3 (see Fig. 2) by circulating water (at $T_3 = T_{\text{in}}$) in the water jacket until the carbon bed temperature is constant at $T_3$.
5. Circulate cooling water (i.e., $T_{\text{cw}}$) to generate the isosteric cooling process, in which the generator is isolated from the evaporator.

### Table 1 Parameters for the simulation.

<table>
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<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
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</table>

$M_{\text{cb}}$ | 2 | kg
(6) Record the temperatures and pressures during this process. The temperature measured in the bed, i.e., by temperature sensors TS1, TS2 and TS3, represents the temperature of the solid phase. The influence of the fluid can be neglected. A detailed explanation can be found in the literature [29].

(7) When the pressure reaches the designed evaporator pressure, connect the evaporator to the generator by opening valve V1.

(8) Record the adsorption amount using the liquid level indicator of the evaporator and continue recording temperatures and pressures.

After the cooling water has circulated for a sufficient amount of time, the generator reaches status point 1, i.e., the temperature and concentration are evenly distributed. The experimental procedures for process 1–2–3 are basically the same as above, with the addition of following steps.

(1) Circulate hot water (T_{hw}) to generate an isosteric heating process and, at the same time, circulate cooling water in the water jacket of the condenser.

(2) Record the temperatures, pressures and amount of desorbed refrigerant.

5. Results and discussion

Fig. 5 shows adsorbent (activated carbon) temperature histories measured at three radial positions of the carbon bed corresponding numerical simulation results for the first half of the cycle, i.e., process 1–2–3. Fig. 5 shows that the experimental data agree well with the numerical results over the entire process, except for a delay caused by the neglecting external heat transfer resistance. In addition, in Fig. 5, two observations are worth noting. One is that the temperature development is comprised of two stages, i.e., a sharp climb for the first several minutes, followed by a relatively gentle rise. It is thought that this phenomenon occurs because, as the macro desorption process begins, a considerable part of the input heat is transformed to desorption heat to desorb the methanol, rather than to sensible heat to increase the bed’s temperature. The other point is that the temperatures of the two relative inner positions of the cylindrical carbon bed are almost the same for approximately the first 10 min. This seems to go against the normal perception that there should be a clear difference among the temperatures of these three positions because the heating time is short and the heat conductivity of the bed is poor. The cause of this phenomenon is local or micro methanol migration, i.e., desorption and adsorption, between the carbon layers.

Fig. 6 shows the numerical prediction of the methanol concentration variation at the three radial positions, which is used to illustrate the micro concentration migration phenomenon mentioned above. In Fig. 6, it is clear that there are two stages in the concentration development at the three measured positions, i.e., stage 1–2 and stage 2–3. In stage 2–3, it is obvious that the concentrations at each position start to drop as the macro desorption process starts, after the system pressure reaches the condensing pressure. However, in stage 1–2, the concentration variation at different radial positions shows that the methanol migrates from the outer radial positions to the inner positions of the carbon bed. This phenomenon can be termed “centripetal migration” of the methanol, which is similar to the finding in [30]. During this “centripetal migration”, desorbed methanol from the outer layers is re-adsorbed by the inner carbon layers. The adsorption heat adds to the temperature increase of the inner layers of the carbon bed, resulting in the similar rates of temperature increase for the outer and inner carbon layers.

To validate the two-temperature treatment used in the model, the temperature development of the fluid phase (gaseous methanol) must also be experimentally validated. However, it is not practical to measure the gas temperature using inserted-type thermocouples as when measuring the carbon temperature, because the temperature sensor would be in contact with the solid phase. Therefore, a temperature sensor was arranged very close (r = 53 mm) to the inner boundary of the carbon column to measure the temperature at that point, which is assumed to be approximately equal to the gas temperature at the inner boundary (r = 55 mm). Fig. 7 shows that the simulated fluid temperature development exhibits good agreement with the measured data. According to Eichhorn and White [31], the temperature difference between the solid and gas phases is the same throughout the bed. Therefore, the temperature field of the gas phase in the bed is validated because the solid phase temperature and the one-point gas phase temperature were verified experimentally.

Fig. 8 shows a comparison of the pressure development obtained from the numerical results and experimental data. The predicted system pressure builds up slightly faster than the measured data. The reason is thought to be that the numerical results of the system pressure are actually the calculated pressure at the inner

Fig. 5. Comparison of experimental and numerical results: adsorbent temperature development at selected radial positions in process 1–2–3.

Fig. 6. Numerical results of adsorbate concentration variation at selected radial positions for the first 1200 s of process 1–2–3.
boundary of the carbon bed while the pressure measurement point is on the top of the condenser (see Fig. 3); thus, there is a delay between the simulated results and the pressure-gauge-measured data due to vapour building up in the volume in the condenser and connected pipes.

In terms of the dynamic desorption amount, a comparison between the experimental data and numerical results is shown in Fig. 9. Unfortunately, the simulated results do not agree very well with the experimental data. One main factor that could cause this discrepancy is the fact that some of the desorbed methanol condensed and adhered to the inner surface of the condenser and the tube connecting to the evaporator, causing the measured amount of desorbed methanol to be lower than the theoretical value.

Fig. 10 shows experimental data for the system pressure during process 3–4–5 and the predicted pressure development with $\alpha = 0.02$ (1/s) as well as the pressure development with $P_r = P_c$, which is the traditional treatment of the adsorption pressure in process 4–5. The experimental data show that the system pressure experiences a sudden jump just as the evaporator is connected to the generator (i.e., corresponding to point 4 in Fig. 2), and then the system pressure drops and gradually approaches the designed evaporator pressure. The parameter $\alpha$ in the mathematical definition of the transient pressure (i.e., Eq. (26)) was determined to be 0.02 (1/s) for this case, through a fitting technique using the experimental data. Using this transient pressure definition (i.e., Eq. (26)), this transient pressure process can be well represented. Comparing to using a transient pressure, the traditional treatment using a static adsorption pressure in process 4–5 misses a dramatic pressure jump, which makes such models unable to predict and reflect the impact of the pressure jump; however, such models have been used in the literature [13].

Fig. 11 shows the measured temperature histories against the simulated results from the proposed model (in this paper) with the transient pressure (at $\alpha = 0.02$ 1/s) and the results from current model but assuming a constant pressure, i.e. assuming $P_r = P_c$, respectively. It can be seen that the proposed model with the transient pressure is able to capture the temperature jump while using a constant pressure would fail to predict the detailed temperature development, such as the model in [13]. This indicates that the transient pressure process cannot be neglected in modelling a valve-controlled and long cycle-time based ART or other adsorption refrigeration system. The phenomenon of temperature jump has also been experimentally observed in [32].

Similar to the results shown in Fig. 6, the methanol migration phenomenon also occurs for process 3–4, as shown in Fig. 12.
However, in this process, the migration is not “centripetal migration”, but “centrifugal migration”. After this migration process, the concentration of the bed increases once the evaporator is connected to the generator to initialise the adsorption process. The dynamic adsorption amount for process 4–5 is compared for the experimental data and numerical results, as shown in Fig. 13. The comparison shows that the experimental data are in better agreement with the numerical results obtained by using the transient pressure process, as compared with desorption process 2–3 (Fig. 9), especially when the adsorption process takes longer than 3 h. The reason for the better agreement is that the condensation issue is naturally avoided in this part of the experiment. However, the agreement for the first 3 h of the adsorption process is not as good, which may be the result of passive adsorption caused by the transient pressure process which has not been thoroughly examined in the static adsorption equation (D–A equation) or LDF model. Additionally, the adsorption amount simulated with $P_t = P_e$ is compared with that simulated with the transient pressure assumption. The comparison in Fig. 13 shows that the result obtained for $P_t = P_e$ has an almost constant deviation (about 18%) from the experimental data. From Fig. 13, it can be seen that the deviation is mostly generated during the first hour of the adsorption process, and then, the gap is maintained throughout the rest of the adsorption process. In other words, the omission of a pressure jump leads to a deviation in the beginning of the adsorption process.

6. Conclusions

A dynamic model based on the typical configuration of a generator within an adsorption refrigeration tube was established with the consideration of a local thermal non-equilibrium and a transient pressure process. The model was numerically solved and experimentally validated. Numerical simulations with transient and constant adsorption pressures were conducted, respectively, to compare and highlight the impact of the transient pressure process. The conclusions of this study are summarized as follows:

1. The simulation results of the model show satisfactory agreement with the experimental data in terms of the adsorbent temperature development, system pressure variation, and dynamic desorption/adsorption amount. The model can be used for a valve-controlled and long cycle-time based ART or large adsorption systems with tubular generators.

2. The concept and empirical formula of the transient pressure process (at the beginning of the macro adsorption process) improve the mathematical model so that it can predict detailed features inside the adsorbent bed. For valve-controlled adsorption refrigeration systems, especially the daily cycle based systems, a model assuming a constant adsorption pressure is not suitable because the adsorbent temperature jump cannot be well predicted, and more importantly, the predicted adsorption amount is considerably lower than that predicted by the proposed model using a transient adsorption pressure.

3. The temperature of the gaseous fluid (methanol vapour) can be predicted by using a two-temperature treatment in the model. The fluid temperature prediction was experimentally validated using an approximated method.

References
