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Non-equilibrium numerical simulation to investigate the thermal contact resistance effect on the adsorption chiller performance

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***Abstract*— In recent years, using fossil fuels has increased due to population growth which leads to an increase in global concerns about environmental problems. Hence adsorption chiller systems have attracted much attention because of utilizing renewable and clean energies such as solar and geothermal ones and employing non-toxic refrigerants. In spite of mentioned advantages, poor heat and mass transfer in adsorption systems make them fall behind traditional vapor compression ones in global marketing. Thermal contact resistance between the adsorbent granules and thermal surfaces (metal tube and fins) impairs the adsorption performance by decreasing the coefficient of performance and specific cooling power. Therefore, in this paper, a transient three-dimensional non equilibrium numerical model is presented to evaluate the effect of thermal contact resistance on the performance of adsorption chillers. The spatial variations of pressure and gas flow inside the bed along with the internal and external mass transfer resistances have been considered in this modeling. The developed model is validated against the available experimental data in literature. As the results show, thermal contact resistance affects the heat and mass transfer in adsorbent bed and the cycle time decreases as a result of contact resistance elimination. Also it is found that the cooling and heating capacities increase by applying zero thermal contact resistance which is larger in quantity for the cooling capacity. Finally it is observed that the coefficient of performance and the specific cooling power improves when the thermal contact resistance is equal to zero.**

Keywords-nonuniform pressure; adsorption; contact resistance; performance

I. INTRODUCTION

Population growth leads to an increase in demand for energy and air pollution is becoming a fast growing problem in recent years. One of the main parts of our consumed energy is associated with air conditioning units such as chillers. Nevertheless, these traditional systems have a significant role in depletion of ozone layer in upper atmosphere due to using the hazard refrigerants such as chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HFCF). Therefore many attempts have been made to find alternative systems to be more compatible with environment [1].

The ability to use waste or renewable energy sources (solar and geothermal ones) and employing environmentally friendly refrigerants are the main advantages of adsorption systems [2]. However, these systems could not gain appropriate position in global marketing due to their poor performance as compared with traditional mechanical ones.

Many studies have been performed to improve the adsorption system performance in terms of coefficient of performance and specific cooling power. As adsorbent-adsorbate pairs have a dramatic effect on the adsorption performance, several studies concentrated their attention on introducing new pairs or improving their properties [3-8]. Some researchers also employed novel thermodynamic cycles and showed that these cycles could improve the adsorption system performance significantly [9-14].

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In addition to the aforementioned methods, many researchers investigated physical specifications of adsorbent bed [15-18]. They showed that using fins is an efficient method to facilitate heat and mass transfer through the adsorbent bed. Accordingly, the effect of geometrical characteristics of fins such as space, height, thickness and shape has been examined. Niazmand and Dabzadeh [19] presented a two-dimensional numerical model to investigate the effects of the annular fins on heat and mass transfer of adsorption chillers. They illustrated spatial variations of pressure and flow patterns in adsorbent bed. Results showed that using extended thermal surfaces influence the chiller performance in terms of coefficient of performance and specific cooling power. Rezk and Al-Dadah [20] modeled a 450 kW two-bed adsorption chiller with silica gel-water working pair to study the bed configuration effect on chiller performance. Their results showed that a decrease in fin space improves cooling capacity whereas the coefficient of performance decreases.

As it is evident, the heat and mass transfer in adsorbent is poor, due to low thermal conductivity and thermal resistance in adsorbent bed namely; thermal resistance between thermal fluid and metal tube, conduction thermal resistance of metal tube, thermal contact resistance between fins and tube, thermal contact resistance between sorption and metal surfaces, conduction thermal resistance in sorption and fins and finally thermal contact resistance between sorption and fins. Studies showed that one quarter of the entire mentioned resistance is related to the thermal contact resistance which exists between the sorption and thermal surfaces including fins and metal tubes [21]. It is worth mentioning that the effect of thermal contact resistance is more important when the thermal extended surfaces is employed [17]. Thus decreasing the thermal contact resistance could improve the adsorption kinetic.

How to place the adsorbent granules in bed has significant effect on the heat and mass transfer resistance in the adsorbent bed. In Most of the commercially available chillers the adsorbent granules are packed in adsorbent bed [22,23]. This method increases the contact thermal resistance which decreases the heat transfer rate throughout the bed [24]. Hence different methods have been suggested for troubleshooting and fixing this problem.

Schnabel et al. [25] used an empirical method to investigate the zeolite coating effect on adsorption kinetics. The experimental results showed that the heat transfer resistance between the adsorbent and the metal seemed to be lower than the reference sample. Furthermore, they found that the cooling power of adsorption system using zeolite coating is greater than the reference sample.

Restuccia et al. [24] presented a transient simulation of zeolite coated adsorbent bed to study the effect of heat transfer enhancement on the adsorption performance. Results showed that utilizing zeolite coated adsorbent bed increases the specific cooling power of the system.

While the coating technique eliminates the thermal contact resistance and increases the heat transfer rate, it reduces the permeability of adsorbent bed which causes the mass transfer to reduce.

Rezk et al. [21] developed a numerical model to examine the effect of eliminating contact resistance and metal additives in adsorbent bed on a two-bed silica gel-water adsorption chiller performance. They eliminate the contact resistance by coating the first layer of silica gel on the extended surfaces and packing the rest of silica gel granules. By applying this method, the permeability remains constant. The results indicated that by eliminating thermal contact resistance, the cooling capacity and the coefficient of performance increase. However, they exerted the log mean temperature difference approach which is equivalent to considering uniform pressure distribution throughout the adsorbent bed. It should be emphasized that uniform pressure approach decreases the degree of accuracy for some bed geometries [26].

The aim of this study is to investigate the effect of thermal contact resistance elimination on adsorption chiller performance. Therefore a three-dimensional transient model is presented to describe a silica gel-water adsorption chiller with plated fins. Also a non-uniform pressure approach is considered in the modeling to improve the accuracy of results.

II. SYSTEM DESCRIPTION

Fig. 1 presents a schematic of an adsorption chiller which consists of condenser, evaporator, expansion valve and adsorbent bed. In fact, the adsorbent bed is a heat exchanger where the adsorbent granules are placed on its surface and the refrigerant is regenerated by heating the sorbent while it is adsorbed when the adsorbent bed is cooled. Consequently the refrigerant circulates without requiring any mechanical power. Thus the adsorbent bed plays the compressor role in an adsorption system. The thermodynamic cycle of adsorption is comprised of four stages which are isosteric heating, isobaric cooling, isosteric cooling and isobaric cooling [2].

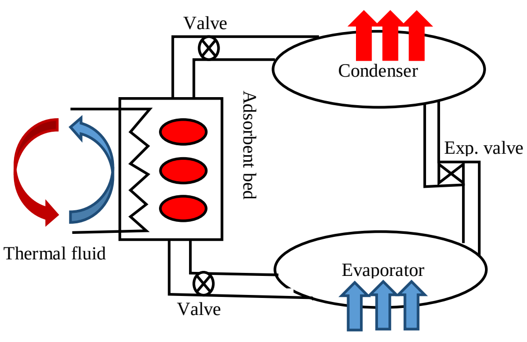

Fig 1: Schematic of an adsorption system

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III. NUMERICAL MODELING

In this study, a plate finned tube heat exchanger is employed as an adsorbent bed. The numerical domain is divided to the thermal fluid, metal tube, extended thermal surfaces (fins) and the sorption. Due to calculations cost and a large number of required grids, modeling of the whole bed is not practically reasonable. Therefore, symmetry conditions are applied to confine the solution domain to the area containing one quarter of a single tube and fins (Fig. 2). In table 1, the values corresponding to physical and operating conditions of adsorption chiller are listed. The metal tube and fins are made of copper and aluminum respectively. Water is assumed as the refrigerant and thermal fluid since it has appropriate thermo-physical properties such as low viscosity, non-toxic, supreme thermal conductivity and high latent heat [21,27].

A. Assumptions

Some simplifying assumptions which are considered in adsorption system modeling are as follows:

- Adsorbent granules are uniform in shape and porosity.
- Except of adsorbate gas density, the thermo-physical properties are considered constant.
- Refrigerant is assumed liquid and ideal gas in adsorbed phase and vapor phase respectively.
- Condenser and evaporator are considered ideal.

B. Governing equations

With regard to high Reynolds number of fluid flow in tube, the convective effects are neglected in the axial direction of metal tube. The energy balance for thermal fluid is calculated as follows:

$$\int_{cv} \rho_f c_{pf} \frac{\partial T_f}{\partial t} d\forall + \int_{cv} \nabla.\left(\rho_f c_{pf} \overrightarrow{U_f} T_f\right) d\forall = -Q_{\text{fluid-tube}} \quad (1)$$

Where ρ, C_p, T, t, U and Q are density, specific heat capacity, temperature, time, velocity and supplied heat and

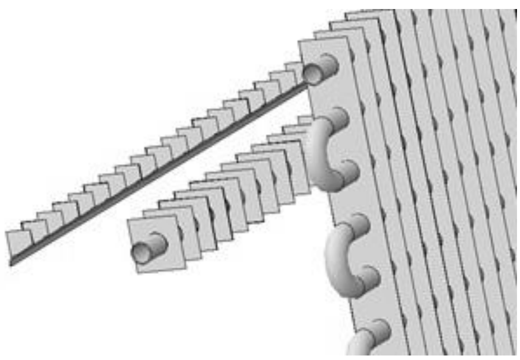

Fig 2: Schematic of adsorbent bed with numerical modeling domain

subscript f refers to thermal fluid. The heat transfer from thermal fluid to metal tube is determined as:

$$Q_{\text{fluid-tube}} = -h_f A(T_{\text{fluid}} - T_{\text{tube-wall}}) \quad (2)$$

Where h and A represent the heat transfer coefficient and area respectively. The heat transfer coefficient of thermal fluid is defined as:

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{m} \quad (3)$$

Where Nu, Pr and Re are Nusselt number, Prandtle number and Reynold number respectively. The coefficient, m, is equal to 0.3 and 0.4 for cooling and heating phases [28].

The transient three-dimensional energy equation for metal tube is written as:

$$\int_{cv} \rho_{\text{tube}} c_{\text{tube}} \frac{\partial T_{\text{tube}}}{\partial t} d\forall = \int_{cv} \nabla.\left(\lambda_{\text{tube}} \nabla T_{\text{tube}}\right) d\forall + Q_{\text{tube-fin}} \quad (4)$$

in which λ is the thermal conductivity.

Since fin thickness is negligible in comparison with its other dimensions, a two-dimensional heat conduction equation for fins is considered in the plane perpendicular to the metal tube axis.

$$\int_{cv} \rho_{\text{fin}} c_{\text{fin}} \frac{\partial T_{\text{fin}}}{\partial t} d\forall = \int_{cv} \nabla.\left(\lambda_{\text{fin}} \nabla T_{\text{fin}}\right) d\forall + Q_{\text{fin-bed}} \quad (5)$$

TABLE 1: GEOMETRICAL PARAMETERS VALUES AND OPERATING CONDITIONS

Parameter	Symbol	value	Unit
Internal diameter of the metal tube	D_i	10	mm
External diameter of the metal tube	D_o	12	mm
Fin thickness	FT	0.2	mm
Fin height	FH	32	mm
Fin space	FS	8	mm
Density of adsorbent	ρ_b	700	kgm^{-3}
Specific heat of the solid adsorbent	C_{pb}	924	$Jkg^{-1}K^{-1}$
Thermal conductivity of solid adsorbent	λ	0.1	$Wm^{-1}K^{-1}$
Bed porosity	ε_b	0.36	-
Particle mean diameter	d_p	200	μm
Heat of adsorption	ΔH	2760	$kJkg^{-1}$
Evaporator temperature	T_{evap}	283.15	K
Heating temperature	T_{heat}	363.15	K
Cooling temperature	$T_{cooling}$	303.15	K
Mass flow rate of thermal fluid	$\dot{m}_f$	0.03	Kgs^{-1}

It is worth mentioning that the above equation is written in general coordinate system because the grids associated with fins are non-orthogonal.

In most studies, pressure distribution is assumed uniform throughout the adsorbent bed. However in present study, regardless of its heavy calculation cost, the non-uniform pressure approach is used to solve the governing equations. In other words, inter-particle mass transfer resistance is considered. Therefore, the continuity, momentum and energy equations should be solved simultaneously.

The energy balance for the adsorbent bed is given by:

$$\int_{cv}(\rho C_p)\frac{\partial T_b}{\partial t}d\forall+\int_{cv}\nabla.\left(\rho_g c_{pg}\overrightarrow{U_g}T_b\right)d\forall=\int_{cv}\nabla.\left(\lambda_{bed}\nabla T_{bed}\right)d\forall$$
$$+\int_{cv}\rho_b\Delta H\frac{\partial w}{\partial t}d\forall \qquad (6)$$

Where the subscripts b and g refer to solid particles and gas phase respectively. ρC_p is the total heat capacity which is a function of density and heat capacity of desorbed vapor, adsorbate and dry adsorbent [18].

To determine the adsorbed amount, w, the linear driven force model is employed. In fact this model determines the intra-particle mass transfer resistance in an adsorbent bed. The linear driven force model is evaluated as:

$$\frac{dw}{dt}=15\mathrm{D_{so}}\mathrm{Exp}\left(-\frac{\mathrm{E_a}}{\mathrm{R_u T_b}}\right)/\mathrm{R_p^2}\left(w^*-w\right) \qquad (7)$$

Where w^*, D_{so}, E_a, R_u and R_p are equilibrium uptake, activation energy, pre-exponent constant of surface diffusivity, universal gas constant and particle radius respectively [29].

The continuity equation is written as:

$$\int_{cv}\varepsilon_t\frac{\partial\rho_g}{\partial t}d\forall+\int_{cv}\nabla.\left(\rho_g\overrightarrow{U_g}\right)d\forall+\int_{cv}\rho_b\frac{\partial w}{\partial t}d\forall=0 \qquad (8)$$

Where εt refers to the total porosity of bed [17].

Darcy law is used to determine the velocity of the vapor among the adsorbent particles [30].

$$\overrightarrow{U_g}=-\frac{k_{app}}{\mu}\vec{\nabla}P \qquad (9)$$

Where k_{app} is the permeability of the bed and μ refers to the vapor viscosity [8].

Finally the equation of state is used to evaluate the density of refrigerant vapor in desorbed phase:

$$P=\rho_g R_g T_b \qquad (10)$$

Parameters describing the adsorption performance in terms of specific cooling power (SCP) and coefficient of performance (COP) are defined as follows:

$$\mathrm{SCP}=\frac{Q_{evap}}{m_{bed}\times t_{cycle}} \qquad (11)$$

$$\mathrm{COP}=\frac{Q_{evap}}{Q_{heating}} \qquad (12)$$

C. Boundary conditions

The energy balance for thermal fluid is solved in the axial direction, hence only one boundary condition is needed. Thus the thermal fluid temperature at the inlet of tube is considered equal to hot water and cold water temperatures for heating and cooling stages respectively.

For the interface between the metal tube and the adsorbent and fins, conduction heat flux is assumed as boundary condition while for the interface between the adsorbent and chamber, zero temperature gradient is considered.

Also zero pressure gradients are applied to all adsorbent bed boundaries except for the bed chamber interface, where pressure is set equal to the chamber pressure.

D. Numerical procedure and validation

In order to validate the numerical modeling, numerical results are required to be compared with an empirical sample. Thus, the geometrical specifications and the operating conditions of the numerical scheme have been adjusted to their counterparts in the tested lab-scale chiller of Restuccia et al. [31].

Fig.3 compares the time variations of mean temperature and pressure of the lab-scale adsorbent bed with present modeling. Considering simplifications and assumptions, it is observed that the numerical solution is in reasonable agreement with experimental results with maximum error of 6% in temperature calculation.

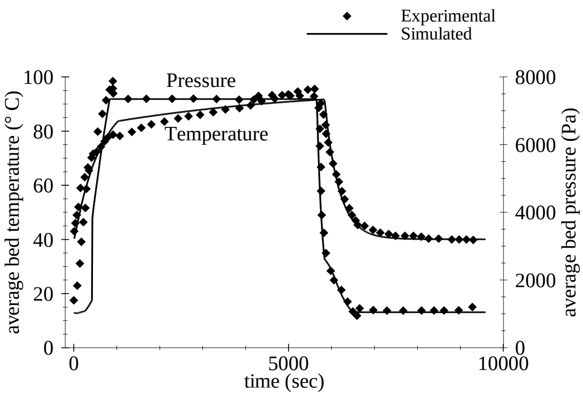

Fig 3: Comparison between the numerical modeling and empirical results

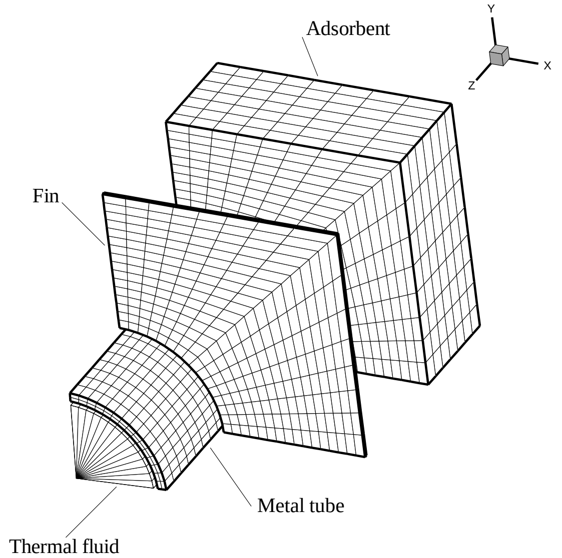

Fig 4: Schematic of the grid system

The set of governing equations are solved simultaneously using finite control volume method and a fully implicit scheme. Fig. 4 shows the adopted control volumes in all various domains. Forward differencing scheme is used for unsteady terms and central differencing scheme is employed for both of the diffusion and convective ones.

IV. RESULTS AND DISCUSSIONS

In Fig. 5 the time variations of total mass flow rate of the bed is plotted. It should be mentioned that the positive values refer to output mass from the adsorbent bed whereas the negative values stand for the input mass to adsorbent bed. Except for the early times of the isobaric heating and cooling stages, the total mass flow rate is higher when the thermal resistance is zero. This is due to the superior heat transfer which increases the adsorption and desorption rates.

Fig. 6 represents the variations of cycle time as function of fin height at different fin spaces. Since the heat transfer surfaces is proportional to fin spaces, it is observed that the cycle time decreases at smaller fin spaces while it increases with fin height due to higher mass transfer resistance for thicker sorption layer. Fig. 6 shows that eliminating the thermal contact resistance improves the cycle time. For instance, at fin space of 10mm, cycle time decreases 17.6%, 14.6% and 12.7% for fin heights equal to 20mm, 24mm and 28mm respectively. Clearly, eliminating thermal contact resistance causes the heat transfer rate to increase and decreases the cycle time.

Fig.7 exhibits the heating capacity variations with fin spaces at different fin heights. Heating capacity represents the total supplied heat by the thermal fluid in desorption process. As the fin spaces increase, the thermal surfaces reduce and as a result, the heating capacity decreases.

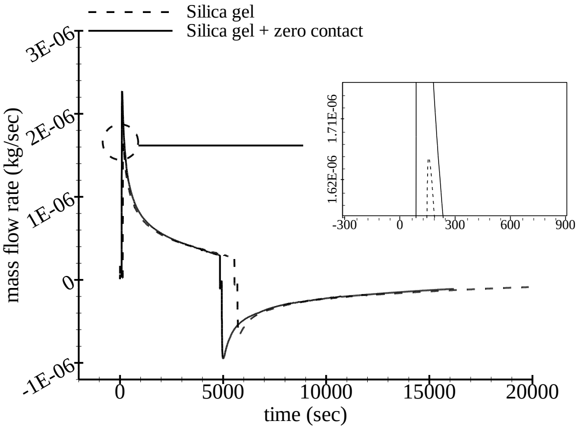

Fig 5: Time variation of total mass flow for beds with and without thermal contact resistance

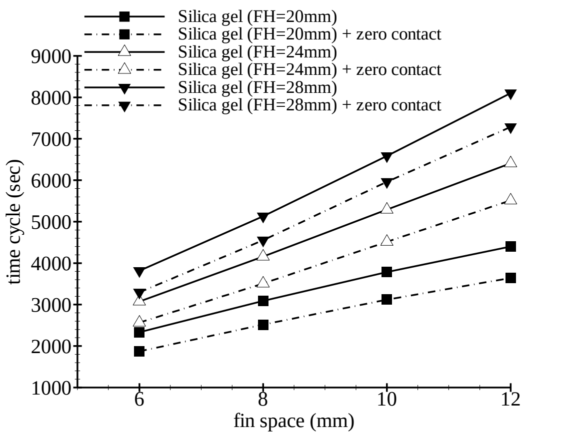

Fig 6: The variation of cycle time with fin space for different fin heights in the bed with and without thermal contact resistance

However, as the thermal surfaces and the adsorbent mass increase at higher fin height, the change in the heating capacity is in opposing direction with the fin space. It could be observed that omitting the thermal contact resistance significantly increases the heating capacity.

Fig. 8 shows the cooling capacity variations for different fin heights and spaces. The cooling capacity is the cooling rate produced by the evaporator. Similarly, the cooling capacity is influenced by contact resistance. At fin space equal to 6mm, the cooling capacity improves by 20.7% and 16.8% for fin heights of 24mm and 28mm respectively. Also the cooling capacity increases approximately by 17.1% and 14% at fin space of 12mm for fin heights equal to 24mm and 28mm respectively. Hence it is concluded that the contact resistance is a significant factor affecting the heat and mass transfer throughout the bed.

Prior to discussing the adsorption performance in terms of COP and SCP, it is worth mentioning one of the

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important advantages of adsorption systems which is their ability to use the renewable and waste energies. Therefore the cost consumed for supplying the driven energy is low. As the COP is the ratio of the cooling capacity to the heating capacity; its variation becomes important when using expensive energy sources like fossil fuels. On the contrary, the SCP parameter is associated with economic costs of system construction and its size. Consequently, as the main reason of employing adsorption systems is their ability to apply renewable and low grade energies, the SCP is the most important parameter in evaluating the adsorption system performance.

In Fig. 9 the effect of thermal contact resistance on COP of the system is presented. It is observed that the variation of COP with fin space is negligible. However, increasing fin height leads to an increase in COP. In spite of higher heating capacity for zero contact case at a given fin space, it is observed that the COP increases when the thermal contact resistance is eliminated. The reason could be attributed to the cooling capacity which enhances at higher rate comparing with the heating capacity. For example at fin space of 10mm and fin height of 24mm, the cooling and heating capacities increase averagely by 17.92% and 17.60% respectively.

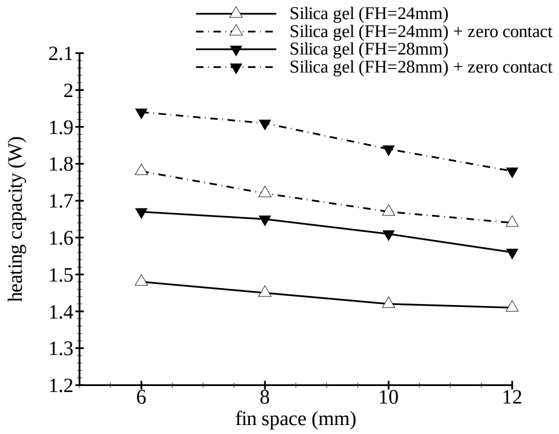

Fig 7: The variation of heating capacity with fin for different fin heights in the bed with and without thermal contact resistance

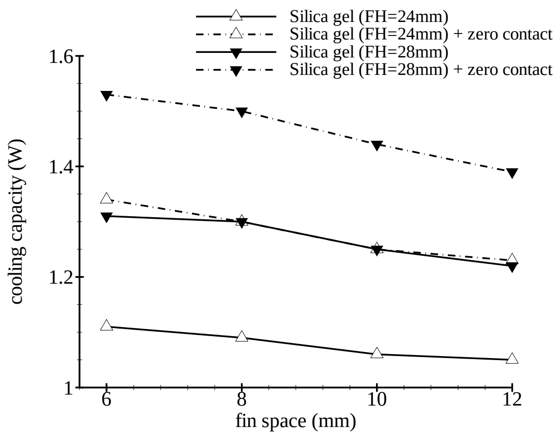

Fig 8: The variation of cooling capacity with fin space for different fin heights in the bed with and without thermal contact resistance

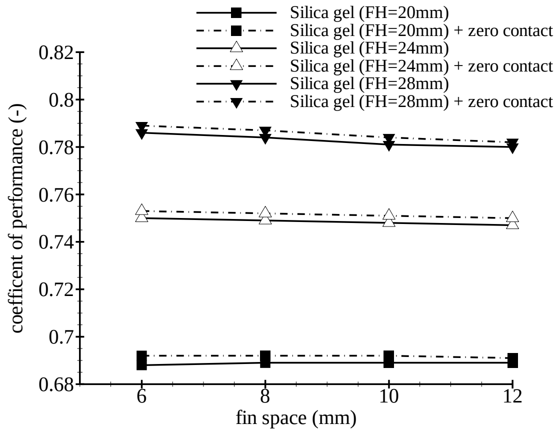

Fig 9: The variation of coefficient of performance with fin space for different fin heights for the bed with and without thermal contact resistance

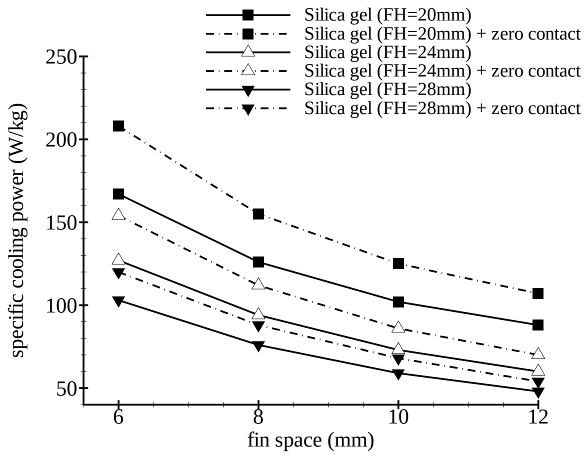

Fig 10: The variation of specific cooling power with fin space for different fin heights for the bed with and without thermal contact resistance

Fig. 10 shows the SCP variation for different fin heights and spaces. As the heat transfer increases by employing fins, SCP decreases with an increase in fin space. Also it is concluded that SCP decreases at higher fin heights. Both mass flow rate and time cycle are important in determining the SCP corresponding the “11”. As mentioned previously, mass flow rate and time cycle improves for bed with zero thermal contact resistance. Apparently as shown in Fig. 10, eliminating the thermal contact resistance enhances the SCP. For instance, at fin space of 6mm, the SCP increases by 25.15%, 21.25% and 16.50% for fin heights of 20mm, 24mm and 28mm respectively, and at fin space equal to 12mm, the SCP improves by 21.50%, 16.66% and 12.50%.

V. Conclusions

A detailed transient three-dimensional modeling of a silica gel-water adsorption chiller is developed to examine the effect of eliminating the thermal contact resistance on the adsorption performance in terms of coefficient of performance and specific cooling power.

A control volume technique and fully implicit scheme is employed to discretize the equations. Also the inter-particle mass transfer resistance in adsorbent bed is considered due to the non-uniform pressure assumption.

It was found that the thermal contact resistance influences the adsorption kinetics. The results showed that cycle time and total mass flow rate which affect the specific cooling power directly, improve at zero thermal contact resistance. Thus at a given fin space and height, the specific cooling power enhances significantly for the bed with zero thermal contact resistance.

The cooling and heating capacities also increases in the absence of the thermal contact resistance. The results indicated that the cooling capacity at given fin space and height enhances at higher rate comparing with the heating capacity. Thus eliminating the contact resistance causes the coefficient of performance to increase.

Eventually, as both the SCP and COP improve, the whole cost of the adsorption system in terms of the supplied energy and construction reduces dramatically.

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