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Review

Analysis of Designs of Heat Exchangers Used in Adsorption Chillers

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Abstract: In the face of increasing demands with regard to the share of renewable energy sources in the energy mix, adsorption chillers are becoming a potentially important part of the energy transition. A key component of this type of equipment is the heat exchanger in the adsorption bed, the design of which affects both heat and mass transfer. This study includes an analysis of the geometry and materials used to manufacture such heat exchangers. The geometry analysis is mainly based on the evaluation of the impact of the different dimensions of the exchanger components on heat and mass transfer in the bed. The second part of the study focuses on material-related issues where the main emphasis is on the analysis of the thermal inertia of the exchanger. The paper analyses the latest research on the design of exchangers in adsorption beds, mainly from 2015–2021. Currently, the commonly used SCP and COP coefficients and various test conditions do not provide sufficient information for comparative analysis of adsorption bed heat exchangers, so the authors propose to introduce a new index for the evaluation of heat exchangers in terms of the effect of the design parameters on the energy efficiency of an adsorption chiller.

Keywords: adsorption chiller; heat exchanger; heat and mass transfer; HEX heat capacity

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1. Introduction

The policy currently pursued, both globally and locally, is based on the strategy of sustainable development, i.e., meeting the needs of contemporary society without limiting the development opportunities for future generations. It should be stated that for the rational management of raw materials, decarbonization, increased energy efficiency, and environmental protection are the main components of a sustainable development policy. The quest for a balance between the economic and the environmental aspects is one of the main challenges of modern times [1,2]. Bearing in mind that the residential sector is largely responsible for growing electricity consumption [3], it becomes an important need to look for alternative, environmentally friendly heating and cooling solutions.

Refrigeration, understood as keeping the temperature lower than the ambient temperature, has a huge impact on industry, lifestyle, economy, and consequently the environment. Electricity consumption for cooling purposes has increased three-fold relative to 1990 [4]. This results in a heavy load on power grids and a continuous increase in carbon dioxide emissions. CO_2 emissions resulting from cooling are approx. 1.13 Gtonnes and global CO_2 emissions are approx. 34 Gtonnes [4]. This is due to the fact that 8.5% of the global energy consumption is accounted for by the refrigeration industry. Nevertheless, during extremely warm days, refrigeration accounts for up to 50% of energy consumption at peak times [3,4]. It is the occurrence of increasingly higher peaks in energy consumption that is the biggest problem as they significantly disrupt the operation of electricity systems. Due to the increasing standard of living and frequent occurrence of heat waves, this trend is expected to continue in the coming years. The continuation of the current development of refrigeration will make it impossible to meet the climate targets—above all a significant reduction in carbon dioxide emissions. Therefore, the growing demand for cooling should be accompanied by the development of alternative cooling technologies in order to guarantee access to cooling for all consumers without a negative impact on the environment [3,5].

The first commercially available refrigerators that were designed based on the phenomenon of sorption appeared as early as in the 19th century, but it was only in the 20th century that the market of compressor refrigerators was established, the further development of which turned out to be much more intensive than that of sorption chillers. It should be added that refrigerators weighed several tons at the turn of the 20th century and were not a commercial success. Hence, the disadvantages of sorption chillers, such as great weight and considerable dimensions, can be eliminated by making efforts towards intensifying heat and mass transfer in the sorption bed, which may result in a significant development of these alternative cooling appliances.

Adsorption chillers differ from compressor ones primarily in terms of the type of compression of the cooling medium. Compression chillers have a mechanical compressor, while adsorption chillers work with a thermal compressor. The operating cycle of an adsorption chiller consists of four basic phases (Figure 1). In the evaporator, the cooling medium is evaporated at a boiling pressure appropriate for the particular cooling medium, as a result of which heat is removed from the refrigerated space (Q_{evap}). To maintain low pressure in the evaporator, the cooling medium vapors are directed to the sorbent bed where they are adsorbed by the adsorbent material. The process of adsorption is exothermic; therefore, heat (Q_{ad}) must be removed from the bed to maintain adequate adsorption performance. Once the adsorption bed is saturated, another transformation, i.e., preheating, takes place to prepare the bed for desorption. During the pre-heating, the bed is pre-heated by heat (Q_h) being provided to it. The next phase is bed regeneration, i.e., desorption. Depending on the sorbent used, heat (Q_d) of a specific temperature is supplied to the bed and the cooling medium evaporated during the process of desorption is transferred to the condenser, which is cooled down. The condensed cooling medium enters the evaporator and the heat of condensation (Q_{cond}) is released to the environment. When the adsorption bed is completely desorbed, the process of pre-cooling begins. This involves cooling the bed until the adsorption temperature is reached and the heat (Q_c) thus obtained can still be used—the adsorption phase with heat recovery [6].

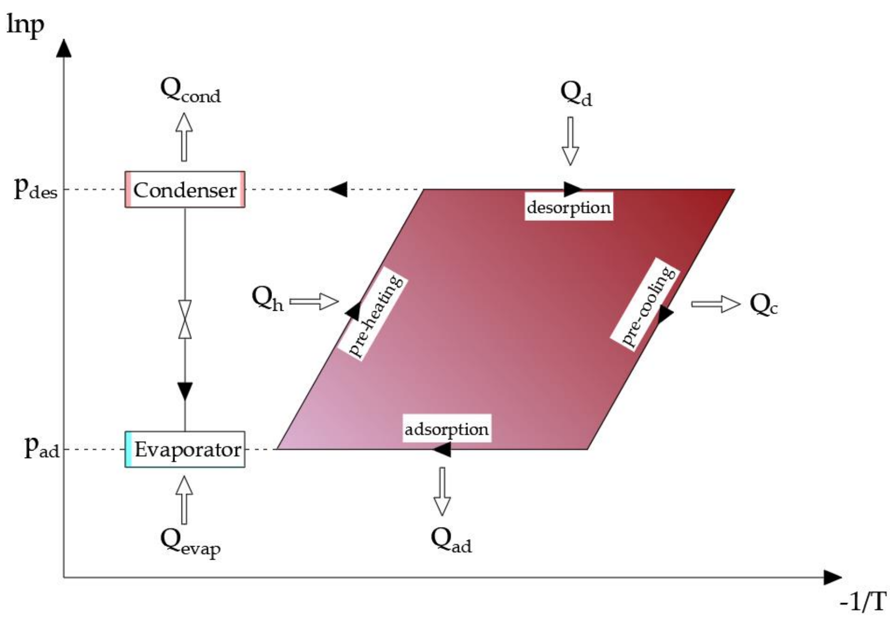

Figure 1. Basic operating cycle of a two-bed adsorption chiller.

When analyzing the adsorption refrigeration cycle presented, one should pay attention to the fact that heat and mass transfer plays a key role in each transformation. It influences the values of basic parameters characterizing an adsorption chiller, i.e., Specific Cooling Power (SCP) and Coefficient of Performance (COP), which are described by expressions 1 and 2 [6]:

$$\mathrm{SCP} = \frac{Q_{evap}}{m_a \cdot t_{cycle}}, \tag{1}$$

$$\mathrm{COP} = \frac{Q_{evap}}{Q_d + Q_h}, \tag{2}$$

where, Q_{evap} is the amount of heat removed from the space cooled by the cooling medium, m_a is the mass of the adsorbent, t_{cycle} is the adsorption cycle time, Q_d is the amount of heat required to regenerate the bed, and Q_h is the amount of heat supplied to the bed during the pre-heating. It should be noted, however, that the SCP and COP values are instantaneous or averaged values related to specific operating conditions. Departure from these conditions causes a change in the values of these indicators [7].

When analyzing Equations (1) and (2) in the context of improving the performance of adsorption chillers, it should be stated that the values of SCP and COP are directly related to the design of the bed, which is the main element of the chiller and largely determines the sorption and thermal capacity of the system. The COP depends on how efficiently the driving energy is used in the processes of pre-heating and desorption and can be improved mainly by heat and mass recovery, extending the cycle time and reducing the thermal capacity of the system. On the other hand, the SCP parameter strongly depends on the adsorption working pair and the cycle time, so the improvement of SCP should be sought mainly in the proper choice of materials, intensification of the heat transfer in the bed, and improvement of mass transfer, especially during adsorption [8]. The analysis of the SCP and COP parameters carried out in this way draws attention to the need to improve heat transfer in the adsorption bed, but the modification of the heat exchanger design must be carried out with particular attention to maintaining adequate mass transfer in the bed.

When analyzing the options to improve heat transfer in the bed, new types of heat exchangers should be considered, and these can be classified into three groups, i.e., plate, plate-fin and finned tube heat exchangers. Current research in this area focuses on the analysis of the geometry of the fins and their configuration, i.e., values such as their radius and spacing. Common research conclusions are that heat transfer improves with increasing fin pitch and decreasing fin thickness [9,10]. Nevertheless, research is also being conducted on new heat exchanger designs that are based on plate and fin tube heat exchangers. One concept is the possibility of using an elliptical tube with a twisted tape insert that turbulizes the flow [11]. The geometry proposed by Mashayekhi et al. [11] allows for more intense heat transport and reduced pressure drop compared to classical tubular exchangers. The elliptical tube with insert significantly improves the mixing of the exchanger working fluid and effectively reduces the formation of a boundary layer, while providing efficient mixing in the flow core. Analyzing still other new heat exchanger designs, one should pay attention to the plate heat exchanger with zigzag configuration, which was presented by the team of Talebizadehsardari et al. [12] in the context of energy storage in phase change materials. The mentioned HEX was analyzed in terms of the effect of the zigzag angle on the heating of the phase change material. It should be noted that a larger zigzag angle translates into an increase in pressure drop and thus an increase in the necessary driving force of the HTF circulating pump by about 40%. Nevertheless, the PCM phase transformation time is reduced by about 40%. The study was conducted for HTF temperatures in the range of 45–55 °C, which allows us to analyze the investigated exchanger in the context of its application in an adsorption bed. Other ways to increase heat transfer in the bed are the use of heat pipes and modification of the bed by using coating, bonding techniques, and additives [10,13–19]. Nevertheless, the solutions proposed often involve an increase in the heat transfer surface area, which, beside the power of the exchanger, is the most important parameter that characterizes it. This fact makes it possible to reduce the thickness of the adsorbent layer, which means the bed heats up faster, but at the same period it causes an increase of the heat capacity of the heat exchanger (C), which is defined according to expression 3 [6] as the amount of heat that must be supplied to the system (ΔQ) in order to cause a unit change in its temperature (ΔT):

$$C = \frac{\Delta Q}{\Delta T}, \tag{3}$$

Due to the fact that adsorption cooling requires alternate heating and cooling of the bed, it becomes important to assess the heat capacity of all structural elements of the bed, i.e., primarily the design of the heat exchangers and housing. With reference to Equations (1) and (2) and Figure 1, it should be stated that an increase in the heat capacity of the adsorption bed results in a higher consumption of driving heat in the adsorption cycle due to the necessity of supplying more heat in the pre-heating process. This contributes to a decrease in the COP performance of the system. Another aspect of the increased heat capacity is the rapid heating of the adsorbent, which significantly reduces the desorption time while increasing the pre-heating and pre-cooling times. Therefore, the changing heat capacity of the bed is related to the cycle time, which directly affects the value of the SCP. On this basis, it should be concluded that the best energy effect resulting from the modification of the heat exchanger design will be obtained by supplementing the adsorption cycle with heat recovery, the duration of which should be optimized for specific operating conditions [20].

When analyzing the simplified schematic of an adsorption chiller presented in Figure 2 in terms of the heat capacity of the system, attention should be paid not only to the aforementioned heat exchanger in the bed; an equally important aspect is the heat transfer properties of the medium cooling/heating the bed during the successive stages of the adsorption cycle. It is the medium filling the heat exchanger that is the heat transfer medium. Therefore, it should be noted that some amount of heat medium is left in the bed after each stage of the cycle, which negatively affects the performance of the system. This draws attention to the need to analyze the distribution of the heat medium channels in the heat exchanger so as to avoid excessive heat loss from the system. On this basis, it should be concluded that the optimum solution is to use the same medium for bed heating and bed cooling because this avoids unnecessary increase in the liquid capacity of the system.

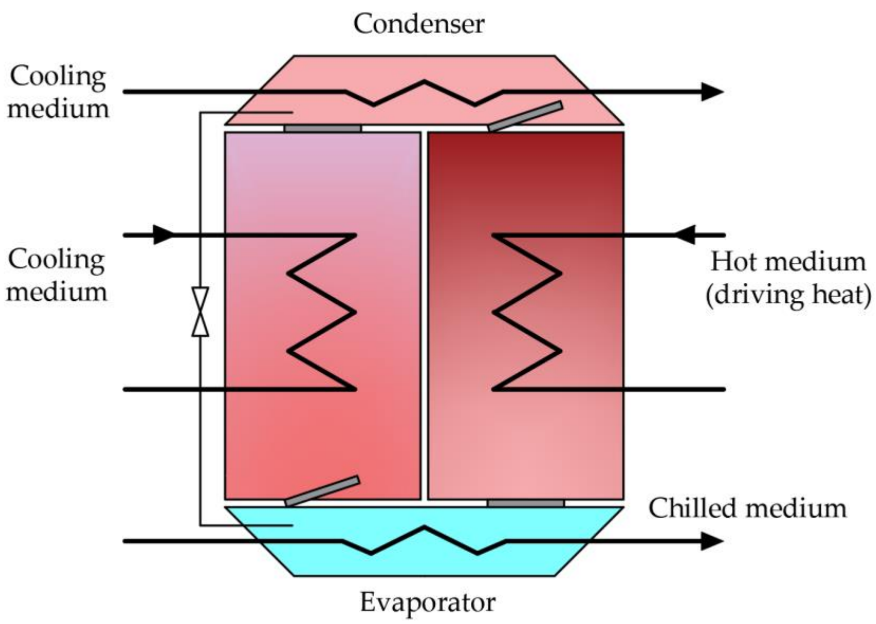

Figure 2. Design diagram of two-bed adsorption chiller.

The design of a heat exchanger in an adsorption bed should also take into account the working pair used. Adsorbates such as water or methanol require maintenance of low pressure, which affects the mass transfer in the bed. It is important to select a heat transfer fluid (HTF) with low viscosity and high thermal conductivity, such as water [6,7]. This draws attention to the fact that the sorbent as well as the design and material of the heat exchanger, and the HTF should be analyzed and selected in detail so that SCP and COP indices are optimized simultaneously [21]. Additionally, the selected materials must exhibit resistance to the operating conditions in the chiller, such as the corrosiveness of the atmosphere, the temperature, and the pressure.

Due to the complexity of the problem presented above, the main objective of this paper is to provide a review of the designs of heat exchangers used in adsorption chillers, based on papers published between 2015 and 2021, in the context of the geometries used, materials, manufacturing technologies, and characteristic parameters, such as power, heat transfer area, adsorbent to bed weight ratio, heat capacity, the shape of the channels filled with heat-transfer fluids, temperature distribution in the exchanger, and flow resistance. The analysis will include modifications of heat exchanger designs made in order to intensify heat and mass transfer in the sorption processes with the objective to improve the SCP and COP of adsorption chillers. The paper focuses on a list of parameters relevant to the design of heat exchangers for adsorption beds. The work undertaken is a first step towards the development an evaluation index for heat exchangers for adsorption chillers and to indicate possible directions for further work related to the modification of heat exchangers and adsorption beds.

2. Heat Exchanger Design

There are a number of aspects that need to be considered when evaluating the design of adsorption bed heat exchangers. The basic parameter for the evaluation of heat exchangers is the power expressed in watts. The power value is strongly related to the heat exchange surface area between the adsorbent and the components of the exchanger. The exchange surface can be increased by using additional elements, e.g., fins. The second important aspect is the optimum mass of adsorbent that can be embedded in the adsorption bed. The ratio of heat transfer surface area to adsorbent mass (S/m) is one of the most important factors besides SCP and COP in the studies analyzed. The S/m value indicates the potential intensity of heat transfer. The higher the value, the greater the proportion of adsorbent mass that is in direct contact with the exchanger components, resulting in better heat transfer. Another important aspect is the geometry of the heat transfer fluid (HTF) channels. The first parameter taken into consideration in this context is the surface area of the diaphragm, i.e., the wall separating the HTF from the adsorbent. Increasing this surface area minimizes the heat conduction resistance and additionally improves heat transfer by convection. The next issue is the temperature distribution on the surface of the heat exchanger. A uniform temperature distribution is expected, which can be achieved by proper channel arrangement. However, if the exchanger geometry is incorrectly designed, so-called dead zones may occur, i.e., areas with reduced HTF mass transfer intensity. This phenomenon generates losses related not only to non-uniform temperature distribution, but also to residual water from this zone of the exchanger. Residual water is the part of HTF remaining in the exchanger after the process of desorption or adsorption. At the level of the adsorption cooling system, the problem of energy loss for heating/cooling residual water is solved by flushing the exchanger. However, it should be noted that with inadequate HTF channel design, flushing has limited effectiveness. The last aspect considered in the context of HTF channel geometry is the flow resistance and the impact of the exchanger components on the nature of the fluid flow. It is advantageous for exchangers to have as low a flow resistance as possible, thus reducing the electrical energy required to power the circulation pumps. In terms of flow characteristic, it is beneficial for heat transfer and heat exchanger (HEX) self-cleaning to maintain turbulent flow. This can be achieved, e.g., by using obstacles that turbulize the flow. Well-designed exchanger geometry should provide sufficiently low resistance to the flow of the cooling medium vapor. This is achieved by using appropriately large fin pitch or additional vapor channels in the bed. A significant influence on the mass flow resistance of the refrigerant vapor is the pressure during adsorption. Depending on the refrigerant used, a classification is given between low pressure systems (water, methanol, ethanol as a refrigerant) and high pressure systems (ammonia, carbon dioxide as a refrigerant). The higher the pressure, the higher the average particle velocity and therefore the better the diffusion of the refrigerant vapor. As a result, for low-pressure systems it is important to minimize the vapor resistance. It should be noted, however, that more important in this case is the selection of the type of bed and of the adsorbent grain size. For high pressure systems on the other hand, it is necessary to ensure adequate durability of the heat exchanger and chamber components. A summary of the above parameters for the studies analyzed in this paper is presented in Tables 1 and 2.

Among the adsorption bed heat exchangers currently in use, there are basically three types of design, i.e., plate heat exchanger, plate-fin heat exchanger, and fin-tube heat exchanger. Most designs presented in the literature are based on the above-mentioned types. The few proposals for new designs include designs based on heat pipes [22] and metal foam [23]. Draft views of the aforementioned HEX designs are presented in Figure 3.

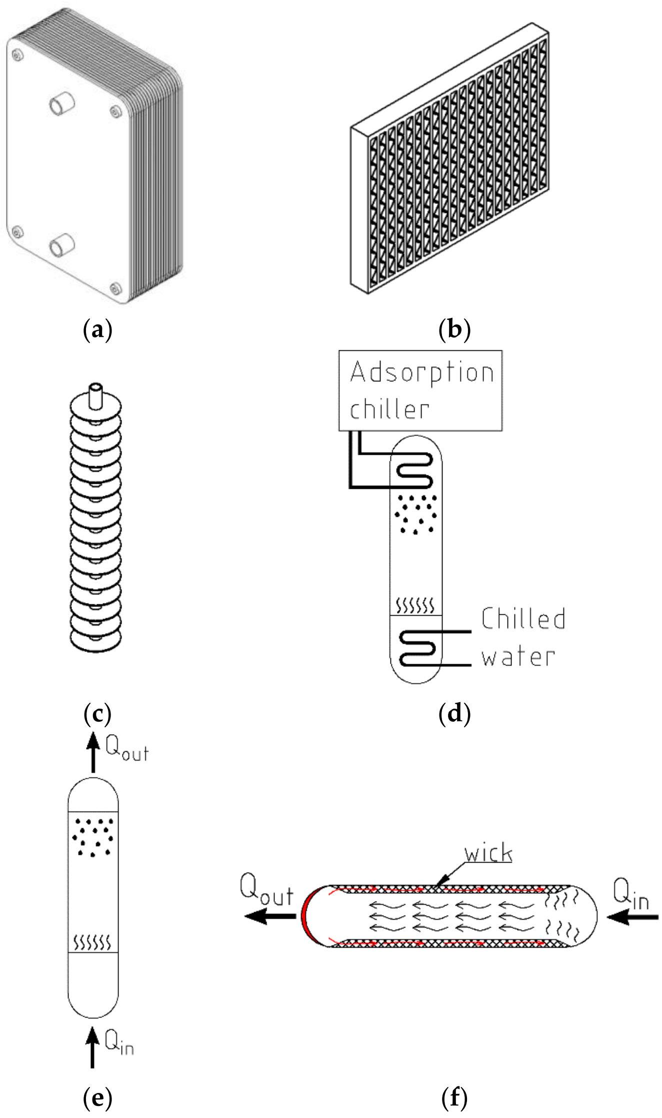

Figure 3. Types of heat exchangers used in adsorption beds: (**a**) plate HEX; (**b**) plate-fin HEX; (**c**) fin-tube HEX; (**d**) split type heat pipe; (**e**) vertical heat pipe; (**f**) horizontal heat pipe.

Table 1. Overview of plate and plate-fin heat exchangers used in adsorption beds.

Ref.	Heat Transfer Surface [cm^2]	Adsorbent Mass [g]	S/m Ratio [cm^2/g]	Dimensions [1] [mm]	Fin Pitch [mm]	Fin Thickness [mm]	Fin Height [mm]	Fin Width [mm]	Bed Type	SCP [W/kg]	COP [-]	Ratio of HEX Mass to Bed Mass [-]
[24] [2]	-	5990	-	-	3	0.5	14	20	packed	835	0.623	0.593
[24] [2]	-	4510	-	-	3	0.5	3	20	packed	1108	0.449	0.829
[24] [2]	-	5990	-	-	3	0.5	20	20	packed	835	0.638	0.566
[24] [2]	-	7030	-	-	3	0.5	20	20	packed	711	0.642	0.545
[24] [2]	-	5680	-	-	3	0.5	14	20	packed	880	0.619	0.611
[25] [2]	-	410	-	-	-	-	-	-	packed	51.6	0.5	0.399
[26] [2]	-	4800		h = 80, w = 120, l = 22	1.7	0.09			packed	161	0.17	0.714
[8] [2]	2300	49.86	46.1	h = 22, w = 120, l = 80	-	-	-	-	packed	3320	-	0.762
[8] [2]	2100	43.41	48.4	h = 40, w = 120, l = 40	-	-	-	-	packed	1360	-	0.750
[8] [2]	2000	46.05	49.9	h = 85, w = 120, l = 20	-	-	-	-	packed	170	-	0.752
[27] [2]	-	6000	-	h = 30, w = 365, l = 410	1.5	-	-	-	packed	140.1	0.26	-
[27] [2]	-	3600	-	h = 30, w = 170, l = 240	1.5	-	-	-	packed	169	0.28	-
[28] [2]	-	n/a	-	h = 170, w = 200, l = 200	1	0.05	0.9	-	coated	177	0.22	-
[29] [2]	83,100	2230	27.5	h = 22, w = 160, l = 508	-	-	-	-	coated/packed	1520–1970	0.3–0.35	0.649
[30] [2]	-	-	-	-	0.7–30	0.5–5	5–41	-	packed	150–551	-	
[31] [3]	-	-	-	h = 0.7	18	-	0.8	-	packed	622.8	0.512	-
[32] [3]	6180	500	12.36	-	-	-	-	-	coated/packed	1000–2000	-	0.489
[8] [3]	134	0.27	49.8	-	-	-	-	-	packed	4300	-	-
[13] [3]	-	700	-	h = 0.3, w = 54, l = 400	-	-	-	-	packed	-	-	-
[33] [3]	-	3300	-	h = 45, w = 313, l = 700	-	-	-	-	coated	1680	0.4	0.720

[1] h = height; w = width; l = length. [2] values for plate-fin HEX. [3] values for plate.

Table 2. Overview of fin-tube heat exchangers used in adsorption beds.

Ref.	Heat Transfer Surface [cm^2]	Adsorbent Mass [g]	S/m Ratio [cm^2/g]	Dimensions [3] [mm]	Fin Pitch [mm]	Fin Thickness [mm]	Fin Height [mm]	Fin Width [mm]	Bed Type	SCP [W/kg]	COP [-]	Ratio of HEX Mass to Bed Mass [-]
[34]	1880	820	2.29	-	9.53	0.007	90	90	packed	-	-	0.466
[34]	2660	770	3.45	-	6.35	0.007	90	90	packed	-	-	0.559
[35]	-	500	-	d = 15. D = 32. l = 250	-	0.7	-	-	coated	-	-	-
[36]	64,000	-	-	-	8	0.23	480	150	-	-	-	-

Table 2. *Cont.*

Ref.	Heat Transfer Surface [cm^2]	Adsorbent Mass [g]	S/m Ratio [cm^2/g]	Dimensions [3] [mm]	Fin Pitch [mm]	Fin Thickness [mm]	Fin Height [mm]	Fin Width [mm]	Bed Type	SCP [W/kg]	COP [-]	Ratio of HEX Mass to Bed Mass [-]
[37]	-	-	-	-	1.8	0.115	264	100	coated	400	0.37	-
[38]	-	3300	-	-	1.8	-	274	116	coated/packed	318	-	0.633
[39]	-	5600	-	-	-	-	-	-	coated	208.2	0.24	-
[40]	-	-	-	-	1.8	0.12	12.5	10.8	packed	295.19	0.47	-
[41]	24,000	6700	3.58	d = 8.81. D = 9.53. l = 1500	-	-	-	-	coated/packed	-	-	0.460
[42]	126,000	9900	12.73	d = 8.92. l = 0.685	-	-	-	-	packed	-	0.587	0.484
[10]	-	-	-	d = 15.875	1	-	15	-	coated/packed	max 290 [1]. max 720 [2]	-	-
[43]	-	-	-	d = 8.81. D = 9.53	1.8	0.12	5.24	-	coated	max 700	max 0.65	-
[44]	28,000	2770	10.11	D = 38	2.54	0.2	-	-	coated	456	0.27	0.475
[45]	-	-	-	d = 10. D = 12. l = 500	-	Up to 1	Up to 15	475	packed	Up to 590	Up to 0.64	-
[46]	12,100	2100	5.76	d = 18. l = 1000	8	1	16	-	packed	68	0.53	-
[47]	16,600	400	41.50	-					packed	318	0.176	0.614
[48]	1700	1750	0.97	d = 16. l = 560					coated	150–200	0.15–0.30	0.777
[30]	-	-	2.29	-		0.5–4	5–40	-	packed	150–550	-	0.466

[1] values for packed beds. [2] values for coated beds. [3] d = inner diameter; D = outer diameter; l = tube length.

2.1. Plate Exchangers

The main components of plate heat exchangers are serially connected plates. The plates are usually made by embossing a wavy pattern on a sheet of metal. The embossed pattern has two basic functions. Firstly, when plates are connected with bolts or by welding, the embossed patterns form fluid channels. Secondly, they are used to increase the exchange surface area. Hong et al. [31] proposed embossments on exchanger plates in the form of smaller and larger circles (Figure 4). They analyzed the effect of geometrical parameters, such as plate thickness, shape of HTF channels, and their height. They observed that the system performance strongly depends on the embossing pattern. It was found that with larger embossing diameters and heights, improved heat and mass transfer is obtained. A similar correlation was observed for smaller plate thicknesses where the conduction resistance is reduced. Another aspect analyzed was the HTF flow. For larger embossment spacing, smaller heights, and a lower ratio of small and large embossment diameters, the HTF flow is less turbulent, which negatively affects the heat transfer. The effect of channel geometry was also analyzed by Palomba et al. [32]. As part of a simulation study, the flow in the HTF channels was investigated for two proposed exchanger plate concepts with parallel channels (Figure 5a) and channels forming a mirrored double serpentine (Figure 5b). The simulation results show that temperature distribution is very uneven in both longitudinal and transverse direction for the parallel channels compared with the other concept. The temperature difference for parallel channels is 5 K and for serpentine-arranged channels it is 2 K. This shows that the use of the serpentine is more beneficial than the parallel channels, which are simpler to make.

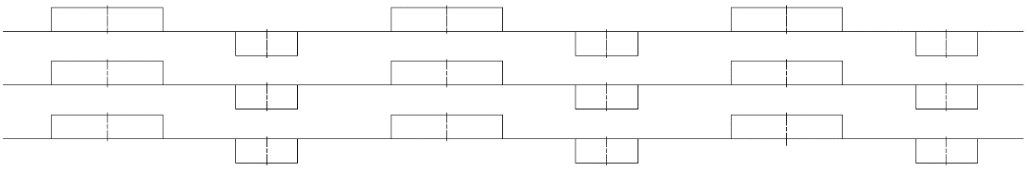

Figure 4. Plate HEX type with embossed patterns.

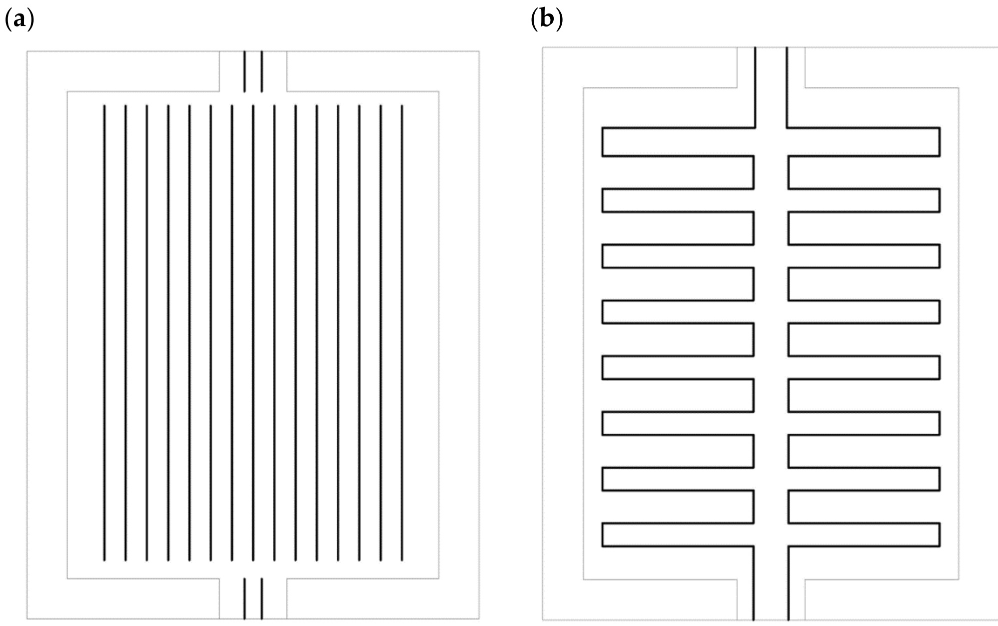

Figure 5. Types of HTF channels geometry: (**a**) parallel; (**b**) mirrored double serpentine.

The advantages of plate heat exchangers in the context of application in an adsorption bed include the fact that the bed is relatively easy to make. The spaces between pairs of connected plates are filled with adsorbent to form layers. The thickness of the adsorbent material layer affects the heat transfer from the HEX plate to the adsorbent. The greater the thickness of the bed, the lower the SCP values obtained, but at the same period the COP value increases due to increasing the total mass of the adsorbent [31]. It is also possible to use coated plates, which was proposed by Palomba et al. [32] in their investigation. The design presented is based on plates made of graphite with an adsorbent coating applied. In comparison with plate exchanger configurations with packed bed, this solution firstly provides lower mass flow resistance of the cooling medium vapor and lower contact thermal resistance between the HEX plate and adsorbent particles. In this context, it should be noted that the design of plate heat exchangers is characterized by a large diaphragm area, which gives better heat conduction directly from the liquid to the adsorbent in comparison with fin-tube exchangers [8,31]. On the other hand, plate HEXes are characterized by a weaker mass flow relative to tube-based heat exchanger.

Plate HEXes have a compact geometry and relatively low mass. This allows to achieve better heat transfer and reduced losses during pre-heating and pre-cooling of the bed [31]. When using polymers for the plate material, the problem of excess mass usually arises. Due to the fact that it is not possible to join such plates by welding, brazing, or soldering as in the case of metals, it is necessary to use seals and screw connections, which causes an undesired increase in the mass of the passive components. The issue of minimizing the mass of the connecting components was addressed by Palomba et al. [32] and Sapienza et al. [49]. Sapienza proposed a design of a plate HEX made by 3D printing using PLA, ABS, and nylon. The geometry of the plates allowed the use of connecting screws only at the top and bottom of the plate. Palomba [32], on the other hand, used an adhesive to bond pairs of plates. She tested two kinds of adhesives, i.e., silicone and epoxy resin. After adequate curing of the joints, the samples were tested for strength and sealing properties. The epoxy resin exhibited better sealing properties compared with silicone. No leakage was observed at two bars. However, in the case of the silicone joint, leakage already occurred at 100 mbar.

2.2. Plate-Fin Heat Exchangers

A further development of the concept described above is the plate-fin heat exchanger. The design of this type of heat exchangers is based on two components. The first of them is a rectangular plate in which there are HTF channels. The second component is a fin set, which has the function to expand the heat exchange surface between the exchanger and the adsorbent. The fin set is made by bending sheet metal. The fins are connected with the plates by brazing or welding. The sheet forming the fins usually forms a pattern in the form of an array of triangles in cross-section. The basic dimensions of the fins included in this analysis are shown in Figure 6.

Abdel et al. [50] investigated the effect of the use of fins and their shape on heat transfer in an adsorption bed with a plate-fin exchanger. Four cases were investigated, i.e., zigzag fins, flat fins, bed with added aluminum slices (20 mm × 6 mm × 2 mm), and HEX without fins as a reference case. After activated carbon was deposited, they were tested in an Adsorption Refrigeration Tube (ART). From the results it was found that it was the zigzag fin that provided the highest increase in COP (27%). An increase was also observed for flat fins (8%) and aluminum pieces (17%). On the other hand, the highest increase in SCP (the authors used the term specific cooling effect (SCE)) was recorded for aluminum slices (203%). For zigzag fins, an increase in SCP value was also obtained (45%). However, for flat fins, a slight decrease in SCP values was recorded (10%). Mitra et al. [51] also studied the effect of the shape of plate-fin HEX fins on heat and mass transfer. They studied three cases of domains in a simulation study where each domain corresponded to a single interfin space. The domains differed in pitch (p_F) and fin height (h_F), but had a constant heat transfer surface area. Each case was considered for two adsorbent particle sizes. From the results, it was found that for particles of a given size, there is an optimal geometry described by an Aspect Ratio, i.e., the ratio of the height of the fins to their pitch. Furthermore, the researchers found that for larger particles, a higher fin height is preferable. On the other hand, for smaller particle sizes, a larger fin pitch is recommended.

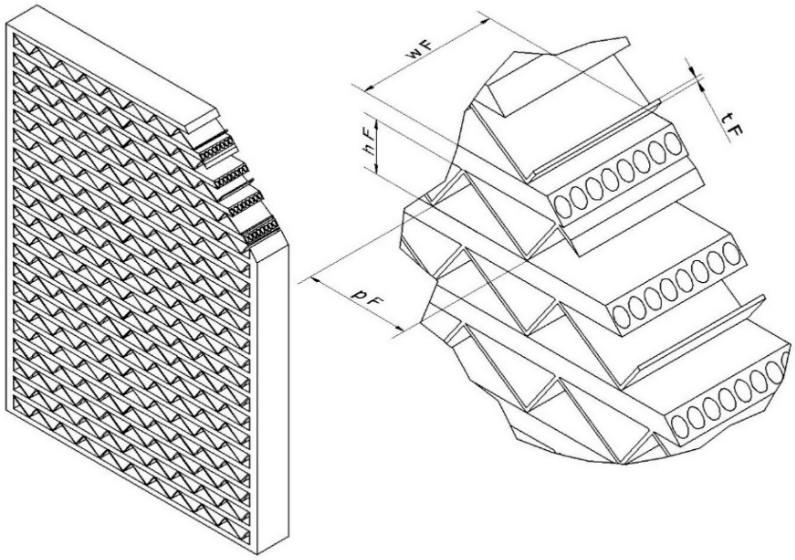

Figure 6. Basic dimensions of plate-fin HEX: h_F—fin height; p_F—fin pitch; t_F—fin thickness; w_F—fin width.

Another key parameter in the analysis of plate-fin HEX geometry is the dimensions of the fins, i.e., height, width, thickness, and pitch. An example of fin geometry along with the basic HEX dimensions is shown in Figure 6. Sapienza et al. [29] studied the effect of fin width (w_F) on mass transfer. The aim of the investigation was to optimise the HEX geometry with respect to the resistance of the intergranular mass transfer of the cooling medium. From the results it was found that with a constant S/m ratio but variable fin width, the sorption dynamics are variable. With a larger fin width, a slowing down of the sorption phenomenon was observed due to the longer route that the molecules of the adsorbate vapor have to travel in the bed. The effect of bed thickness is particularly significant when using adsorbent with smaller grains, where the intermolecular mass transfer resistance is higher. On the other hand, Brancato et al. [8] noted that with larger grains the intramolecular mass transfer resistances become important, but these are not affected by the HEX geometry. The problem of intergranular mass transfer was also addressed by Mohammadzadeh Kowsari et al. [24]. They found that mass transfer resistance is solely dependent on the bed thickness. However, it should be noted that the larger the contact area between the adsorbent and the interior of the adsorption chamber, the better the course of the processes of adsorption and desorption, and thus the mass transport is also affected by the height and pitch of the fins. In the context of the effect of fin geometry on heat transfer in the bed, the researchers found that heat transfer resistance depends mainly on the pitch and height of the fins. For higher fins, higher COP values were obtained, i.e., 0.642 and 0.623 for a height of 20 and 14 mm, respectively. When the fin pitch was changed, performance improvement was also observed for increasing distance between the fins. For SCP values, the trend is the opposite, i.e., higher values are achieved for smaller bed sizes at the expense of a lower COP value. From the point of view of heat transfer, and consequently even temperature distribution in the bed, it is more beneficial to use smaller fin pitches since, as indicated by Verde et al. [52], thermal conductivity decreases drastically when pitch is increased. When pitch is decreased, the number of fins increases, which allows to minimize the proportion of dead volume in the bed. Therefore, a larger number of smaller exchangers are recommended [27,30]. Rogala [53] points out that the adsorbent to metal mass ratio (M_{ads}/M_{met}) and the HEX exchange surface area to adsorbent mass ratio (S/m), related to COP and SCP, respectively, reach their maxima for different conditions. The M_{ads}/M_{met} ratio increases with increasing pitch, while the S/m ratio reaches its highest value with tight fin spacing. For this reason, in order to achieve high performance of an adsorption chiller, it is necessary to optimize the dimensions, something that Rogala did. The optimization carried out made it possible to improve the COP by 3.7% and the SCP by 6.3%. The last of the geometrical parameters of the fins under consideration is the thickness. Fin thickness affects the performance of an adsorption chiller in two ways. Firstly, as observed by Papakokkinos et al. [30], increasing thickness causes an increase in the solid volume fraction (SVF), i.e., the ratio of the HEX mass to the total mass of the bed. The increase in SVF, in turn, also causes a drastic increase in the SCP value but with a simultaneous decrease in the COP. This is due to an increase in the share of the passive mass of the exchanger in the bed, and thus an increase in the thermal capacity of the system.

Among the studies analyzed, the issue of HTF channels in plate-fin exchangers was addressed by Verde et al. [52], and in the context of the introduction of microchannel exchanger applications by Paul et al. [25] and Pahinkar et al. [28]. In a study conducted to optimize the exchanger geometry, Verde found that a decrease in the water to adsorbent mass ratio improves the COP and SCP. This is related to the thermal inertia of the system; the lower the water mass in the exchanger, the lower the energy loss for heating/cooling the heat transfer fluid (HTF). Hence, the authors recommend minimizing the amount of water in the exchanger. One of the methods to reduce the water mass can be increasing the number of channels at the expense of reducing their pitch and dimensions. Such an action will additionally facilitate heat transfer by convection because, as Verde points out, the flow velocity of the HTF increases with a reduction in channel size. Furthermore, due to the poor conductivity of adsorbent, reduction of the channel pitch significantly reduces the mean heat transfer resistance of the bed. An extension of the outlined direction of HEX geometry modification on the HTF side is designs based on the concept of microchannels. The study by Paul et al. [25] suggests that the application of a microchannel heat exchanger allows to reduce the cooling time by about 10% in comparison with the classical fin-tube heat exchanger design. The improved heat transfer is due to a significant increase in the flow rate of the HTF. The authors also point out the problem of heat transfer limitation caused by the reduction of the distance between the inlet and the area of full flow development. This is because microchannel heat exchangers use shorter HTF channels to reduce flow resistance. Therefore, in order to achieve a further reduction in the proportion of cooling period in the total cycle time of an adsorption chiller, it is necessary to propose a microchannel plate design where a larger part of the flow would be developed. The design described above was based on the classical concept of a plate-fin HEX with a series of plates connected alternately with a layer of fins. Another step in the development of microchannel heat exchangers was proposed by Pahinkar et al. [28], where both the process side (adsorption/desorption) and the HTF side are based on microchannel geometry. The authors proposed an exchanger design where successive plates prepared, e.g., by photochromatic technology, are connected to each other to form a series of microchannels for the HFT and for the refrigerant vapor. The method of deposition of adsorbent particles that was chosen was coating technology, which will further improve thermal conductivity and thus the SCP. The use of a microchannel heat exchanger allowed an increase in the COP value by 77% and a twelve-fold increase in the SCP value. It should be noted, however, that the presented concept of the bed and exchanger has not been translated into practice, but only tested in simulation studies. Moreover, the investigation presented did not take into account the flow resistance that can potentially be relatively high when using such a HEX geometry.

In terms of adsorption bed design, plate-fin HEXes enable easy deposition of adsorbent particles in the free spaces formed by the fins, but an additional mesh is required to prevent adsorbent displacement. First of all, this mesh increases the mass of the passive components, increasing the thermal inertia of the system and additionally causes an increase in the mass transfer resistance [8]. Due to the high heat transfer surface area, plate-fin HEXes have great potential for use with bonded beds [28,29].

2.3. Fin-Tube Heat Exchangers

Another frequently used heat exchanger design in adsorption beds is the fin-tube heat exchanger. This concept uses the method of increasing the heat surface area referred to in Section 2.2, i.e., adding fins to the components that function as HTF channels. The fin arrangement can take various forms. The most common are rings attached perpendicularly to the HTF tubes. Such designs are analyzed in terms of the effect of the fin geometry on heat and mass transfer. Apart from the aforementioned classical design, fins can be wavy, oblique, or spiral, and an extension of this concept is the exchanger proposed by White [35]. In the HEX mentioned before the function of the fin was performed by a spiral made of wire, which was then also spirally wound on the HTF tube. This design allowed to increase the heat transfer surface area while reducing the weight of the exchanger. Moreover, the presented fin design allows increasing the amount of adsorbent particles that are in direct contact with the inner space of the adsorption chamber, which altogether intensifies the heat and mass transport increasing the efficiency of the chiller. However, it should be noted that the proposed design requires the use of an additional mesh to support the adsorbent particles. Additionally, it is quite difficult to make proper connections at the interface between the fins and HTF channels. Thus, White's proposal in its present form does not qualify for wider application. Examples of finned-tube HEX geometries are shown in Figure 7.

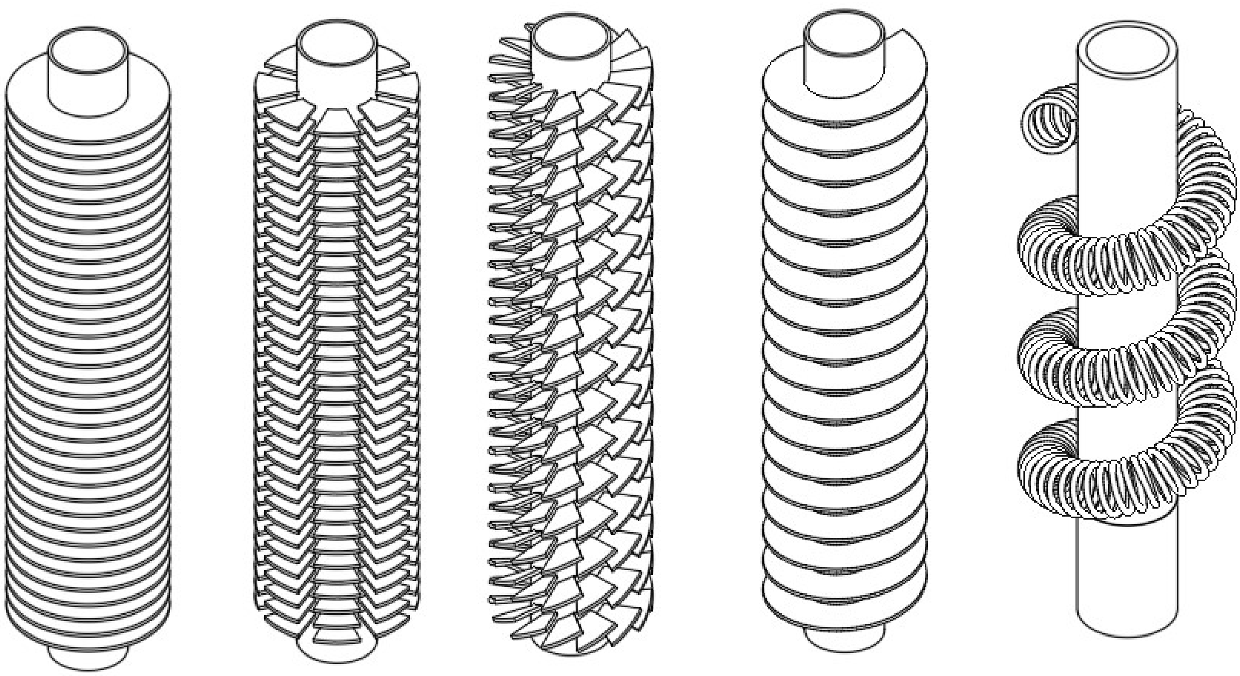

Figure 7. Types of fin geometry used in fin-tube HEX.

As in the plate-fin HEXes described in Section 2.2, the geometry of the fins has a significant effect on heat and mass transfer. The basic dimensions of the fin-tube HEX included in this analysis are shown in Figure 8. Although heat transfer through metal fins is less efficient than through HTF tubes, as noted by Brancato [8], the use of fins is necessary for higher adsorbent mass to provide the required heating or cooling of the bed during the cycle. In addition, if the proportion of HTF tubes were to be increased while discarding fins, it would not be possible to achieve an adequate heat transfer surface and the thermal inertia of the bed would be compromised. In comparison with plate-fin HEXes, in the case of fin-tube HEXes we are dealing with thicker adsorbent layers, which involves an increase in the temperature gradient in the bed. The effect of the use of fins in tubular HEXes on the temperature gradient was investigated by Zhao et al. [54]. The authors found that the use of fins reduces the temperature gradient in the bed, which is 13.8–21 °C and 3.8–4.1 °C for tubular HEXes and fin-tube HEXes, respectively. Therefore, it should be concluded that the use of fins allows a more even temperature distribution in the bed. Sharafian et al. [34] studied the effect of fin pitch on temperature distribution. It was observed that the adsorbent particles experience different temperatures in different parts of the bed. For particles located in the center of the fins, i.e., at a distance from the HTF pipe heat transfer takes place mainly by transfer through the fin. The temperature of particles at the edge of the fins is additionally influenced by the temperature of the adsorbate. Additionally, the adsorbent particles that are closest to the HTF pipe sooner reach the right temperature. Thus, the greatest temperature gradient is observed in the middle of the fins. Hence it follows that it would be beneficial to decrease the radius (height) of the fins. Additionally, Elsheniti et al. [55] noted that higher SCP values are obtained for smaller fin heights due to an increase in the bed heating rate. Regarding the effect of fin height on the COP value, opinions are divided. On the one hand, lower fins reduce the mass of the heat exchanger, thus limiting the losses associated with the heating and cooling of the passive components [55]. On the other hand, given that the fin height determines the adsorbent layer thickness, decreasing the fin height decreases the adsorbent mass and thus the adsorption uptake, i.e., the useful energy of evaporation is also lower. Furthermore, Bahrehmand et al. [56] found that for higher fin height the COP increases because the ratio of the thermal inertia of the metal to the thermal inertia of the adsorbent decreases and the energy of evaporation increases more than the energy supplied to heat a larger mass of the HEX. Another important geometrical parameter is the fin pitch. Sharafian [34] found that the effect of fin pitch is greater for shorter cycle times than for longer ones. This is due to the fact that for shorter cycle periods of the adsorption chiller, rapid heating and cooling of the bed becomes more important, which is indirectly related to the SCP value, and the fin pitch affects this parameter to a greater extent compared with the COP value. As noted by Abd-Elhady et al. [9], when reducing the fin pitch by 50%, the SCP value (the authors use the term SCC) increases by approx. 147% and the COP value by approx. 21%. Reducing the pitch is firstly associated with an increase in the number of fins, which increases the heat transfer surface area, thus increasing the SCP value [55]. Secondly, it reduces the average distance between the adsorbent particles and the HEX components, decreasing the heat conduction resistance of the bed. The last of the geometrical parameters of the fins under consideration is their thickness. Papakokkinos [30] states that the thickness of the fins should be as small as possible. As the fin thickness increases, the HEX mass increases, which in turn increases the thermal inertia thus lowering the COP [56]. A 50% reduction in fin thickness results in an increase in the COP of approx. 9%. In applications where a bed with higher adsorbent mass is required, it is more advantageous to dispense with one HEX with a large fin radius in favor of several smaller HEXes. Duong et al. [40] investigated the effect of a configuration of several HEXes on the performance of the adsorption system. In their simulations they investigated parallel, series, and series-parallel connections. For the serial case, the problem is the non-uniform temperature distribution in the bed as the temperature of the HTF decreases along with the flow to subsequent exchangers. For the parallel connection, the flow through the individual HEXes is limited as the input HTF mass flow is distributed to the subsequent HEXes. Therefore, it will be most advantageous to combine the two concepts in a series-parallel arrangement where, on the one hand, the flow is split into fewer branches, and on the other hand, the HTF route is shortened, which makes it possible to make the temperature distribution in the bed more even.

The problem concerning fin-tube HEXes is adsorbent deposition. In each case when a packed bed is used, it is necessary to use an additional mesh to hold the adsorbent particles. This implies increasing the mass of the passive components. Additionally, the mesh restricts the mass flow of the refrigerant vapor [8]. As in the case of plate-fin HEXes, it is possible to use a coated bed, but due to the smaller heat transfer surface area, such a solution is not equally effective. Tables 1 and 2 present a summary of the most important parameters of the heat exchangers being analyzed.

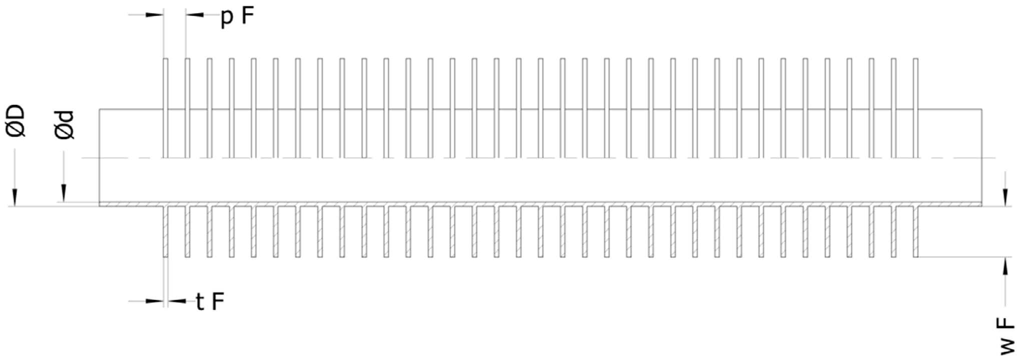

Figure 8. Basic dimensions of fin-tube HEX.

Based on the data summarized in Tables 1 and 2, it should be concluded that several times higher values of the S/m factor can be obtained for plate and plate-fin heat exchangers relative to tube heat exchangers. On the other hand, when analyzing the ratio of the exchanger mass to the mass of the bed, it should be stated that this ratio is in the range of 0.5 to 0.7 for plate heat exchangers, but for tube heat exchangers these values are lower, being in the range of 0.4 to 0.6. This fact, however, does not influence the values of the SCP and COP parameters of exchangers, as higher SCP values were obtained for plate heat exchangers in comparison with tube HEXes. It should also be noted that consolidated beds are more frequently used in combination with finned-tube exchangers due to the simple application of coatings and the analysis of coating thickness and quality compared with plate exchangers. In addition, the use of consolidated or coated beds in this type of HEX (fin-tube) allows the omission of a mesh to securely contain adsorbent particles in appropriate position.

2.4. Heat Pipes

The last of the HEX types used in adsorption beds and included in this analysis, are heat pipes exchangers. Due to the use of phase transition heat and the convection phenomenon, these structures are characterized by high heat transfer intensity. The heat pipe design is usually based on a copper pipe with one end in the high temperature area and the other end in the low temperature area. The pipe is partially filled with an appropriately selected liquid (e.g., liquefied ammonia, acetone, water, ether, Freon, methanol, or ethanol). The inside of the tube is often coated with a porous material that acts as a wick. The heat transfer process takes place in the following way. First, heat is removed from the high-temperature area through evaporation and heating of the medium vapor. The heated vapor then travels to the low-temperature area where it cools down and condenses as a result of giving off the heat. Subsequently, the liquid trickles down the walls by gravity or seeps through the wick, i.e., the porous layer on the tube wall.

Compared with the previously mentioned HEX designs, the main advantage of heat pipes is the possibility of application in chemically aggressive environments. Such an application of heat pipes has been described by Wang [18]. An adsorption chiller mounted on a fishing boat used exhaust fumes as a heat source in the desorption process. At the same time, seawater was circulated to cool the bed. The use of heat pipes minimized corrosion of the heat exchanger components. Thanks to the high efficiency of the heat pipes and the use of a composite adsorbent ($CaCl_2$ and AC), a high SCP value of 730 W/kg was obtained at a refrigerant evaporation temperature of −15 °C. A similar solution has been presented by Wang et al. [57] where a split heat pipe and a thermosiphon heat pipe were used. For the split heat pipe, the evaporation and condensation zones are separated. The condensation zone is a section of the pipe placed in an adsorption bed. The evaporation zone is made as an exhaust fumes/water exchanger. Another advantage of heat pipes over classical solutions is the less complex design of the system. This makes it possible to reduce the number of components, and thereby improve the reliability of the system, and also to reduce the weight of the device.

The main problem with adsorption systems using heat pipes is the phase switching between the processes of bed desorption and adsorption [58]. In the course of desorption, the bed operates together with the condensation zone, and during adsorption with the evaporation zone. Hence, it is necessary to adapt the design of the adsorption chiller accordingly to take advantage of the free fluid flow in the heat pipe.

In addition to the above application of heat pipes as a HEX in chemically aggressive environments, this solution is also used in heat recovery systems. Such an application has been described by Li et al. [15]. In the two-bed system presented, after the desorption process the high-temperature bed is responsible for providing heat to the evaporation zone of the heat pipe. On the other hand, the low-temperature bed is responsible for receiving heat from the heat pipe condensation zone after the adsorption process. The use of heat pipes allowed efficient heat exchange with a relatively small increase in the mass of the passive elements.

3. Evaluation of the Heat Capacity of an Adsorption Bed

The design aspects of the heat exchangers described in Chapter 1, such as the heat transfer surface and the design of the liquid channels, have a significant impact on the SCP and COP parameters of the adsorption chiller. Nevertheless, the geometric parameters of heat exchangers are related to the properties of the materials used for their construction. With regard to the material from which a particular heat exchanger (HEX) is made, a number of factors must be taken into account. It is necessary to consider the physical and chemical parameters of the working environment of the exchanger. The ones that are essential for exchangers in adsorption beds include the corrosiveness of the HTF, which is related to the problem of corrosion, and the physical parameters of the process, i.e., mainly the operating pressure and temperature. The main properties relevant to heat exchanger materials are strength and formability (material selection) and thermal capacity and conductivity (thermal performance). Therefore, the following steps, shown in Figure 9, must be considered when designing a HEX.

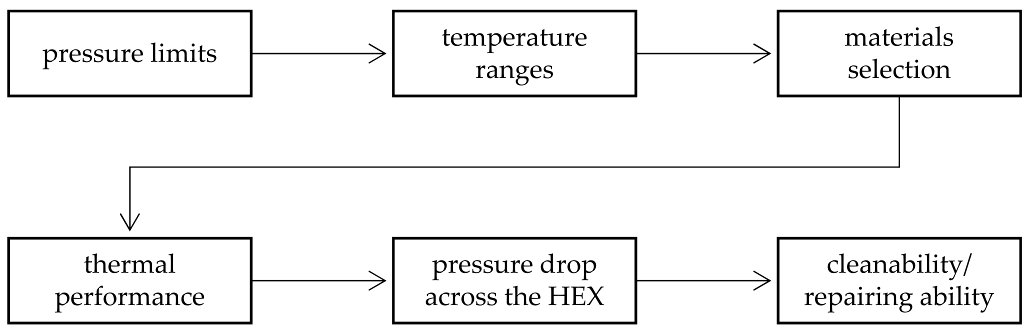

Figure 9. HEX design.

The durability of the exchanger for given pressure and temperature ranges is related to the choice of the working pair for the adsorption chiller. The correct choice of the pair enables the selection of the materials used in the construction of the HEX. Apart from selecting the right material, it is essential to evaluate the heat capacity of the exchanger, the pressure drop and the susceptibility of the exchanger to dirt contamination.

The basic materials used in the construction of HEXes are aluminum, copper, and steel. Alternative materials include graphite, nylon, acrylonitrile butadiene styrene (ABS), and polylactic acid (PLA). Table 3 shows the basic properties of the materials used to build HEXes.

Table 3. Properties of heat exchanger materials. Own elaboration based on: [59].

Material	Thermal Conductivity [1] [W/(m·K)]	Specific Heat [J/(kg·K)]	Coefficient of Thermal Expansion [10^6/K]	Melting Point [°C]	Corrosion Resistance	Possibility of Contamination
Copper	390	385	16.5	1084	high, passive layer	low
Steel	50	490	10.8–12.5	1500	low	low
Aluminum	240	897	21–24	660	high, passive layer	medium
Graphite	168	717	4–8	3600	very high	high
Nylon	0.25	1600	50–90	270	very high	medium
ABS	0.22	1600–2100	72–108	200	very high	low
PLA	0.25	1200–1800	68	130–180	very high	medium

[1] values are given at 25 °C.

The material for an exchanger used in an adsorption bed should be characterized by high thermal conductivity, where according to Verde et al. [52], the limit value of thermal conductivity is 140 W/(m·K) and its further increase does not significantly affect the performance of the chiller. In this respect, the best materials are copper, aluminum, and graphite, which are most commonly used as HEX materials due to their thermal conductivity and economic aspects. An interesting aspect concerning exchangers built from these materials is reducing the thermal resistance at the interface between the adsorbent material and the heat exchanger because the modern design solutions are highly developed, and further development in this field is difficult. Another thermophysical parameter is the specific heat of the material that affects the heat exchanger capacity. This characteristic of the exchanger directly affects the pre-cooling and pre-heating times and thus also the performance of the chiller. The other important parameters of heat exchanger materials are the coefficient of thermal expansion and the melting point. The former allows evaluation of the elongation of the individual HEX parts due to alternate heating and cooling of the bed. Melting point makes it possible to determine the maximum operating temperatures of the HEX. Teng et al. [60] determined that a heat exchanger in a sorbent bed must be characterized by very good heat transfer and low mass. These authors use two parameters to evaluate exchangers, i.e., fluid alpha number and inert material alpha number. The former is defined as the ratio of the HTF heat capacity to the adsorbent heat capacity. The second parameter is defined as the ratio of the heat capacity of the passive part of the bed to the heat capacity of the adsorbent. On this basis, it should be concluded that the mass of the adsorbent in the bed, the mass of the heat exchanger itself, and the amount of HTF should be taken into account when developing an indicator for evaluating a heat exchanger in an adsorption bed. Nevertheless, Verde et al. [52] found that the working fluid of the HEX does not have a significant effect on the performance of the system. The case is different when it comes to the weight of the system. Chiller performance increases when the mass of metal in the bed decreases because the proportion of inert mass in the bed decreases and the heat recovery cycle can be dispensed with in favor of the basic adsorption cycle. Therefore, the current trend in technology is to reduce the heat capacity of HEXes used in adsorption beds.

Kowsari et al. [24] dealt with heat distribution (allocation) in an adsorption chiller. The authors listed five main aspects responsible for the efficiency of heat utilization by the chiller:

- Properties of a heat transfer fluid;
- Design of liquid channels of the heat exchanger;
- The use of fins in the heat exchanger;
- Bed design/housing;
- Amount of latent heat of sorption.

With regard to the performance of the device, it is desirable for the proportion of heat used for desorption of adsorbate from the adsorbent bed to be as high as possible. The optimal use of heat can be achieved by optimizing the geometry. An increase in fin height has a very good effect on the heat distribution in the chiller system. For a fin height of 3 mm, approx. 50% of the heat is used for the desorption process; and for a fin height of 20 mm, almost 75% of the heat is used for desorption. Similarly, increasing fin pitch from 3 to 21 mm increases the amount of heat used in the desorption process from 51 to 57%. With both changes made at the same time, the effect of optimizing heat utilization in the chiller is enhanced because almost 80% of the heat supplied to the chiller is consumed in the desorption process. Therefore, the design of a heat exchanger must take into account the effect on the temperature distribution. As in the case analyzed, decreasing the number of fins with a simultaneous increase in their height generates better heating of the adsorbent [24].

The adsorption bed is alternately heated and cooled, which makes it necessary to analyze the heat capacity of the heat exchanger. Taking into account heat and mass transfer in the adsorption bed, it is preferable for the heat exchanger to have as thin walls and low heat capacity as possible. However, where beds operate in overpressure conditions (e.g., with ammonia as adsorbate), thicker walls are necessary, which increases the capacity of the system. It should also be noted that, e.g., for steel the resilience decreases with increasing temperature, which can also be compensated for by a thicker wall. Likewise, when the adsorbent has a low density and a low filling level in the bed, the system must be increased in size in order to achieve sufficient power, which also increases the thermal capacity.

When analyzing the data in Table 4, it should be stated that the material most frequently used for the construction of HEXes used in adsorption beds is aluminum. Based on the mass ratio of the exchanger to the mass of the whole bed, it should be stated that the mass of the exchanger accounts on average for 40 to 80% of the bed mass. Based on the data from Table 3, it can be observed that the COP of the system decreases with an increasing share of the exchanger mass in the total mass of the bed. This is due to the increasing heat demand. As the mass of the exchanger increases, its heat capacity increases, and thus less heat is directed to the desorption process. On the other hand, it is impossible to unambiguously determine the effect of the exchanger mass share in the bed mass on the value of the SCP. Nevertheless, it should be stated that it is advantageous to reduce the mass of the exchanger in order to decrease the inertia of the system and reduce the mass of the chiller, which is important with regard to spreading the use of adsorption refrigeration equipment.

Table 4. Material parameters of heat exchangers in adsorption beds.

Ref.	HEX Mass [g]	Adsorbent Mass [g]	HEX Material	Ratio of HEX Mass to Bed Mass [-]	Heat Capacity [kJ/K]	Cooling Power [kW]	SCP [W/kg]	COP [-]	Adsorbent	Adsorbate
[61]	636	400	aluminum	0.614	0.566	0.127	318	0.176	Zeolite	water
[62]	6080	1750	aluminum	0.777	5.41	0.338	150–200	0.15–0.30	Zeolite	water
[32]	478	500	graphite	0.489	0.378	0.5–1	1000–2000	-	Zeolite	water
[24]	8740	5990	aluminum	0.593	7.840	8.02	835	0.623	SWS-1L	water
[24]	21,840	4510	aluminum	0.829	19.590	11.14	1108	0.449	SWS-1L	water
[24]	7810	5990	aluminum	0.566	7.006	7.84	835	0.638	SWS-1L	water
[24]	8420	7030	aluminum	0.545	7.553	7.79	711	0.642	SWS-1L	water
[24]	8920	5680	aluminum	0.611	8.001	8.07	880	0.619	SWS-1L	water
[34]	716	820	copper	0.466	0.279	-	-		Silica gel	water
[34]	978	770	copper	0.559	0.381	-	-		Silica gel	water
[25]	271.8	410	aluminum 3003	0.399	0.244	-	51.6	0.5	Silica gel	water
[8]	160	49.86	aluminum	0.762	0.144	-	3320	-	SRD 1352/3	ethanol
[8]	130	43.41	aluminum	0.750	0.117	-	1360	-	SRD 1352/4	ethanol
[8]	140	46.05	aluminum	0.752	0.126	-	170	-	SRD 1352/6	ethanol
[41]	5700	6700	aluminum	0.460	5.113	4–6	-	-	Zeolite + Silica gel	water
[42]	9300	9900	aluminum + copper	0.484	6.29	3.62	-	0.587	Silica gel	water
[44]	2510	2770	aluminum	0.475	2.251	3.31	456	0.27	FAM Z02	water
[33]	8500	3300	aluminum	0.720	8.747	5.56	1680	0.4	SAPO 34	water
[29]	4130	2230	aluminum	0.649	3.70	3.4–4.4	1520–1970	0.3–0.35	AQSOA FAM Z02	water

Paul et al. [25] proposed modifications of metal heat exchangers to reduce their heat capacity. In the publication under review, the authors compared a classical tubular heat exchanger with a new exchanger based on microchannels. The new solution enables a better distribution of heat over the bed as well as a reduction of the weight of the heat exchanger by almost 20% in relation to tubular exchangers. In addition, the ratio of the mass of the metal to the mass of the sorbent has also been reduced by approx. 65%. However, the heat exchanger with microchannels under analysis has not been investigated in terms of its effect on the SCP and COP, and the authors only predict that it will enable an improvement of these parameters by 24 and 4%, respectively. A similar solution has been proposed by Palomba et al. [32] who built their bed as several assembled graphite plates, between which there is a flow path for the heat transfer fluid (HTF). Each graphite plate consists of a process side onto which zeolite is deposited and a HTF side. This solution makes it possible to form substrates for adsorbent deposition, which allows to significantly improve heat and mass transfer while reducing the mass of the HEX. The authors optimized the thermal mass of the HEX by placing connecting bolts only in the upper and lower part of the plate instead of bulky and heavy components, such as the clamping plates usually used in conventional plate heat exchangers. It is worth noting that the graphite of which the HEX in question is made cannot be brazed or welded so the bonding has to be obtained by adhesives. The use of adhesives additionally reduces the mass of the system, but also its volume, which is important with regard to reducing the heat capacity of the HEX and the dimensions of the adsorption chiller. Palomba et al. [32] bonded graphite plates using silicone and epoxy resin. A pressure test of the new exchanger revealed a separation of the HEX plates bonded with silicone and good leakage resistance for bonding with epoxy resin at a target pressure of two bars. However, one more aspect of the new exchanger design should be noted, i.e., the flow resistance. Replacing liquid pipes with a network of microchannels will enable a reduction in the thermal capacity of the system, but may generate significant flow resistance, which will translate into an increase in energy consumption to drive the circulation pump of the HTF system. Wittstadt et al. [33] drew attention to another aspect of reducing heat capacity of a HEX, which is the option to simplify the adsorption cycle. The low mass of the heat exchanger makes it possible to dispense with the heat recovery cycle. In their study, the authors analyzed an exchanger built by brazing aluminum-sintered metal fiber structures on flat liquid channels. However, this system is characterized by a high share of the exchanger mass in the bed mass and thus a low COP, which is 0.4. On the other hand, other systems with a HEX of similar mass [24] shown in Table 3 are characterized by a 50% lower SCP relative to the study by Wittstadt's team.

Another way to reduce the heat capacity of heat exchangers in adsorption beds is to use thermally conductive polymers. Sapienza et al. [49] analyzed the possibility of using 3D printing to create exchangers as a competitive solution to metal exchangers. The authors tested the following HEX materials: ABS, Nylon and PLA, which are used as filaments in 3D printing. Flat HEXes constructed from polymers were compared with a flat aluminum exchanger of the same design. For the HEXes built from polymers, the ratio of the weight of the exchanger to the weight of the whole bed is between 0.27 and 0.31, which is much better than the best exchangers made from metals (0.4 and higher). The authors found that a polymer HEX has the same unit cooling capacity as a metal HEX. This means that the low thermal conductivity of polymers does not have any negative effect on the SCP of the chiller. On the other hand, it should be noted that the tests were carried out with an adsorbent monolayer, for very simple exchanger geometry, for which reason it seems that the effect of using polymer HEX on the COP and SCP of the chiller cannot be realistically assessed. It should also be taken into account that the thermophysical properties of the polymers analyzed allow their use in adsorption chillers regenerated with low-temperature heat (<90 °C). Additionally, these polymers have a very high heat capacity, which in combination with low thermal conductivity generates the problem of too long pre-cooling and pre-heating times. Therefore, modifications of the polymers are necessary to maintain the possibility of 3D printing while improving the thermophysical parameters of the HEX. Hinze et al. [63] analyzed the possibility of reducing the heat capacity of an exchanger by using polymers doped with graphite. The authors reduced the heat capacity of the HEX by 30% compared with an exchanger constructed from aluminum. The material of the new exchanger is PA6 (polyamide 6) doped with expanded graphite particles. The addition of graphite in an amount of 40% by weight allows a significant improvement in the thermal conductivity of the polymer from 0.25 to 6.9 W/(m·K). The exchanger was analyzed in the context of working with two sorption materials (SG and TAPSO-34), and an improvement in adsorption kinetics was observed for both of them, with increasing thermal conductivity of the exchanger material. On the other hand, the use of HEX made of pure PA6, which has poor thermal conductivity, results in a deterioration of the sorption kinetics relative to HEX made of aluminum. Such a heat exchanger doped with polymer also absorbs 30% less heat than an equivalent aluminum HEX. Such a reduction in heat absorption may enable an improvement of the COP of the chiller. However, it should be borne in mind that the studies carried out by Hinze's team [63] as well as Sapienza's team [49] concerned a simple plate heat exchanger consisting of only one plate. Therefore, it cannot be stated unequivocally that the observed effect will be scalable to include other, more complex HEX designs.

Polymers as alternative materials used for the construction of HEXes give the possibility to form various, very complex HEX geometries without using many joints that cause an undesirable increase in the mass of the system in metal exchangers. Nevertheless, attention must be paid to the working conditions of each HEX in the context of mechanical properties of printed exchangers. In addition, when considering any HEX design, one should keep in mind the option of using coatings, the way they are applied, and the mechanical strength of different coatings on different materials [64].

4. Discussion

The results of studies by various authors presented in this paper show a small variety of parameters of heat exchangers used in adsorption beds. Two types of exchangers predominate, i.e., the fin-plate and the fin-tube ones. For fin-tube HEXes the fin pitch is in the range of 2–10 mm and their maximum thickness is 4 mm. However, it is fins with a thickness of approx. 0.2 mm that predominate. For fin-plate HEXes, the fin pitch is in the range of 1–18 mm, with a maximum fin thickness of 5 mm. However, it should be noted that fins with a thickness of approx. 0.5 mm predominate. On the other hand, the ratio of the HEX mass to the bed mass for the exchangers under analysis is within a wide range of 0.4 to 0.8. Figures 10 and 11 show the correlation between the coefficient of performance (COP) and specific cooling power (SCP) values and the proportion of the HEX mass in the bed mass for finned, plate, and tube heat exchanger types.

When analyzing Figures 10 and 11, it should be stated that the optimum ratio of the HEX mass to the mass of the bed for fin-plate heat exchangers is in the range of 0.5 to 0.7. With such heat exchangers, it is possible to operate the chiller at the highest values of the COP and SCP parameters. For fin-tube heat exchangers, the optimum ratio of exchanger mass to bed mass is in the range of 0.45 to 0.6. These HEX designs enable the most efficient operation of the adsorption chiller. It should also be added that chillers with fin-plate exchangers can operate at higher COP and SCP values than those with fin-tube exchangers. Furthermore, it should be stated that the current state of the art does not allow further reduction of HEX mass share in the bed mass without a negative impact on the adsorption chiller performance.

When selecting an exchanger design and material, a multivariate analysis should be performed taking into account the complexity of the adsorption chiller system. Thermal expansion of the heat exchanger is very important in the context of the possibility of coating the exchanger and the risk of coating separation. The choice of exchanger material must take into account the operating conditions, i.e., operating pressure and temperature, but also the atmospheric corrosivity and the resistance of the material to alternate heating and cooling.

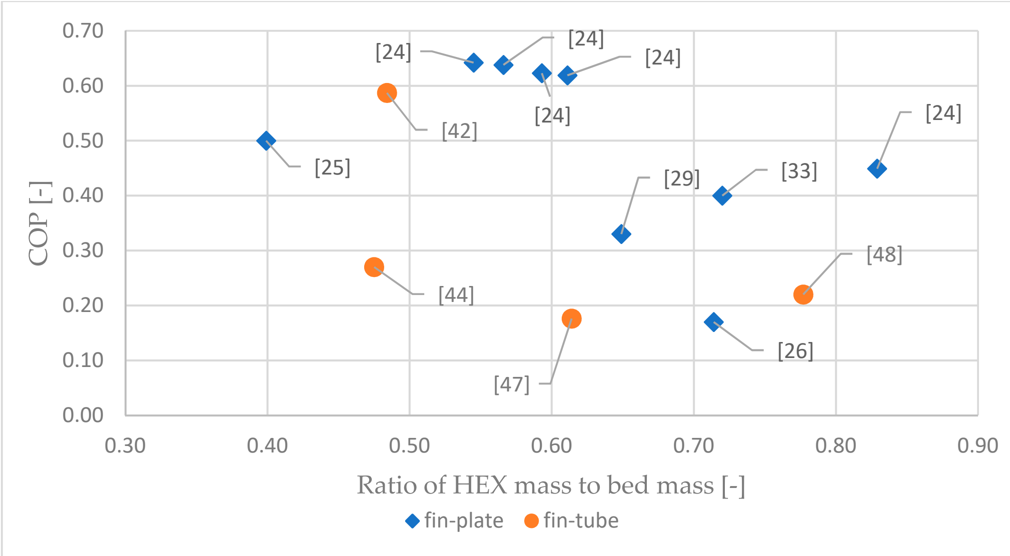

Figure 10. Correlation of COP with the mass ratio of the HEX in the bed [24–26,29,33,42,44,47,48].

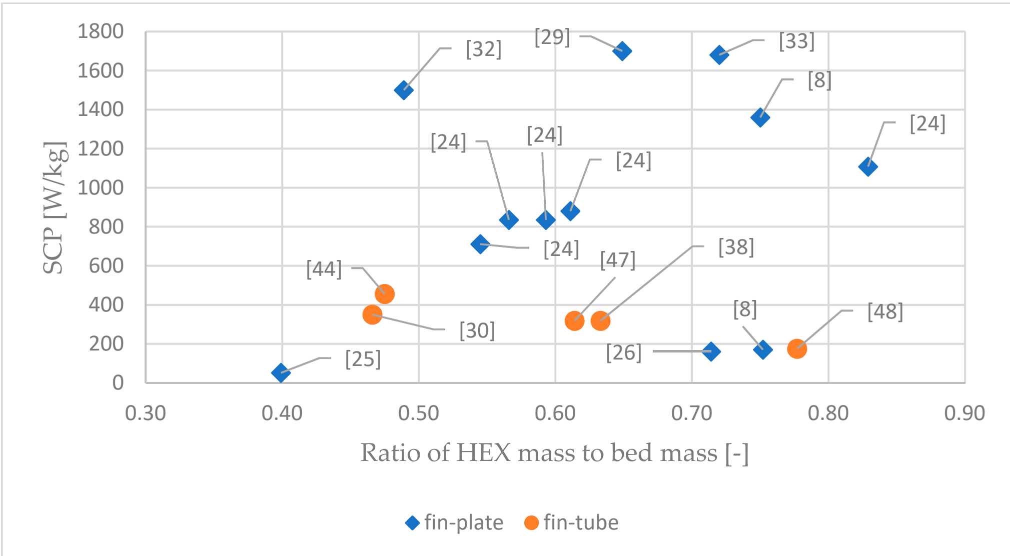

Figure 11. Correlation of SCP with the mass ratio of the HEX in the bed [8,24–26,29,30,32,33,38,44,47,48].

When designing a heat exchanger, attention should be paid to its susceptibility to dirt contamination. A strongly developed heat transfer surface (e.g., use of microchannels) improves heat transfer, but later on it may entail the risk of deposition of pollutants in the heat exchanger leading to impaired heat transfer.

The design requirements for the heat exchanger are that it should generate as low a pressure drop as possible and be lightweight with a large heat transfer surface area. On the other hand, the material from which the HEX is made must meet the process requirements (intensive heat transfer); at the same time, it must have a low price and operating cost.

The studies analyzed in this paper did not consider the HTF flow resistance. In the authors' opinion, the omission of a key factor that affects the power consumption makes it difficult to compare these devices with compressor chillers.

The flow resistance affects the efficiency of the chiller mainly through the electricity consumption to power the circulating pumps. However, in addition to this aspect, the maximum HFT mass flow and the flow characteristic must also be considered. Increasing flow resistance increases energy consumption decreasing the energy efficiency of the chiller and reduces the maximum HTF mass flow, which plays an important role in heat transport.

Larger HTF mass flow improves heat transport, thereby increasing the power of the heat exchanger. It follows that it would be beneficial to maximize the cross-sectional area of the HTF channels so as to reduce flow resistance. However, in terms of the flow characteristic from the heat transport point of view, it is advantageous to achieve a turbulent flow character by using obstacles and reducing the cross-sectional area of the HTF channel, which increases the flow velocity. This obviously involves an increase in flow resistance. Thus, it is necessary to optimize the channel dimensions and shape to maximize heat transport while minimizing the flow resistance.

Taking into account the variety of adsorption chiller designs and the diversity of operating conditions, such as bed heating and cooling temperatures and HTF mass flow rate, it should be stated that for a proper evaluation of a HEX in an adsorption bed, the SCP and COP indicators are insufficient, and it is impossible to compare individual exchangers based on the parameters referred to. The reason for this is the fact that the above-mentioned indicators take into account the adsorption chiller as a whole, and their value is influenced by, e.g., the sorption material used, the design of the evaporator and condenser, the circuit and control system used and, above all, the operating parameters. Therefore, an important step towards further analysis and development of future designs of HEXes used in adsorption beds might be the development of an evaluation index for exchangers where important parameters in terms of heat and mass transfer would be distinguished.

In the authors' opinion, the evaluation index for a HEX in an adsorption bed should take into account the following parameters:

1. Ratio of heat transfer surface area to adsorbent mass (S/m). As the value of this parameter increases, the mass of the adsorbent that is in direct contact with the exchanger components increases;
2. Mass of adsorbent in the bed (M_{ads})—the larger the value, the greater the uptake;
3. Mass of the heat exchanger (M_{HEX}) and the specific heat of the HEX material (c_p)—the lower the value, the lower the thermal inertia, which reduces cycle time and improves the COP;
4. Heat exchanger power (P)—the higher the HEX power, the more intensive the heat transfer, which allows a shorter cycle time and thus an increase in chiller efficiency;
5. Mass of HTF in the exchanger (M_{bed}—M_{HEX}—M_{ads}) and the flow resistance of HTF in the exchanger (Δp)—the lower the value, the lower is the electricity consumption for powering the circulation pumps and the shorter the exchanger flushing time;
6. Chiller operating temperatures (adsorption, desorption, condensation, and evaporation temperatures)—additional information on the conditions under which the tests were carrying out. It enables comparison of experimental results of studies conducted under similar conditions;
7. Geometrical parameters of the HEX fins.

Table 5 summarizes the values of selected parameters for fin-tube and fin-plate heat exchangers from the literature, for which the adsorption chiller operates at the highest values of the SCP and COP. The values collected constitute a preliminary database for optimizing the heat exchanger evaluation indicator. Nevertheless, it is also necessary to know the values of the other HEX characteristic parameters mentioned earlier by the authors. Such a complete set of data would allow to fully compare and evaluate the suitability of any new design of this type of exchangers for use in an adsorption chiller bed for a given adsorbent–adsorbate working pair.

Comparing fin-plate (plate) and fin-tube heat exchangers, it should be noted that fin-plate heat exchangers allow achieving higher SCP and COP values, mainly due to higher heat transfer surface area of fins and higher membrane surface area, which improves heat transport. In addition, fin-plate heat exchangers are characterized by a higher ratio of HEX mass to adsorbent mass, which on the one hand increases the SCP value but on the other hand increases the thermal inertia of the system, lowering the COP. When applying plate heat exchangers, one should also pay attention to the fact that matching their geometry to the chamber of the adsorption chiller is more difficult compared to fin-tube heat exchangers, as the chambers are usually in the form of a cylinder.

Table 5. Overview of optimum values for the main HEX parameters.

HEX Type	Parameter	Unit	Value for SCP_{MAX}	Value for COP_{MAX}
fin-tube	S/m	-	10.11	12.73
	M_{ads}	g	2770	9900
	M_{HEX}	g	2510	9300
	p_F	mm	1	1.8
	t_F	mm	0.12	0.12
	h_F	mm	0.15	5.24
	w_F	mm	475	475
	M_{HEX}/M_{bed}	-	0.633	0.777
fin-plate	S/m	-	49.8	b/d
	M_{ads}	g	49,86	7030
	M_{HEX}	g	160	8420
	p_F	mm	3	3
	t_F	mm	0.5	0.5
	h_F	mm	3	20
	w_F	mm	20	20
	M_{HEX}/M_{bed}	-	0.649	-

When designing an adsorption bed heat exchanger, the following design optimization conditions should be considered:

1. Maximization of the amount of adsorbent particles in direct contact with the HEX elements
2. Minimization of mass of HEX elements
3. Maximization of adsorbent mass
4. Minimization of flow resistance of HTF and refrigerant vapors
5. Minimize dead zones and residual water mass
6. Design exchanger geometry to avoid need for additional adsorbent particle stabilizing elements such as metal mesh
7. Minimize thermal expansion of HEX elements, especially when using coated bed
8. Maximization of heat conductivity coefficient by appropriate selection of HEX material

To summarize the authors' analysis of the available literature, it is important to note the lack of extensive research on the use of 3D printing technology in the context of manufacturing heat exchangers for adsorption beds. This technology provides many opportunities for the development of HEX designs for adsorption beds, both in terms of geometry shaping and materials used. Within the framework of HEX development perspectives, it is worth noting the possibilities of practically independent shaping of HEX surfaces on the HTF channel side and on the adsorbent side. Moreover, the wide flexibility in geometry design allows the use of topology optimization tools based on machine learning. On the other hand, in the material context, the printing technology allows the use of dopants in the HEX material to improve the thermal conductivity coefficient.

5. Conclusions

When analyzing the recent papers, some dominant trends in the design of heat exchangers used in adsorption beds can be observed. It seems that fin-plate exchangers can be characterized by higher COP and SCP values than fin-tube exchangers. However, the individual authors analyze HEXes in terms of different parameters, and this fact makes it difficult to compare the research results presented by different research teams. Therefore, it must be concluded that the knowledge on the HEXes that are used in adsorption beds is incomplete and requires some supplementation and research standardization.

Heat exchangers are typically studied in the context of their influence on the COP and SCP parameters of a particular adsorption chiller. However, this approach makes it impossible to compare specific HEX designs due to the different operating parameters of the chillers discussed in the papers, such as sorption, condensation, and evaporation temperatures, as well as the different working pairs (adsorbent-adsorbate) used in the beds. A certain similarity that exists in the heat exchanger designs analyzed is the geometry of the fins. As thin as possible fins with a thickness of up to 0.5 mm for both fin-tube and fin-plate HEXes. As for the fin pitch, it is in the range of 2–10 mm for fin-tube heat exchangers while for fin-plate heat exchangers this range is twice as wide.

It should be noted that 75% of the heat exchangers used in adsorption beds are made of aluminum or copper and the use of alternative materials, such as graphite, nylon, ABS, or PLA is limited to laboratory tests. The use of aluminum and copper as HEX materials is determined by their high thermal conductivity and low heat capacity as well as the widespread availability of these materials. However, the negative side of this is proof that modifications to the geometry of the exchangers in question can only be made in a limited scope because the current technology is not adequate to allow production of any metal HEX. It is due to this fact that attention is focused on the option of using polymers and 3D printing to manufacture heat exchangers for adsorption beds. This new approach allows virtually unlimited modifications of the geometry of heat exchangers, which, together with the use of thermally conductive binders, further reduces the heat capacity of HEXes.

While analyzing HEXes used in adsorption beds, the authors pointed out the impossibility of comparing various exchanger designs due to the lack of uniform parameters for HEX evaluation by various teams of researchers. The authors' further work will focus on developing an index for evaluating HEXes used in adsorption beds using computer simulations to make sure that such an index will be as accurate as possible in determining the effect of HEX parameters on the energy efficiency of an adsorption chiller. The authors' further work will focus on developing an index for evaluating HEXes used in adsorption beds using computer simulations to make sure that such an index will be as accurate as possible in determining the effect of HEX parameters on the energy efficiency of an adsorption chiller. The future works will involve the preparation of design proposals for plate-fin and tube-fin heat exchangers. The 3D HEXes models will be parameterized so that the geometrical parameters can be changed quickly and a series of simulation studies can be carried out. An important aspect of the research will be to keep constant the values of the other parameters of the systems for the following HEXes variants. This will allow us to determine the weighting factors of individual parameters defining the HEX design based on the influence of their values on the energy efficiency of the adsorption cooling system. This approach will enable comparison of different heat exchangers designs and is an opportunity to make significant progress in the development of the adsorption chiller industry.

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Nomenclature

C	heat capacity, kJ/K
c_p	specific heat, J/(kgK)
h_F	fin height, mm
M_{ads}	mass of adsorbent, g
M_{bed}	mass of bed, g
M_{HEX}	mass of heat exchanger, g
P	heat exchanger power, W
p_F	fin pitch, mm
Q_{ad}	heat removed from the bed during adsorption process, kJ
Q_c	heat removed from the bed during pre-cooling process, kJ
Q_{cond}	heat of adsorbate condensation, kJ
Q_d	heat delivered to the bed in the desorption process (driving heat), kJ
Q_{evap}	heat of adsorbate evaporation, kJ
Q_h	heat delivered to the bed in the pre-heating process, kJ
S	heat transfer surface, cm^2
S/m	ratio of heat transfer surface area to adsorbent mass, cm^2/g
t_{cycle}	cycle time, s
t_F	fin thickness, mm
w_F	fin width, mm
Δp	flow resistance of HTF in the HEX, Pa

Abbreviations

ABS	acrylonitrile butadiene styrene
AC	activated carbon
ART	adsorption refrigeration tube
COP	coefficient of performance
HEX	heat exchanger
HTF	heat transfer fluid
PA6	polyamide 6
PLA	polylactic acid
SCC	specific cooling capacity
SCE	specific cooling effect
SCP	specific cooling power
SG	silica gel
SVF	solid volume fraction

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