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System options for cooling of buildings making use of district heating heat

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Issues stemming from district heating utilization during summer periods and the conversion of low-temperature heat into cold in adsorption chillers have been investigated in this paper. Due to the high vulnerability of adsorption chillers to ambient conditions, in the case of relatively low ambient temperatures, adsorption-based air-conditioning systems would be characterized by excessive cooling power. Moreover, adsorption chillers are also characterized by high investment costs and big time constants, and the vulnerabilities found in their regulatory processes have yet to be sufficiently investigated. The authors recommend the application of hybrid air-conditioning systems, consisting of adsorption and compressor chillers. The adsorption chiller works as a base while the compressor chiller contributes missing cooling power, working as a regulation unit. Sixteen configurations of the hybrid air-conditioning system have been analysed. It has been shown that 100 kW cooling power hybrid air-conditioning system, with respect to its configuration, enables the utilization of 0.5 to 0.9 TJ of low-temperature heat per year, while simultaneously providing comfortable air-conditioning. The authors have concluded that the adsorption share in the analysed hybrid system should not exceed 50%.

Choix de systèmes pour le refroidissement de bâtiments faisant usage de la chaleur du chauffage urbain

Mots clés : Chauffage urbain ; Refroidissement par adsorption ; Refroidissement par compression de vapeur

1. Introduction

In European countries characterized by moderate climate, a recommended electrical and thermal energy production technology is cogeneration. It results from high heat requirement in wintertime and stable consumption of some amount of heat in summertime. A typical profile of cogenerative heat consumption by Wrocław – a middle-sized Polish city (about 600 thousand inhabitants) is depicted in Fig. 1. Similar district heating demand profile is also reported by other researchers in Netherlands (Geus et al., 2015). A ratio of winter to summer heat consumption is

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Nomenclature

AP	adsorption share
C	unit cost of energy [EUR MWh^{-1}]
CO	overall annual cost of energy [EUR a^{-1}]
COP	coefficient of performance
CP	cooling power [kW]
E	energy [MWh]
H_{ads}	heat of adsorption [kJ kg^{-1}]
H_{vap}	heat of evaporation [kJ kg^{-1}]
ic	specific investment cost [EUR kW^{-1}]
IC	investment cost [EUR]
P	electric input [kW]
q	adsorption equilibrium [kg kg^{-1}]
Q	heat [kW]
Q_0	heat load of building [kW]
RCP	relative cooling power
T	temperature [°C]
V	volume flow [m^3 s^{-1}]
Δm	adsorbed/desorbed water mass [kg]
Δp	pressure drop [Pa]
ΔT	temperature difference [K]
τ	daily duration [s]
η	efficiency

Subscripts

a	ambient
ads	adsorption chiller
comp	compressor chiller
cond	condensation
ct	cooling tower
cw	cooling water
d	driving
dp	dew point
ds	distribution system
el	electric energy
fans	fans
in	inlet
out	outlet
pump	pumping system
th	thermal energy

approximately 10:1. A significant problem of cogeneration (CHP) plants during periods of prevailing external high temperatures is found in the utilization of low-temperature heat, which has been derived from cogeneration. This heat must be delivered to the receivers as they require warm utility water. However, the minimum amount of power required for CHP plant operation has significantly exceeded receipts, particularly during the summer months. This indicates the necessity of controlled excess heat dissipation at temperatures of 60 °C and higher, what aggravates cogeneration economics. Simultaneously, the demand for air-conditioning increases during summer periods. An increased demand for air-conditioning corresponds to the additional demand of electric energy. However, cogenerative production must be limited due to the lack of heat demand. The perfect solution would be the implementation of technology that enables the conversion of district heating into a cold directly in the receiving end, suggested also by Daßler and Mittelbach (2012). Sorption chillers are devices that enable the conversion of heat into a cooling effect. The essence of operation of sorption chillers is based on utilizing the sorption bed capacity's dependence on pressure and temperature. These chillers are further divided into widely spread absorption chillers and adsorption chillers. The working medium for absorption chillers is water absorbed under low pressure in water solution LiBr and which, for sorbent regeneration purposes, also desorbs when a bed is heated to a minimum of 85 °C. These chillers can be installed close to cogeneration sources, but cannot be powered by district heating, which has a temperature of approximately 65 °C during the summer (Chorowski and Pyrka, 2015). Adsorption chillers can use water as a refrigerant adsorbed under low pressure by a bed filled with silica-gel. Silica-gel regeneration is possible in temperatures as low as 50 °C (Chorowski and Pyrka, 2015; Rahman et al., 2013; Saha et al., 2001) and district heating can be used for this purpose. Principles of operation and current advances on adsorption chillers are widely discussed in Li et al. (2014); therefore, they are not presented in this paper.

In the Polish climate, where summertime temperatures rarely exceed 30 °C, adsorption chillers with water–silica-gel working pair are a good choice. The experimentally determined adsorption isotherms for this working pair are presented in Fig. 2. The water vapour pressure and corresponding saturation tempera-

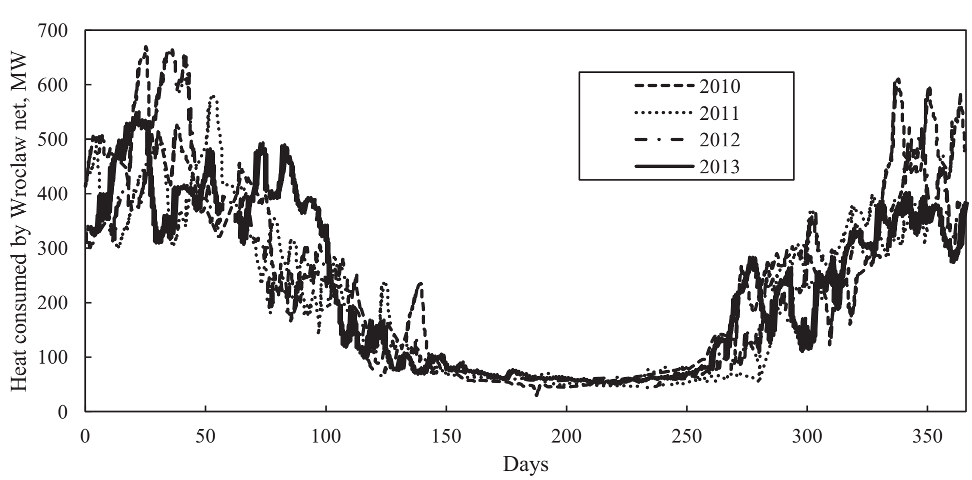

Fig. 1 – Yearly profile of cogenerative heat consumption in Wrocław, Poland.

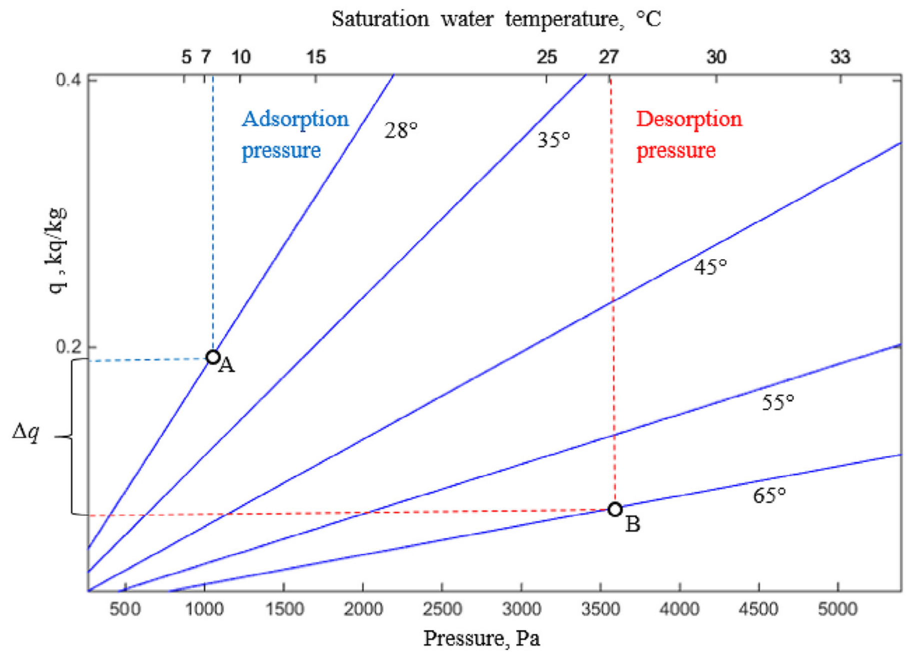

Fig. 2 – Silica-gel–water isotherms. Exemplary calculation for maximum q in one cycle: A – adsorption at 28 °C silica-gel for pressure equal to water saturation at 8 °C; B – desorption at 65 °C silica-gel for pressure equal to water saturation at 27 °C.

ture are presented on the bottom and top horizontal axes respectively. On the vertical axis, the degree of adsorption expressed in kilograms of adsorbed water per each kilogram of dry silica-gel is marked. The exemplary state points of the adsorption chiller operation are also marked in Fig. 2: it was assumed that during the time period it would take for adsorption to occur, the silica-gel could be cooled down to a temperature of 28 °C, and adsorption would be conducted under pressure corresponding to water vapour pressure at 8 °C (the obtained chilled water temperature being greater than 8 °C), which corresponds to point A. During regeneration, the silica-gel is heated to a temperature of up to 65 °C. The desorbed water is then condensed at 27 °C, which corresponds to point B. The difference in height between points A and B is represented by q, which corresponds to the amount of water transported by one kilogram of the bed during one cycle. It is possible to combine the mass of the water transported together with the cooling power. The slope of adsorption isotherms is steeper than the slope of regeneration curves. Therefore, the cooling capacity is more strongly affected by chilled and cooling water temperatures than by the heating temperature. It can also be noted that raising the silica-gel temperature by about 7 °C (from 28 °C to 35 °C) for adsorption will result in lower adsorption chiller performance, more so than it would by decreasing the silica-gel temperature by 10 °C (from 65 °C to 55 °C) during regeneration. Adsorption physics makes adsorption chillers sensitive to the input parameters, especially cooling water temperature, which influences both the temperature of the silica-gel during adsorption and the pressure in which water is desorbed during regeneration. The temperature of the cooling water provided to the chiller depends on ambient temperature. The performance of the adsorption chiller becomes depleted when the ambient temperature is high. The cooling power and *COP* of the chiller are reduced when air-conditioning is the most necessary.

Due to the high vulnerability of the adsorption chiller to ambient conditions, it is not suitable for autonomic exploitation. Moreover, adsorption chillers are characterized by high investment costs (1000–1500 €/kW of cooling) and great inertia, which is typical for heat devices. The authors recommend the application of hybrid air-conditioning systems, consisting of adsorption and compressor chillers. The adsorption chiller works as a base while the compressor chiller contributes missing cooling power, working as a regulation unit. The year-round operation of systems with adsorption shares in the scope of 0–100% were modelled. Investment and operational costs, electric energy and heat demand were compared for sixteen configurations of the hybrid air-conditioning system. The economic comparison of compression and adsorption systems, which make use of district heating is not totally original. It has already been discussed by Grzebielec et al. (2015). The techno-economic performance of different configurations of the hybrid air-conditioning system has not been investigated yet. The following investigation will provide guidelines for the commercial application of adsorption technology, especially in terms of reasonable adsorption share and the configuration of the system.

2. Detailed literature review

The authors Grzebielec et al. (2015) have compared the possibility of using district heating in adsorption chillers instead of electrically driven chillers in terms of technical and economic issues. They have found it to be uneconomical due to the low *COP* of the adsorption chiller at 0.14 and the high district heating water prices.

The typical district heating water temperature does not exceed 70 °C; therefore, the expected heating water temperature ranges from 55 to 70 °C. The typical cooling water temperature, dependent on ambient temperature and type of recooler, ranges in Poland from 20 to 35 °C. Due to the aforementioned vulnerability of the chiller to the chilled water temperature, the authors recommend that the temperature is maintained not lower than 10 °C. The literature review is focused on research concerning the performance of the chiller at temperatures close to those mentioned above. The main parameters used to describe the chiller are cooling power and the coefficient of performance. These factors strongly depend on certain working conditions, such as circuit temperatures, cycle time, and flow rates, therefore the performance of the chillers cannot be compared without regard to these conditions. Thermal *COP*, which refers to the ratio of the obtained cooling power to the thermal power input of the chiller, is commonly used. The thermal *COP* does not include the additional electric power input used to generate cooling power.

$$COP = \frac{CP_{ads}}{Q_d} \tag{1}$$

CP_{ads} – cooling power of adsorption chiller
Q_d – heat input of adsorption chiller

Ideally, the thermal *COP* of the adsorption chiller would be a ratio of heat obtained as a result of the evaporation of some water weight Δm in the evaporator to the heat that needs to be provided in order to desorb the same amount of water from the adsorption bed. This limits the value of the thermal *COP* of silica-gel/water chillers to about 0.9.

$$COP_{id} = \frac{H_{vap} * \Delta m}{H_{ads} * \Delta m} = \frac{H_{vap}}{H_{ads}} \approx 0.9 \tag{2}$$

Table 1 presents an overview of the newest research works for adsorption chillers. Khan et al. (2006) simulated the operation of 4-bed 2-stage adsorption chiller for heating water temperature 50–70 °C and cooling water temperature 30 °C. Obtained thermal *COP* was of up to 0.39.

Similar installation was simulated by Hamamoto et al. (2005) and for waste heat temperature of 55 and 70 °C researchers obtained a *COP* of 0.23 and 0.26, respectively.

Akahira et al. (2005) examined a prototype chiller with mass regeneration with heating, cooling and chiller inlet water temperatures of 70 °C, 30 and 14, respectively. For a cycle time of 1020 s (adsorption/desorption 420 s, mass recovery 60 s), the obtained *COP* ranges from 0.1 to 0.35.

Xia et al. (2008) examined the chiller with mass recovery with heating, cooling and chiller inlet water temperatures of 82.5 °C, 30.4 and 11.9, respectively. Achieved *COP* was 0.388 and cooling power reached 8.69 kW. Increase of chilled water temperature to 16.5 °C results in increase of *COP* and cooling power of about 11% and 26%, respectively.

Saha et al. (2001) in order to utilize 55 °C solar/waste heat proposed and examined a 2-stage chiller. For the chilled water temperature of about 9 °C and cooling water temperature of 30 °C, the obtained *COP* was 0.36. The chiller combined with flat solar collectors was found to be profitable in tropical climate.

Rahman et al. (2013) have conducted extensive theoretical calculations on a 3-bed chiller with mass recovery, optimizing cycle time. For temperatures of heating water ranging from 50 to 90 °C, these researchers have managed to obtain a *COP* of up to 0.61. Adsorption chiller performance was found effective for the heating water temperature as low as 50 °C.

Chorowski and Pyrka (2015) have investigated the possibility of utilizing low-temperature heat from CHP plants in 3-bed 2-evaporator adsorption chiller. They examined the chiller for the heating water temperatures from 47 °C to 65 °C reaching a *COP* of up to 0.64. Chiller was found to be suitable for trigeneration and desalination systems.

Li et al. (2014) simulated and experimentally validated the performance of 4-bed chiller at various operating conditions. The chiller was analysed for heating water temperature 75 and 65 °C, cooling and chilled water temperatures 29.4 and 12.2 °C, respectively, and for cycle time ranging from 110 to 430 s. Achieved *COP* was up to 0.32 and 0.27.

Di et al. (2007) provided extended study of adsorption chiller operating on different heating water temperatures from 65 to 85 °C. The inlet cooling water temperature is always 30 °C, while inlet chilled water temperature is 20 °C. Study shows the depletion of *COP* and cooling power when heating water temperature decreases. Achieved *COPs* are 0.44, 0.42 and 0.35 for heating water temperatures 85, 75 and 65 °C.

Wang et al. (2005) performed the study of a prototype chiller with mass recovery under different heating and chilled water temperatures. The chiller is proved to be effectively driven by low temperature heat achieving *COP* about 0.4.

Geus et al. (2015) provided an extended data about commercially available chillers for different temperatures. It shows how strongly cooling and chilled water temperature affect *COP* of the chiller. For fixed heating and chilled water temperatures 72 °C and 21 °C *COPs* are 0.548, 0.488, 0.407 for cooling water temperatures 27 °C, 33 °C and 36 °C. An increase of cooling water temperature from 27 °C to 36 °C results in reduction of *COP* by 26%.

Design and experiment performed by Pan et al. (2016) shows good performance of full scale chiller. Chillers were investigated for heating water 86 °C, cooling water 30 °C and chilled water 11 °C achieving *COP* 0.501 and cooling power of about 43 kW.

The presented literature overview proves the adsorption chiller to be effectively driven by low-temperature heat such as district heating. The vulnerability of the *COP* and cooling power of the adsorption chiller to the cooling and chilled water temperature is also shown. Within the investigated temperature range, the adsorption chillers are able to reach a *COP* of up to 0.64, which is about 70% of the theoretical value for the working pair of silica gel–water. This means that 1 kW of low-temperature heat can be converted to about 0.7 kW of cooling power. Based on the literature review as well as the authors' own experience, the assumed characteristics of the adsorption chiller's *COP* and relative cooling power, as presented in Fig. 3, were considered. Moreover, the assumed characteristics of the compressor chiller *COP*, as presented in Fig. 3, were also taken into consideration.

3. Hybrid air-conditioning system architecture

The modelled system consists of two main elements: the chilled water distribution system and the hybrid air-conditioning

Table 1 – Overview of adsorption chiller research.

Citation	Description	Cooling water temperature [°C]	Heating water temperature [°C]	Chilled water temperature [°C]	Cooling power [kW]	COP [-]	Remarks
Khan et al. (2006)	2-stage 4-bed adsorption chiller	30	50–70	10		0.39	Theoretical model, steady operation
Hamamoto et al., (2005)	2-stage 4-bed adsorption chiller	30.2	55 and 70	10	–	0.23 and 0.26	Theoretical model and experimental validation, steady operation
Akahira et al. (2005)	2-bed adsorption chiller with mass regeneration and indirect methanol heat pipe evaporator	30	70	14		0.1–0.33	Experimental research, steady operation, cycle time of 1020 s (adsorption/desorption 420 s, mass recovery 60 s)
Xia et al. (2008)	2-bed adsorption chiller with mass regeneration and indirect methanol heat pipe evaporator	30.4	82.5	11.9 and 16.5	9 and 11	0.388 and 0.432	Experimental research, steady operation
Saha et al. (2001)	Adsorption chiller supplied with waste/solar heat	30	55	9		Up to 0.36	Experimental research, steady operation, combined with flat solar collector
Rahman et al. (2013)	3-bed adsorption chiller with mass regeneration and cycle time optimization	30	50–90	14		Up to 0.61	Experimental research, steady operation, 50 °C found effective to drive the adsorption chiller
Li et al. (2014)	4-bed adsorption chiller at various operating conditions	29.4	65 and 75	12.2	–	Up to 0.27 and 0.32	Theoretical model and experimental verification, cycle times from 110 to 430 s
Di et al. (2007)	2-bed adsorption chiller with heat pipe indirect methanol evaporator and mass recovery	30	65, 75 and 85	20		0.44, 0.42 and 0.35	Theoretical model and experimental validation
Wang et al. (2005)	2-bed adsorption chiller with mass recovery	30.5	65–85	15.7–20.6	–	Up to 0.4	Experimental research, different cycle and recovery cycle times
Geus et al. (2015)	2-bed adsorption chiller integrated with recooler	27, 33 and 36	72	21	3.075 and 3.832	0.548, 0.488 and 0.407	Experimental research on early stage commercial chiller
Chorowski and Pyrka (2015)	3-bed 2-evaporator adsorption chiller	20–27	47–65	6–20	up to 120	Up to 0.64	Direct cooling circuit, found to be suitable for trigeneration and desalination systems
Grzebielec et al. (2015)	2-bed adsorption chiller	15 and 25 (ambient temperature)	65	12/7	5 and 1	0.45 and 0.14	Drycooler
Pan et al. (2016)	2-bed adsorption chiller	30	86	11	43	0.501	Theoretical model and experimental verification, simultaneous cooling and desalination

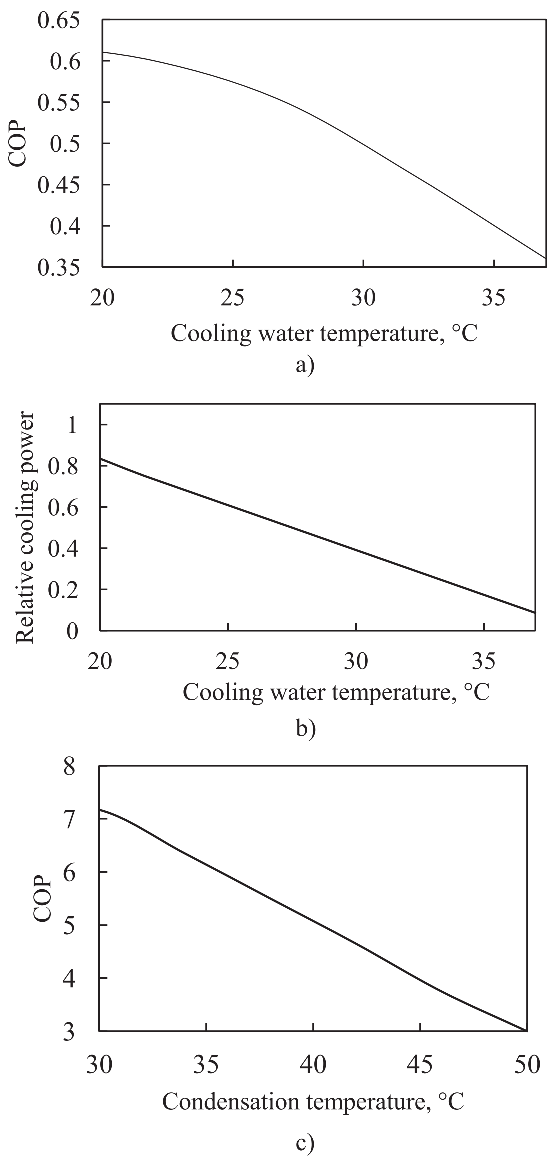

Fig. 3 – COP (a) and cooling power (b) adsorption chiller characteristics with respect to cooling water temperature; COP compressor chiller characteristics (c) with respect to condensation temperature.

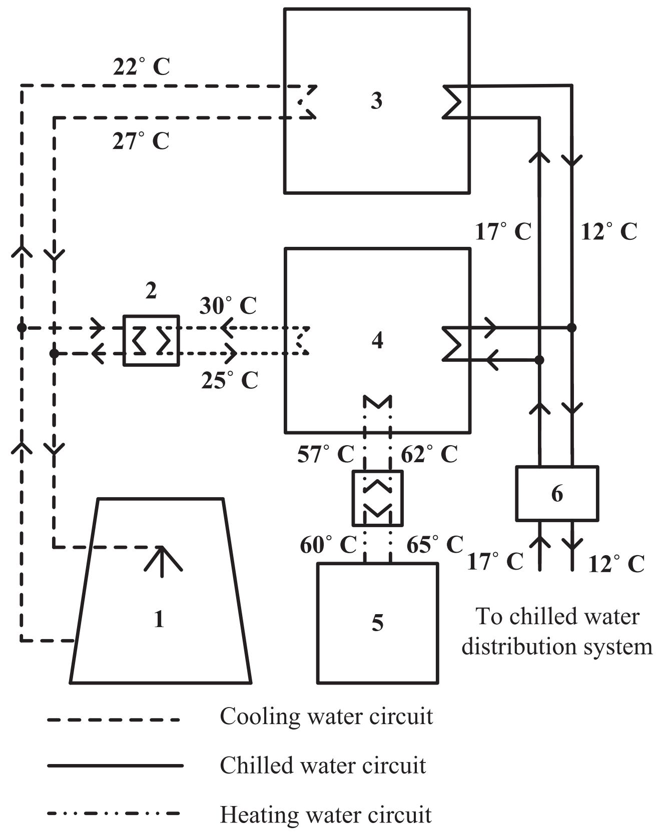

Fig. 4 – Hybrid air-conditioning system with parallel connection to indirect cooling water circuit, open cooling tower and parallel connection to chilled water circuit with exemplary water circuit temperatures: 1 – open cooling tower, 2 – indirect heat exchanger, 3 – compressor chiller, 4 – adsorption chiller, 5 – dictrict heating, and 6 – hydraulic coupling.

system, producing chilled water. The application of a variable-flow chilled water distribution system was proposed – the pump maintains pressure on the discharge of chilled water and the automation of the fan-coil unit controls the valves in order to guarantee the proper chilled water flow with respect to the current heat load. An advantage of the variable-flow chilled water distribution system is that there is smaller water flow compared to that of the constant flow system, resulting in electric energy savings for pumping. Due to the application of the variable-flow chilled water distribution system and the necessity of guaranteeing nominal flow through the chillers, hydraulic coupling is recommended. In the analysed case, the adsorption chiller is supplied with heat from district heating. Due to the high pressure in district heating, the adsorption chiller is indirectly supplied by the heat exchanger. Sixteen different configurations of the hybrid air-conditioning system have been analysed. These configurations differ by way of the connection of the chillers to the cooling water circuit (indirectly, directly, and parallel/serial), chilled water (parallel/serial), and the type of cooling tower (recooler/open cooling tower). The configuration with a closed cooling tower and direct connection due to the parameters of the cooling water can be treated the same as if using an open tower along with the separating heat exchanger. Exemplary hybrid air-conditioning system configurations are presented in Figs. 4 and 5. In Fig. 4, the chillers are connected in parallel to the cooling water and chilled water circuits with an open cooling tower. The adsorption chiller is indirectly connected to the cooling water circuit. Fig. 5 presents a hybrid air-conditioning system where the chillers have a serial connection to the cooling and chilled water circuits. The adsorption chiller is directly connected to the cooling water circuit which is equipped with a recooler.

3.1. *Cooling water circuit*

The heat coming from the heat load of the building, the heat input to the adsorption chiller, and the electric power driving the compressor chiller and pump systems must be finally

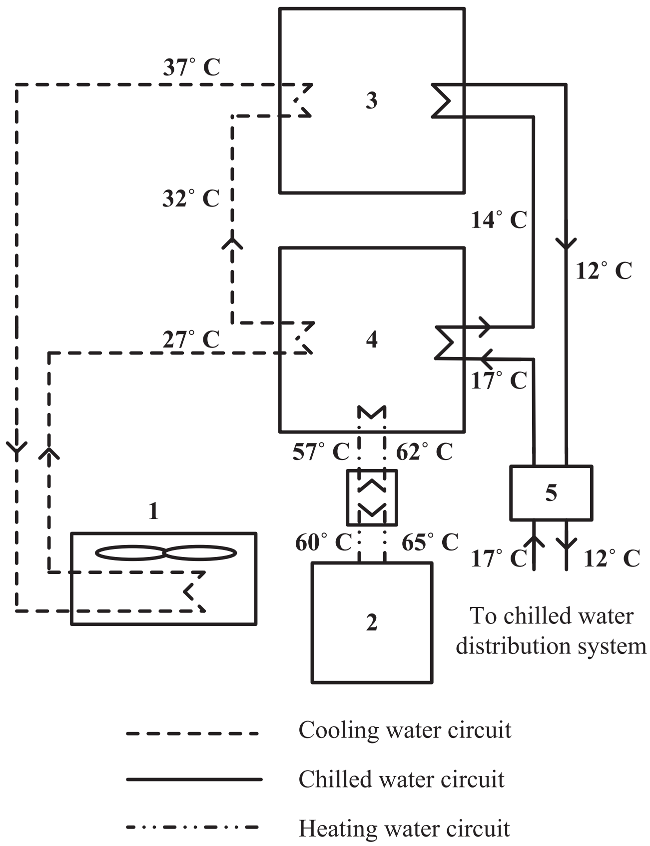

Fig. 5 – Hybrid air-conditioning system with serial connection to direct cooling water circuit, recooler and serial connection to chilled water circuit with exemplary water circuit temperatures: 1 – recooler, 2 – district heating, 3 – compressor chiller, 4 – adsorption chiller, and 5 – hydraulic coupling.

dissipated into the environment. Heat dissipation can be implemented in the recooler, a closed or open cooling tower, in a direct circuit (where there is a common circuit for the chiller and cooling tower), or in indirect circuits (where the chiller and cooling tower circuits are separated by a heat exchanger). The cooling water temperature, which has a big influence on the *COP* and *RCP* of the adsorption chillers, depends on the selection of the cooling device and type of circuit. The application of the open cooling tower makes it possible to reach the lowest cooling water temperature, depending on the dew point temperature and the current heat load. The dew point temperature is maintained at a level of 14–23 °C, even on the hottest days of the Polish climate. Open cooling towers are light and cheap when compared with recoolers. The disadvantages of using an open cooling tower are found in the contact between the cooling water and air with the possibility of water pollution, as well as the necessity of water removal from the cooling tower during periods of sustained negative temperatures. Recoolers are easier in application. However, the cooling water temperature obtained in a recooler is higher than that obtained from an open cooling tower. Moreover, recoolers are also more expensive. Producers of adsorption chillers require the application of a separating heat exchanger in the cooling water circuit. These heat exchangers increase the temperature of the cooling water and cause a further drop in pressure. Some adsorption chillers (Chorowski and Pyrka, 2015) do not require the separation of the cooling water system, and cooling can be executed in the direct circuit. Cooling for the adsorption and compressor chillers was assumed to be provided from the same cooling tower. Two possible cooling water circuit connections were analysed: serial (Fig. 5) and parallel (Fig. 4). The parallel system provides a lower temperature of cooling water for the compressor chiller and increases the reliability of the installation – pumping is executed by separate pumps. The serial connection reduces the required amount of the cooling water. Rather than the amount of cooling water necessary for two chillers, just the amount required to flow through a single, bigger one is sufficient. This can be important if pumping costs are high and derived from e.g. the distance of the chillers from the cooling tower. A disadvantage of such a connection is found in the decreased efficiency of the compressor chiller due to the higher temperature of the cooling water. On the other hand, the higher temperature of the cooling water makes it possible to use a smaller cooling tower than would be allowed with a parallel connection.

3.2. *Chilled water circuit*

Chillers can be connected to a chilled water circuit in two ways: parallel (Fig. 4) or serial (Fig. 5). In a parallel connection, both chillers are independent of each other. The total chilled water flow is greater than that in the case of a serial connection. The electric energy consumed for pumping water is proportional to the amount of pumped water. In a parallel connection, both chillers use an independent chilled water pump. This means that even if one pump were to fail, the system would continue working, albeit with reduced power. In the case of a serial connection, chilled water is first pre-cooled by the adsorption chiller before entering the compressor chiller, and then further cooled down to the desired temperatures. This enhances the performance of the adsorption chiller. On the other hand, the compressor chiller is supplied with chilled water at a lower temperature, which makes its efficiency lower than that in the case of a parallel system. In the case of different nominal chilled water flows of the chillers (which may occur in some ranges of adsorption share), it is necessary to bypass the chiller characterized by a lower nominal water flow. Sixteen different system configurations have been analysed, which are listed in Table 2.

4. Methodology of hybrid air-conditioning system efficiency analysis

The annual distribution of the average daily temperatures as registered for Wroclaw (Poland) in 2013 (Anon, 2013) was used in the calculations, shown in Fig. 6. Calculations were done for the referential object with a maximum heat load of 100 kW. The maximum heat load was estimated with regard to solar irradiation and convective gains from external and internal heat sources such as humans and hardware. Heat loads were then assumed to be proportional to the ambient temperature, from 0 kW below 17 °C, up to 100 kW for 35 °C.

Table 2 – Description of hybrid air-conditioning system configurations.

System	Cooling water	Chilled water	Heat rejection	
1	Parallel	Parallel	Direct	Open cooling tower
2	Parallel	Parallel	Indirect	Open cooling tower
3	Parallel	Parallel	Direct	Recooler
4	Parallel	Parallel	Indirect	Recooler
5	Series	Parallel	Direct	Open cooling tower
6	Series	Parallel	Indirect	Open cooling tower
7	Series	Parallel	Direct	Recooler
8	Series	Parallel	Indirect	Recooler
9	Parallel	Series	Direct	Open cooling tower
10	Parallel	Series	Indirect	Open cooling tower
11	Parallel	Series	Direct	Recooler
12	Parallel	Series	Indirect	Recooler
13	Series	Series	Direct	Open cooling tower
14	Series	Series	Indirect	Open cooling tower
15	Series	Series	Direct	Recooler
16	Series	Series	Indirect	Recooler

Three periods were distinguished: heating period, cooling period, and free cooling period, based on the following assumptions:

- Heating period is for external temperatures below 10 °C (196 days per year);
- Cooling period is for temperatures above 17 °C (170 days per year, 2720 air-conditioning hours per year); and
- Free cooling period occurs when cooling water temperatures drop below 15 °C at the discharge of the cooling tower (69 days within cooling period).

The hybrid air-conditioning system was not analysed in heat pump mode during the heating period. The hybrid air-conditioning system performance was modelled for various cooling aspects. This included operating costs, annual thermal and electric demand, heat dissipated into the cooling towers, and the nominal capacity required for adsorption

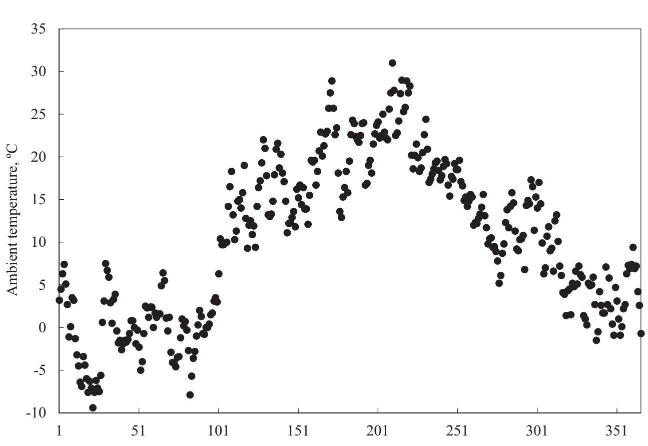

Fig. 6 – Annual distribution of average daily temperatures registered for Wrocław in 2013 (Anon, 2013).

and compressor chillers. Various system configurations and adsorption shares were considered. The adsorption share is defined as a maximum cooling power which can potentially be produced by the adsorption chiller referred to maximum cooling power of hybrid air-conditioning system.

$$AP = \frac{CP_{ads,max}}{CP_{max}} \quad (3)$$

The average daily heat load was defined, as based on the annual distribution of average daily temperatures, and corresponds to the cooling power that needs to be generated by the hybrid air-conditioning system. On the basis of ambient temperature and the type of cooling tower used, the cooling water temperature may be obtained as it leaves the cooling tower. The cooling water temperature nearly equals the dew point of the ambient air temperature for open cooling towers. The assumed difference is 2 K.

$$T_{ct,out} = T_{dp} + 2 \quad (4)$$

In the case of the recooler, the cooling water temperature slightly exceeds ambient air temperature. The assumed difference is 2 K.

$$T_{ct,out} = T_a + 2 \quad (5)$$

T_a – ambient temperature

In the direct cooling circuit, the cooling water temperature, which supplies the chillers is equal to the cooling water temperature as it is discharged from the cooling tower.

$$T_{cw} = T_{ct,out} \quad (6)$$

In the indirect cooling water circuit, the temperature of the water supplying the chillers is higher than the cooling water temperature as it leaves the cooling tower. This is due to the difference in temperature from the heat exchanger.

$$T_{cw} = T_{ct,out} + 3 \quad (7)$$

The condensation temperature of the compressor chiller equals the sum of the cooling water temperature plus a sufficient temperature difference, which is 13 K, providing the appropriate conditions for heat exchange.

$$T_{cond} = T_{cw} + \Delta T \quad (8)$$

The adsorption chiller is powered as the first in a serial connection of chillers to the cooling tower. The compressor chiller is cooled by water which has already received heat in the adsorption chiller condenser.

$$T_{cw\ comp,in} = T_{cw\ ads,out} \quad (9)$$

In hybrid air-conditioning systems with parallel chiller connections to the cooling water circuit, both chillers are supplied with water at the same temperature, coming either directly from the cooling tower discharge (direct cooling water circuit) or the separating heat exchanger (indirect cooling water circuit). For fixed heating and chilled water temperatures, the COP and cooling power of the adsorption chiller and the COP of a compressor chiller depend only on the cooling water temperature, as presented in Fig. 3.

Maximum cooling power, which is generated by the adsorption chiller equals:

$$CP_{ads,max} = CP_{max} * AP \tag{10}$$

Missing cooling power is complemented by the compressor chiller.

$$CP_{comp} = CP_{max} - CP_{ads,max} \tag{11}$$

The actual cooling power of the adsorption chiller depends on the cooling water temperature, according to Fig. 3b.

$$CP_{ads} = CP_{ads,max} * RCP \tag{12}$$

The thermal power input, which must be provided to the adsorption chiller in order to obtain cooling power at a certain COP (Fig. 3a) equals:

$$Q_d = \frac{CP_{ads}}{COP_{ads}} \tag{13}$$

The electric power input that must be provided to the compressor chiller in order to obtain cooling power at a certain COP (Fig. 3c) equals:

$$P_d = \frac{CP_{comp}}{COP_{comp}} \tag{14}$$

Electric power is also required for recooler fans and pump systems. The electric power consumption of the pumps is proportional to the generated pressure drop Δp, flow of the medium, and device efficiency. Based on this, the electric power consumption of the fans can be calculated.

$$P_{pump} = \Delta p * \dot{V} / \eta_{pump} \tag{15}$$

$$P_{fan} = \Delta p * \dot{V} / \eta_{fans} \tag{16}$$

Based on the proportions of the electric power consumption of the pumps and fans to the flow of the medium, which is dependent on the performance of the system, the electric power consumptions of the pumps and fans were assumed to be constants per unit of dissipated power.

$$P_{pump} = 0.02 \frac{kW}{kW_{dissipated\ power}}.$$

For the open cooling tower:

$$P_{fan} = 0.012 \frac{kW}{kW_{dissipated\ power}}.$$

For the recooler:

$$P_{fan} = 0.014 \frac{kW}{kW_{dissipated\ power}}.$$

Heat that must be rejected by the cooling tower is the sum of the heat load of the building, the electric power required for the compressor chiller and pumps, and the driving heat of the adsorption chiller.

$$Q_{ct} = Q_0 + P_d + Q_d + P_{pump}. \tag{17}$$

Thermal and electric energy, which must be delivered within a day to provide cooling, is expressed as:

$$E_{th} = Q_d * \tau \tag{18}$$

$$E_{el} = (P_d + P_{pump} + P_{fan})\tau \tag{19}$$

where τ is the daily duration of the system's operation which was assumed to be $\tau = 16$ h. Based on daily energy demands, the daily heat and electricity expenses which are necessary for compressor chillers, fan-coil units and the pump system were calculated.

$$CO_{th} = E_{ch} * C_{th} \tag{20}$$

$$CO_{el} = E_{el} * C_{el} \tag{21}$$

where C_{th} and C_{el} are costs per unit of heat and electric energy respectively, equals:

$$C_{th} = 8.35\ EUR/MWh$$

$$C_{el} = 130\ EUR/MWh$$

The annual values are sums of daily values. The performances of the modelled configurations of the hybrid air-conditioning system were compared. Investment costs of hybrid air-conditioning system were assumed to be the sum of adsorption chiller, compressor chiller, cooling tower and distribution system investment costs.

$$IC = IC_{ads} + IC_{comp} + IC_{ct} + IC_{ds} \tag{22}$$

Investment costs are proportional to cooling power and dissipated heat, respectively, for chillers and cooling tower. Investment costs of adsorption chiller were derived from manufacturers. Investment costs of distribution systems are assumed to be constant for each configuration of hybrid air-conditioning system.

$$ic_{ads} = 2030\ EUR/kW$$

$$ic_{comp} = 130\ EUR/kW$$

$$ic_{ct} = 50\ EUR/kW$$

$$IC_{ds} = 32\,000\ EUR$$

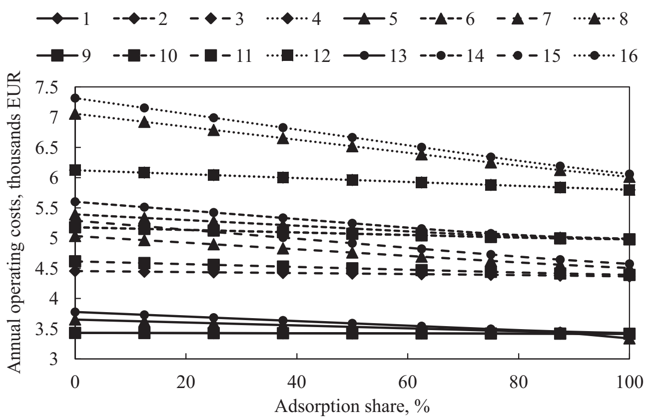

Fig. 7 – Annual operating costs with respect to hybrid air-conditioning system configuration and adsorption share, for a building with a heat load of up to 100 kW, the curves 1–16 as in Table 2.

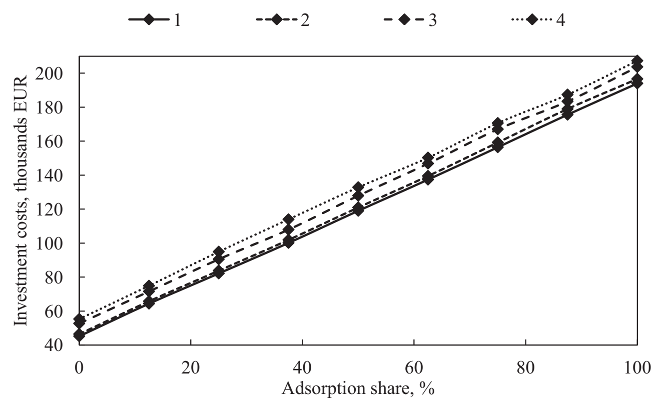

Fig. 8 – Investment costs with respect to hybrid air-conditioning system configuration and adsorption share of chosen hybrid air-conditioning system configurations, for a building with a heat load of up to 100 kW, the curves 1–4 as in Table 2.

5. Modelling of hybrid air-conditioning system installed in the referential building with a maximum heat load of 100 kW

Fig. 7 shows the annual operating costs of a hybrid air-conditioning system with respect to its configuration and adsorption share. Irrespective of the configuration, operating costs decrease with an increase in adsorption share. The configuration significantly affects the operating costs of a hybrid air-conditioning system. For systems provided with the same equipment and apparatus, but variously configured, differences of operating costs reach up to 50%. The lowest costs were obtained for hybrid air-conditioning systems with open cooling towers and direct connections to the cooling water circuit; the highest costs for systems with a recooler indirectly connected to the cooling water circuit for the entire range of the adsorption share. Hybrid air-conditioning systems with cooling and chilled water circuits which are serially connected to the chillers for low shares of adsorption, are significantly more expensive, but their operating costs decrease sharply with an increase in adsorption share. The application of these configurations can be found to be beneficial for high adsorption shares.

Fig. 8 shows the relationship of investment costs with respect to hybrid air-conditioning system configuration and adsorption share. In order for the figure to be clear, only the investment costs of the selected hybrid air-conditioning system configurations are shown.

The main factor that determines the investment costs of hybrid air-conditioning systems is adsorption share. At maximum adsorption share, investment costs reach up to 0.2 million EUR, which corresponds to 460% of the lowest investment costs for 0% adsorption. The lowest investment costs were obtained for hybrid air-conditioning systems with chillers that had parallel connections to the cooling and chilled water circuits; the largest investment costs, for systems with serial connections.

Operational costs are the sum of costs incurred by thermal energy for supplying the adsorption chiller, and for electric energy which powers the compressor chiller, pumps, and fans.

Fig. 9 shows the annual electric energy demand of a compressor from the hybrid air-conditioning system.

The demand for electric energy reduces with respect to the adsorption share of the hybrid air-conditioning system. When the open cooling tower in the direct cooling water circuit is applied, in the case of high adsorption share, the electric energy demand of a compressor chiller nearly vanishes. In terms of the indirect recooler, the electric energy demand for the compressor chiller is still significant. Due to the high temperatures of the cooling water, cooling power generated by the adsorption chiller is reduced. The compressor chiller compensates for a shortage in cooling power. Poor configuration of the hybrid air-conditioning system causes the adsorption chiller to work inefficiently. The missing cooling power has to be compensated for by the compressor chiller.

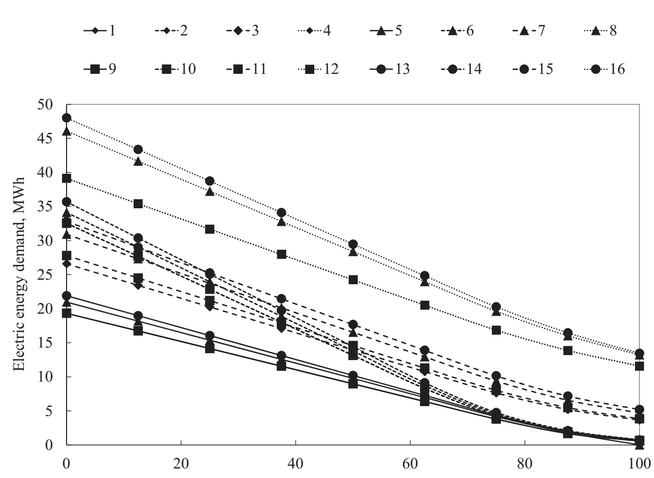

Fig. 9 – Annual electric energy demand of a compressor chiller with respect to the hybrid air-conditioning system configuration and adsorption share, for a heat load of up to 100 kW, the curves 1–16 as in Table 2.

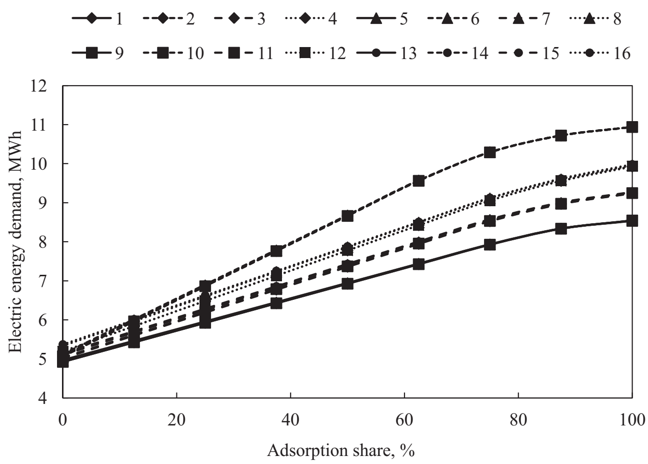

Fig. 10 – Annual electric energy demand of pumping systems with respect to hybrid air-conditioning system configuration and adsorption share, for a heat load of up to 100 kW, the curves 1–16 as in Table 2.

Fig. 10 shows the annual energy demand of the pumping systems. The main factors that will influence the electric energy demands of the pumping system are adsorption share and the type of cooling water circuit used. The electric energy demand rises with an increase in adsorption share. Due to the relatively low *COP* of adsorption chillers, much more energy has to be delivered and subsequently removed from the air-conditioning system. The application of additional heat exchangers, additional pumps (indirect cooling water circuit) or an open cooling tower can cause an increase in the electric energy demands of the pumping systems. Systems with an open cooling tower are characterized by greater hydraulic resistance due to the use of spray nozzels. The method used to connect the chillers to the cooling and chilled water circuits does not seem to have a significant impact on electric energy demands.

Fig. 11 presents the electric energy demands of the fans. Much like the electric energy demands of the pumping systems, the demands of the fans depend mainly on adsorption share and the type of cooling water circuit used in the hybrid air-conditioning

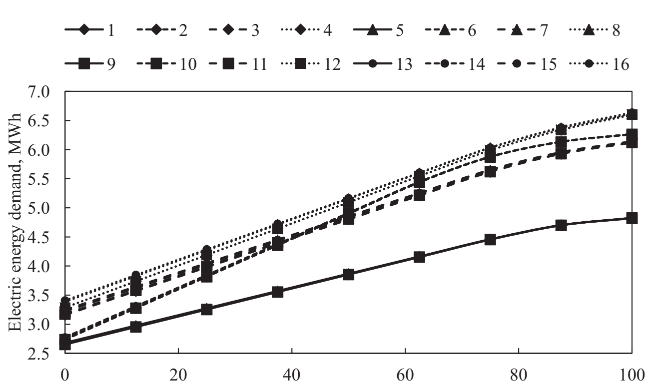

Fig. 11 – Annual electric energy demand of fans with respect to hybrid air-conditioning system configuration and adsorption share, for a heat load of up to 100 kW, the curves 1–16 as in Table 2.

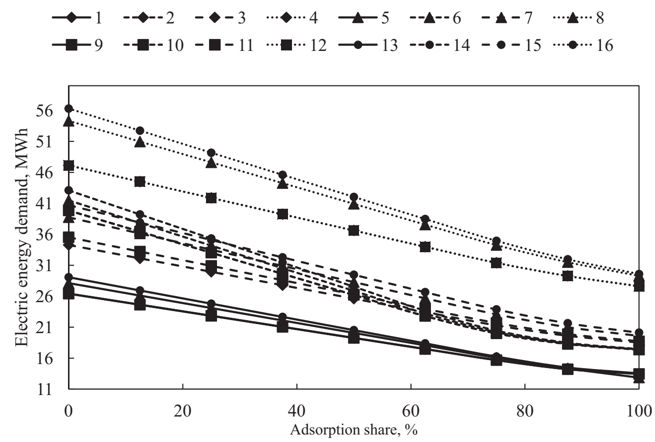

Fig. 12 – Total annual electric energy demand with respect to hybrid air-conditioning system configuration and adsorption share, for a heat load of up to 100 kW, the curves 1–16 as in Table 2.

system. The use of electric energy is proportional to the energy dissipated into the cooling tower. Due to the low *COP* of adsorption chillers, the energy dissipated into the cooling towers increases significantly with adsorption share. The lowest electric energy demand was obtained for hybrid air-conditioning systems with an open cooling tower in a direct cooling water circuit.

Fig. 12 presents the total annual electric energy demands of a hybrid air-conditioning system. The demand for electric energy decreases with an increase in adsorption share. At 100% adsorption share, the electric energy demand drops to around half of its maximal demand. In the case of high adsorption shares, the electric energy demand of the compressor chiller is reduced. Nevertheless, due to the low *COP* of adsorption chillers, there is a significant increase in the electric energy demand for the pumps and fans.

For adsorption shares not exceeding 50%, a dominant factor in determining the total electric energy demand is based on the demands of the compressor. When the adsorption share of a hybrid air-conditioning system exceeds 50%, the dominant factor becomes the electric energy demands of the pumps and fans. The exemplary influence of adsorption share on the source of the electric energy demand of hybrid air-conditioning systems is presented in Fig. 13.

The heat demand of a hybrid air-conditioning system, with respect to its configuration and adsorption share, is presented in Fig. 14. Heat demand largely depends on adsorption share, the connection method used with the cooling water circuit, and on the selected type of cooling tower. Whether serial or parallel connections of the chillers to the cooling and chilled water circuits are applied seems to have no impact on the heat demands of the hybrid air-conditioning system. The curves maintain constant inclination for up to 60% of adsorption share. For the considered case of adsorption share above 60%, the adsorption chiller would operate under a transient heat load regime. The application of the hybrid air-conditiong system enables the utilization of up to 0.7–1.2 TJ of low-temperature heat with respect to the system's configuration. A system with a direct cooling water circuit with an open cooling tower permits

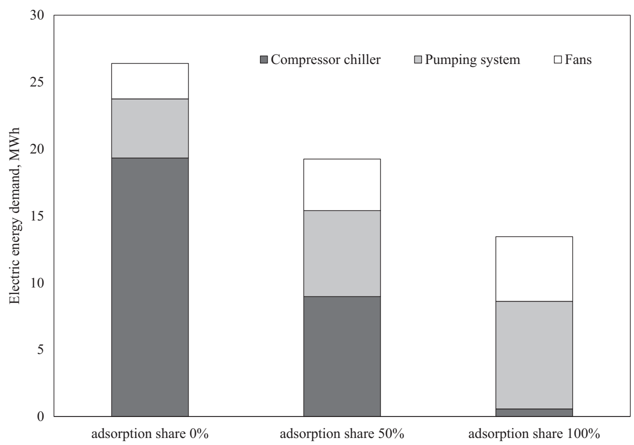

Fig. 13 – The influence of adsorption shares 0, 50 and 100% on the source of the electric energy demand of the hybrid air-conditioning system with parallel connections of the chillers to the cooling and chilled water circuits, the direct cooling water circuit, and the open cooling tower.

the utilization of around 0.7 TJ of heat annually. In the case of an indirect cooling water circuit with a recooler, the system will utilize twice as much heat.

6. Discussion

The electric energy demand decreases for high values of adsorption share. Nevertheless, this is still equivalent to roughly 50% of the electric energy demands for compressor air-conditioning systems (hybrid air-conditioning systems with 0% adsorption share), due to the high electric energy demands for pumps and fans. The authors recommend replacing thermal *COP* with total *COP* to describe the adsorption chillers' performance, as this will also take the electric energy demand into account. In the authors' opinion, total *COP* is a more reliable indicator of the adsorption chiller's performance, which was also shown by Daßler and Mittelbach (2012).

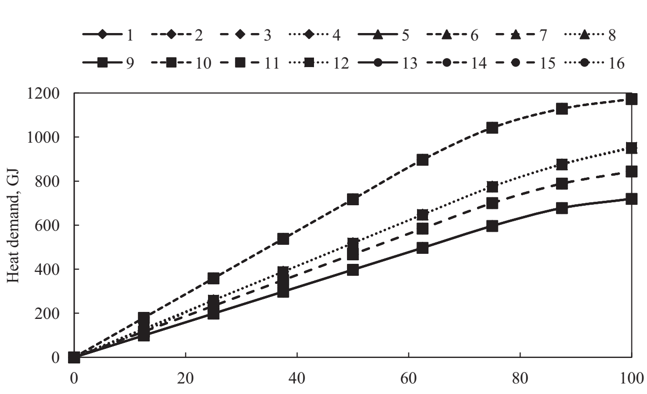

Fig. 14 – Total annual heat demand with respect to hybrid air-conditioning system configuration and adsorption share, for a heat load of up to 100 kW, the curves 1–16 as in Table 2.

The issue that must be considered during the selection of adsorption share is the size of the district heating connection. Adsorption share should not exceed 30% for an average size district heating connection. Otherwise, the district heating would not be able to provide a sufficient flow of heating water.

7. Conclusions

Adsorption chillers allow the conversion of low temperature heat into cold applicable in air-conditioning systems. Due to the control characteristics, high thermal inertia, the significant impact of the input parameters, as well as investment costs, the stand-alone application of adsorption chillers as commercial air-conditioning systems is not recommended. However, the application of this technology enables the efficient conversion of low-temperature heat (i.e. district heating, waste or solar heat) into cold during summer seasons. Therefore, the application of hybrid air-conditioning systems is recommended. A 100 kW hybrid air-conditioning system, with respect to its configuration, enables the utilization of 0.5 to 0.9 TJ of low-temperature heat per year, while simultaneously providing comfortable air-conditioning.

The performance of the hybrid air-conditioning system is strongly affected by its configuration, especially with respect to the types of cooling towers and cooling water circuits used. A direct cooling water circuit with an open cooling tower is a recommended solution. It enhances the performance of the hybrid air-conditioning systems and reduces operating costs by one half, with respect to the alternative configuration of using an indirect cooling water circuit with a recooler. Most commercially available chillers require the application of an indirect cooling water circuit and a recooler in order to maximize efficiency of the chiller. This goes against the optimization of adsorption-based air-conditioning systems. The development and optimization of adsorption chillers should not be in contradiction with the optimization of hybrid air-conditioning systems.

The second factor that has a significant impact on the performance of hybrid air-conditioning systems is adsorption share. An increase in adsorption share reduces operating costs and increases low-temperature heat conversion into cold. However, it also causes an increase in investment costs and the electric power demand of pumps and fans. In the authors' opinion, the adsorption share of hybrid air-conditioning systems should not exceed 50%, due to the resulting increase in investment costs and thermal inertia. Moreover, when the adsorption share is above 50%, the adsorption chiller would have to respond to the transient cooling power demands. As has previously been presented in the overview of research on adsorption chillers, such an operation has yet to be examined.

For low values of adsorption share, the parallel connection of chillers to the cooling and chilled water circuit is recommended. When adsorption share exceeds 50%, due to the increase of the electric energy demands of the pumps and fans, a serial connection of chillers can be implemented. This solution can also be introduced in cases of high pressure drop due to the size and location of the system.

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