Final Deliverable

Pumps Efficiency and Adaptability

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IEA SHC Task 48 / task48.iea-shc.org

IEA Solar Heating and Cooling Program

The Solar Heating and Cooling Programme was founded in 1977 as one of the first multilateral technology initiatives ("Implementing Agreements") of the International Energy Agency. Its mission is *"to enhance collective knowledge and application of solar heating and cooling through international collaboration to reach the goal set in the vision of solar thermal energy meeting 50% of low temperature heating and cooling demand by 2050.*

The member countries of the Programme collaborate on projects (referred to as "Tasks") in the field of research, development, demonstration (RD&D), and test methods for solar thermal energy and solar buildings.

A total of 53 such projects have been initiated to-date, 39 of which have been completed. Research topics include:

- Solar Space Heating and Water Heating (Tasks 14, 19, 26, 44)
- Solar Cooling (Tasks 25, 38, 48, 53)
- Solar Heat for Industrial or Agricultural Processes (Tasks 29, 33, 49)
- Solar District Heating (Tasks 7, 45)
- Solar Buildings/Architecture/Urban Planning (Tasks 8, 11, 12, 13, 20, 22, 23, 28, 37, 40, 41, 47, 51, 52)
- Solar Thermal & PV (Tasks 16, 35)
- Daylighting/Lighting (Tasks 21, 31, 50)
- Materials/Components for Solar Heating and Cooling (Tasks 2, 3, 6, 10, 18, 27, 39)
- Standards, Certification, and Test Methods (Tasks 14, 24, 34, 43)
- Resource Assessment (Tasks 1, 4, 5, 9, 17, 36, 46)
- Storage of Solar Heat (Tasks 7, 32, 42)

In addition to the project work, there are special activities:

- SHC International Conference on Solar Heating and Cooling for Buildings and Industry
- Solar Heat Worldwide – annual statistics publication
- Memorandum of Understanding with solar thermal trade organizations
- Workshops and conferences

Country Members

Australia	Germany	Singapore
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Executive Summary

Subtask A concentrates on developing tools and deliverables permitting to show the level of quality of the most critical components of the solar cooling and heating system. These components are mainly the chiller, the heat rejection device, the pumps and the solar collectors.

This technical report focuses on pump efficiency and adaptability to part load conditions in order to minimize the electricity consumption in the hydraulic circuits to obtain a high seasonal energy efficiency ratio in solar cooling systems.

In a first step a selection of market available chillers is evaluated by manufacturer design data concerning temperature differences, flow rates and pressure drops of the external hydraulic circuits and the resulting auxiliary energy electricity consumption to overcome the friction losses in the heat exchangers. While the EER for the chiller solely varies between 11.9 and 77.6 some market available chillers inherently impede good seasonal performance of the overall SHC-System.

Subsequently the different hydraulic circuits of several measured solar cooling systems are analyzed concerning their portion on the overall seasonal electricity consumption. Typically more than 50 % of the auxiliaries are caused by the heat rejection system including cooling water pumps and fan.

A short observation of the portion of pump costs in SHC-Systems confirms the almost negligible impact on overall investment costs and absence of meaningful cost-saving opportunities. Furthermore due to substantially reduced operation costs high-efficiency pumps help to reduce operational costs.

But the deployment of high-efficiency pumps in solar cooling installations does not implicate an efficient pumping automatically. The strong relationship between pump and plant curve demands a proper system design and pump selection.

The way things are an overall SEER of 20 for well-designed small scale solar cooling systems and more seems to be feasible.

Specific Objectives

A state of the art analysis will be conducted on this component in close cooperation with ongoing IEA-SHC Tasks 44 and 45, where these issues are tackled as well. Furthermore the design criterions of market available chillers concerning temperature levels and pressure drop in the heat exchangers are assessed.

In addition to that a performance coefficient called Auxiliary Energy Consumption Ratio (AECR) for the overall hydraulic efficiency is introduced in order to compare the design of various hydraulic circuits of SHC-systems in different capacity classes.

A short theoretical introduction into the rotodynamic pump design helps to avoid planning errors, adverse duty points and simplifies a correct pump selection.

A particular focus will be addressed to the adaptability of the technology to part load production conditions.

Finally an investigation will be done on the best practices for electric consumption reduction for pumping in the different hydraulic loops of a solar cooling system. Best practice will be valorized always including the compromise between efficiency and simplicity.

1. Abbreviations and Nomenclature

Abbreviations

AECR	Auxiliary energy consumption ratio
AC	Absorber, Adsorber & Condenser
BEP	Best efficiency point
BUS	Binary Unit System
COP	Coefficient of performance
D	Desorber
E	Evaporator
EuP	Energy using Products
EEI	Energy efficiency index
EER	Energy efficiency ratio
HQ	Head and flow
ICA	Instrumentation, control and automation
IEA	International energy agency
ISO	International Organization for Standardization
NPSH	Net positive suction heat
SEER	Seasonal energy efficiency ratio
SHC	Solar heating and cooling
SPF	Seasonal performance factor

Nomenclature

H	m	Pump Head
P_1	W	Electric to pump
P_2	W	Mechanical shaft power to pump
P_H	W	Hydraulic power
Q	m³/h	Pump flow
n	1/s	Rotation speed
H	-	Pump efficiency
η_M		Electric motor efficiency
η_{FC}		Frequency converter efficiency

2. List of Figures and Tables

3. State of the Art Analysis

In general transferring thermal energy in solar cooling systems from one component to the others demands a certain heat transfer medium flow depending on the specific heat capacity and temperature spread between supply and return flow. Due to irreversibility's a non-negligible auxiliary electric energy to overcome friction losses in the pipe work, fitting and heat exchangers is needed. In order to achieve high seasonal energy efficiency (SEER) the overall hydraulic power must be minimized and all pumps of the different hydraulic circuits must be designed to operate in Best Efficiency Point (BEP) under any load condition.

3.1. Pump efficiency

In general the efficiency of a centrifugal pump is defined as the ratio of the power exerted to the fluid in relation to the supplied electricity. This performance figure is not fixed for a given pump. Efficiency changes depend on discharge, operating head and size of the pump. While huge water pumps in power plants e.g. pumping hundreds of cubic meter reach high values up to 90 %, the overall efficiency of small pumps, as implemented in SHC-Systems, is less than 60 %. Due to improper hydraulic system design and inappropriate pump selection the overall electricity consumption of the solar-thermally driven sorption cooling and heating systems is sharply higher than essential. As a result the seasonal energy efficiency ratio SEER hangs considerable behind the expectations.

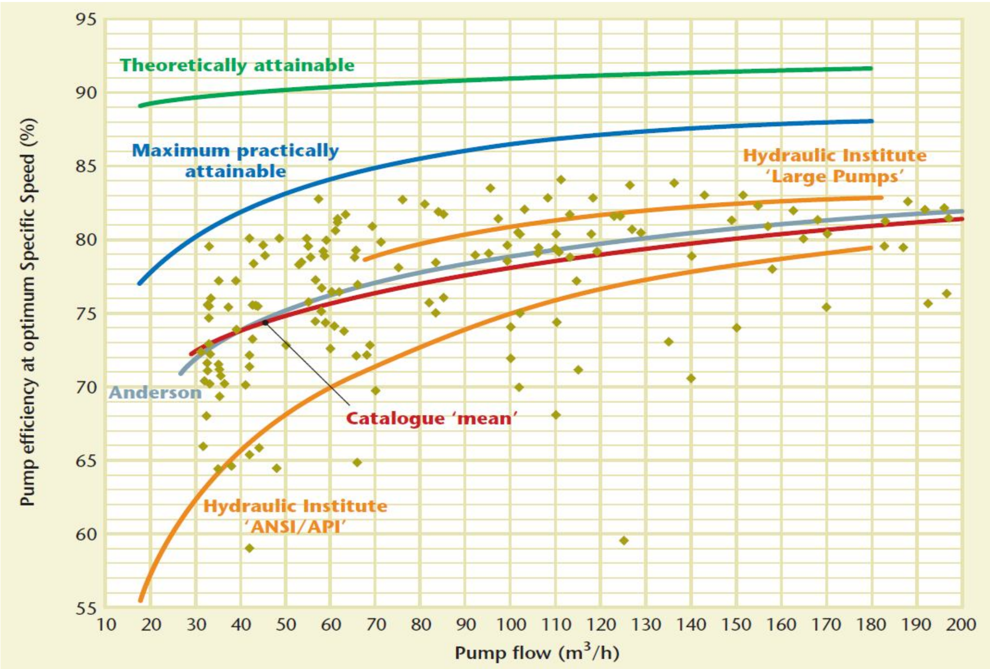


Figure 1: Pump efficiency at optimum specific speed for certain pump flows (Source: European Commission [1])

3.2. Thermodynamic and hydraulic design criteria of solar driven chillers

The following Table 1 shows a short overview about some market available hot water driven single stage Adsorption and Absorption chillers for solar Heating and Cooling Systems in the range from 8 to 100 kW chilled water capacity. Depending on the overall thermal Coefficient of Performance and the temperature difference between inlet and outlet of the chilled water, cooling water and hot water circuit a certain volume flow of the heat carrier medium is needed. Due to friction losses in the internal heat exchangers the pressure drop is not negligible and requires in some cases unacceptable high pumping effort and electricity consumption respectively.

Table 1: Hydraulic circuit design criteria of solar driven chillers and calculation example

						Driving heat circuit				Reject heat circuit				chilled water circuit					eta 40%		
Manufacture	Type	Process	Capacity	Heat	COP thermal	INLET	OUTLET	FLOW	Pressure drop	INLET	OUTLET	FLOW	Pressure drop	INLET	OUTLET	FLOW	Pressure drop	Hydraulic work	min. Electricity for pumps	Electricity internal	EER chiller
			kW	kW	-	°C	°C	m³/h	mbar	°C	°C	m³/h	mbar	°C	°C	m³/h	mbar	Watt	Watt	Watt	-
INVENSOR	HTC18vario	Adsorption Zeolite/H2O	18	34.6	0.52	85	76.5	3.6	310	27	34.5	6	290	18	14	3.9	300	111.8	279.6	20.0	60.1
	HTC18plus	Adsorption Zeolite/H2O	18	34.6	0.52				pump incl.				pump incl.				pump incl.				36.4
	LTC10vario	Adsorption Zeolite/H2O	10	16.7	0.60	72	66	2.5	220	27	31.5	5.1	260	18	15	2.9	170	65.8	164.5	20.0	54.2
	LTC10plus	Adsorption Zeolite/H2O	10	16.7	0.60	72	66	2.5	pump incl.	27	31.5	5.1	pump incl.	18	15	2.9	pump incl.			395.0	25.3
SORTECH	ACS15	Adsorption silicagel/H2O	15	26	0.58	7 K		3.2	260	5 K		7	440	3 K		4	500	164.2	410.6	14	35.3
	ACS08	Adsorption silicagel/H2O	8		#DIV/0!	7 K		1.6	230	5 K		3.7	350	3 K		2	300	62.9	157.2	7	48.7
MITSUBISHI P	AQSOA		9.8	21.8	0.45	70	65.1	3.84	275	32	37.2	7.62	698	16	11	1.69	423	196.9	492.3	36	18.5
AGO	100	Absorption H2O/NH3	100	217	0.46	105	82			25	30			1	-5					5570	18.0
	50	Absorption H2O/NH3	50	109	0.46	105	82			25	30			1	-5					4190	11.9
EAW	Wegracal SE 80	Absorption LiBr/H2O	83	111	0.75	86	71	6.4	70	27	32	33.4	400	15	9	12	70	406.9	1017.2	3400	18.8
	Wegracal SE 50	Absorption LiBr/H2O	54	72	0.75	86	71	4.1	50	27	32	22	450	15	9	7.7	65	294.6	736.5	3400	15.9
	Wegracal SE 30	Absorption LiBr/H2O	30	40	0.75	90	80	3.5	400	30	35	12	500	17	11	4.3	400	253.3	633.3	500	26.5
	Wegracal SE 15	Absorption LiBr/H2O	15	21	0.71	90	80	1.8	400	30	35	5	900	17	11	1.9	400	166.1	415.3	300	21.0
PINK	PC19 Minus	Absorption H2O/NH3	12.3	26	0.47	95	88	3.2	640	24	30	5.5	270	0	-3	3.5	120	109.8	274.5	450	17.0
	PC19 Fan-coils	Absorption H2O/NH3	18.6	30	0.62	85	78	3.6	680	24	30	6.9	440	12	6	2.7	65	157.2	393.0	450	5.5
	PC19 act.ceilings	Absorption H2O/NH3	19.5	27	0.72	75	68	3.3	650	24	30	6.7	410	18	15	5.6	280	179.4	448.6	450	21.7
HUIN	RXZ-58	Absorption LiBr/H2O	58	82	0.71	90	85	14.3	500	30		25	500	15	10	10	400	656.9	1642.4	300	29.9
	RXZ-35	Absorption LiBr/H2O	35	49	0.71	90	85	8.6	400	30		15	400	15	10	6	300	312.2	780.6	300	32.4
	RXZ-23	Absorption LiBr/H2O	23	33	0.70	90	85	5.8	400	30		10	400	15	10	4	300	208.9	522.2	300	28.0
	RXZ-11	Absorption LiBr/H2O	11	16.5	0.67	90	85	2.9	800	30		5	500	15	10	2	600	167.2	418.1	150	19.4
THERMAX	LT-2	Absorption LiBr/H2O	70	100	0.70	90.6	85	15.7	200	29.4	36.7	20	300	12.2	6.7	11	590	434.2	1085.4	600	41.5
	LT-1	Absorption LiBr/H2O	35	50	0.70	90.6	85	7.8	120	29.4	36.8	10	120	12.2	6.7	5.5	680	163.2	408.1	600	34.7
SAKURA	SHL010	Absorption LiBr/H2O	35.2	49.3	0.714	88	83	8.4	60	31	36.5	13.13	300	13	8	6	270	168.4	421.0	180	58.6
	SHL008	Absorption LiBr/H2O	26.1	36.45	0.716	88	83	6.7	60	31	36.5	10.51	320	13	8	4.8	250	137.9	344.8	180	49.7
	SHL005	Absorption LiBr/H2O	17.6	24.65	0.714	88	83	4.2	30	31	36.5	6.57	140	13	8	3	260	50.7	126.8	100	77.6
	SHL003	Absorption LiBr/H2O	10.5	14.58	0.72	88	83	2.5	30	31	36.5	3.94	130	13	8	1.8	210	26.8	67.0	100	62.9
Dummy	Test11	not defined	10	16.67	0.6	90	85	2.871	1060	37	42	4.593	1050	13	10	2.871	550	262.4	655.9	100	13.2
Dummy	Test22	not defined	10	16.67	0.6	90	85	2.871	1060	37	45	2.871	420	13	10	2.871	550	161.9	404.7	100	19.8
Dummy	Test33	not defined	10	16.67	0.6	90	85	2.871	1060	37	45	2.871	420	15	10	1.722	200	127.6	319.0	100	23.9
Dummy	Test44	not defined	10	16.67	0.6	90	80	1.435	260	37	45	2.871	420	15	10	1.722	200	53.4	133.6	100	42.8
Dummy	Test55	not defined	10	12.5	0.8	90	80	1.077	150	37	45	2.422	300	15	10	1.722	200	34.2	85.6	100	53.9
YAZAKI	WFC-SC30	Absorption LiBr/H2O	105	151.2	0.69	88	83	35.9	604	31	35	55.1	464	12.5	7	16.5	701	1633.8	4084.5	310	23.9
	WFC-SC20	Absorption LiBr/H2O	70	100.8	0.69	88	83	17.28	464	31	35	36.7	453	12.5	7	11	658	885.6	2214.0	260	28.3
	WFC-SC10	Absorption LiBr/H2O	35	50.2	0.70	88	83	8.64	904	31	35	18.4	853	12.5	7	5.5	561	738.6	1846.6	210	17.0
	WFC-SC05	Absorption LiBr/H2O	17.5	25.1	0.70	88	83	4.32	770	31	35	8.2	383	12.5	7	2.77	526	220.1	550.3	48	29.3

On the basis of manufacturer data the unavoidable hydraulic power in the three main circuits to overcome the friction losses in the heat exchangers is calculated to estimate the minimum overall electricity consumption for pumping assuming a pump efficiency of 40 %. Taking account of the internal electricity consumption of the chiller (control, refrigerant/sorbens pump etc.) a maximum possible EER for the chiller solely, neglecting the auxiliary energy consumption for solar collector circuit and reject heat dissipation, can be calculated.

The ratio between chilled water capacity and electricity consumption vary between 11.9 and 77.6. As a result the hydraulic design requirements of some market available chillers impede good seasonal performance. By way of illustration a notional chiller manufacturer named dummy offers the same machine for different operational conditions. Chiller Test11 requires very small temperature differences between Inlet and Outlet in the hydraulic circuits whereby the necessary flow of the heat carrier medium and the resulting pressure drop is rather high. By stepwise increasing the temperature differences in the hydraulic circuits a significant increase in EER is obtained. Test 55 shows a very good thermo-hydraulic design with promising temperature levels for widespread applications under European climatic conditions. Compared to Test44 the additional improved thermal COP leads to further significant improvement in EER, due to reduced energy demand and heat rejection effort.

Beside overall pump efficiency and thermal design of the chiller another important aspect is given by the adaptability to part load conditions. Most of the SHC-installations operate mainly under part load conditions depending on climatic conditions and load profile of the building. If there's no flow adaption by adjusting pump speed applied, the overall auxiliary electricity energy consumption remains constant. E.g. if the chilled water capacity of Test55 chiller is reduced by 50 % to 5 kW, without adjusting the flows of the heat carrier medium, the EER drops from 54 to 27 accordingly. On the other hand a dynamic adaption of the pump speed related to the current load of the chiller allows theoretically a cubical reduction of auxiliary energy consumption based on the affinity laws.

The way things are an overall SEER for well-designed small scale solar cooling systems of 20 seems to be feasible. However, currently many manufacturers mainly focus on cheap chillers than good seasonal system performance. As the thermodynamic and hydraulic design criteria of the chiller defines the capacities, temperature levels and/or heat transfer medium flows in the entire solar cooling system the following criteria's should be taken into account:

- Possibility of heat transfer medium flow reduction in the chilled water, cooling water and driving heat circuit to reduce the electricity consumption in part load conditions (some chillers have installed flow detectors at the entrance of each component which shut down the machine automatically at e.g. 80 % of the nominal flow. The same difficulty may occur by using open wet cooling towers for heat rejection. Since the cooling water is sprayed directly into the column, this nozzles often require a certain minimal heat carrier medium flow and pressure to ensure a satisfying operation of the wet cooling tower. At any case if not mentioned in the manual
- High thermal COP in a wide capacity range to diminish the driving heat and reject heat flow to and from the machine. This is a crucial aspect as the thermal COP influences both the hydraulic design of cooling water and driving heat circuit as well as the capacity of the solar collector field and heat rejection device. The higher the thermal COP the less driving heat and therefore pumping effort in the collector area is necessary. As less driving heat is required to produce the same amount of chilled water the thermal discharge a ventilation effort is reduced accordingly.
- High temperature spread in the main hydraulic circuits reduces significantly the heat transfer medium flow per thermal unit and the required piping diameters.
- High reject heat temperatures facilitate the heat dissipation to the ambient and allow for dry heat rejection as well. By avoiding a wet or hybrid cooling tower and its associated electricity consuming water make-up and preparation system (which is often incorrectly not taken into account) a further reduction of auxiliaries and fresh-water consumption is achieved.
- Low pressure drop in the heat exchangers of Evaporator, Condenser, Ad-/Absorber and Desorber. In this context a very well balanced compromise between sufficient heat transfer coefficient and

pressure drop as well as turbulent and laminar flow through the heat exchangers has to be targeted.

- Low internal electricity consumption (reduces the available electricity consumption for pumping and heat dissipation). While Adsorption machines are at an advantage over the absorption machines due to their discontinuous process without any internal refrigeration or solution pump, the overall performance suffers under the far smaller thermal COP.

Another significant effect on EER and system performance constitutes the heat rejection to the ambient by dry, wet or hybrid cooling towers. This difficulty context is described in Working group A3.

3.3. Hydraulic circuits and auxiliary electricity consumption in SHC-systems

There is a multitude of design and performance assessment tools as well as graphical circuit diagrams to describe the general arrangement and connection of the main components in solar heating and cooling systems. The source/sink Approach developed in Task44 and the Monitoring Procedure of Task 38 e.g. outline energy flows between the components without any information about hydraulics and ICA equipment while the “Generic System schemes” [2] of Task 38 condenses technical circuit diagrams based on ISO 1219-2:2012 to sub systems with reduced piping and instrumentation.

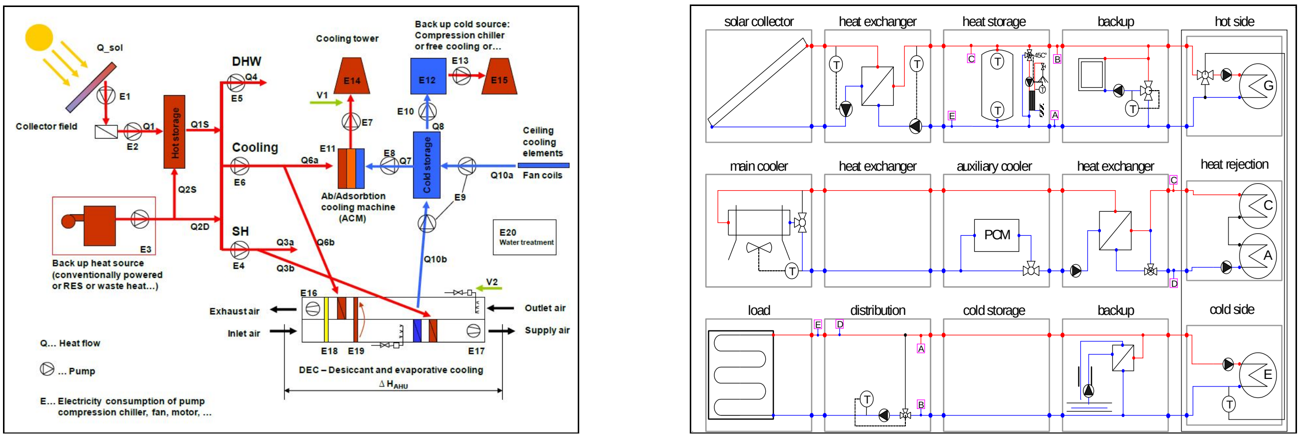


Figure 2: Task38 monitoring procedure (left) and generic system schemes (right)

In general the components of a solar cooling system can be classified into the following sub systems in order to identify their portion on the overall electricity consumption without excessive measuring equipment and detailed analysis of the hydraulic components. The ratio between transferred thermal unit from a heat source to a heat sink and the therefore required electricity is defined as Auxiliary Energy Consumption Ratio (AECR) and allows a direct comparison and validation of the hydraulic efficiency among different installations. The AECR includes both hydraulic circuit design and electrical efficiency of the pumps.

- Solar collector sub-system ($AECR_{Solar}$)

  This hydraulic circuit connects the flat plate, evacuated tube or concentrating solar thermal collectors to a heat storage or directly to the Desorber of the chiller and is defined as follows.

$$AECR_{Solar} = \frac{Q_{SolarHeat}}{\sum_{i=1}^{n} P_{SolarCollectorCircuit}} \qquad (1)$$

- Backup heat sub-system ($AECR_{Backup}$)

  This circuit includes additional back up heat sources e.g. fossil - biofuel boiler, district heating or any waste heat source feeding the heat storage or Desorber of the chiller.

$$AECR_{Backup} = \frac{Q_{BackupHeat}}{\sum_{i=1}^{n} P_{Auxiliaries}} \tag{2}$$

- Driving heat sub-system ($AECR_D$)

  If a heat storage is installed between heat source and Desorber this additional hydraulic circuit is mandatory.

$$AECR_D = \frac{Q_{Desorber}}{\sum_{i=1}^{n} P_{DrivingHeatCircuit}} \tag{3}$$

- Heat rejection sub-system ($AECR_{AC}$)

  The reject heat from Condenser and Ab-/Adsorber has to be transferred to a heat sink or any other heat consumer.

$$AECR_{AC} = \frac{Q_{Absorber\,/\,Condenser}}{\sum_{i=1}^{n} P_{CoolingWaterCircuit}} \tag{4}$$

- Cooling tower sub-system ($AECR_{Cooler}$)

  This subsystem includes the electricity consumption for water treatment, spraying, ventilation ASO.

$$AECR_{Cooler} = \frac{Q_{Cooler}}{\sum_{i=1}^{n} P_{Fan+Auxiliaries}} \tag{5}$$

- Chilled water sub-system ($AECR_E$)

  The chilled water has to be transferred from the Evaporator of the Chiller to a cold storage or directly to the distribution system of the building and considers at least the hydraulic power to overcome the friction losses in the Evaporator.

$$AECR_E = \frac{Q_{Evaporator}}{\sum_{i=1}^{n} P_{ChilledWaterCircuit}} \tag{6}$$

- Chiller internal sub-system ($AECR_{chiller}$)

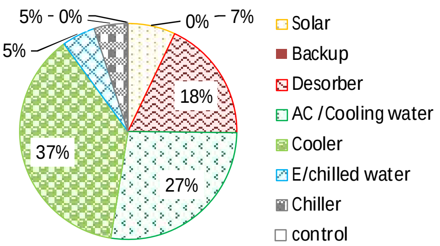
Depending on the chiller type more or less electricity is consumed in the machine itself to maintain the sorption process and vacuum.

$$AECR_{chiller} = \frac{Q_{Evaporator}}{\sum_{i=1}^{n} P_{Auxiliaries}} \quad (7)$$

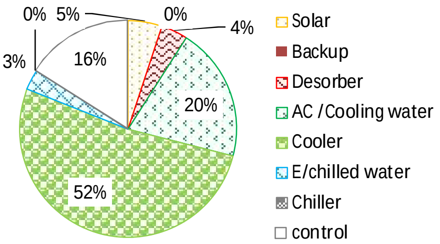
Based on this classification several solar cooling systems are analyzed concerning their electricity consumption in the hydraulic circuit, cooling tower and internal sorption process as well as its control and measuring system. The pie charts in Table 2 show the percentage of electricity consumption in the different sub-systems. Measuring results from several national Research projects (SolarCoolingMonitor / SolarCoolingOpt / Roccoco / SolarRück and Annex34 are listed.

Table 2: Proportion of electricity consumption in the different sub-systems

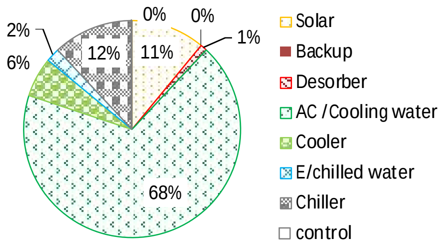
Project:	**SOLID Coolcabin**
Location:	Graz, Austria
Type:	17.5 kW LiBr/Water Absorption chiller
Brand:	Yazaki
SEER:	~ 4



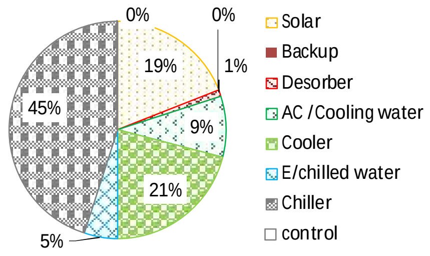
Project:	**MA34**
Location:	Vienna, Austria
Type:	7.5 kW Silicagel/Water Adsorption chiller
Brand:	Sortech
SEER:	~ 6



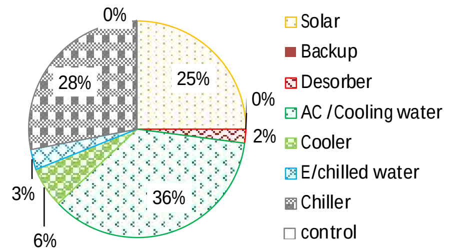
Project:	**BH Rohrbach**
Location:	Rohrbach, Austria
Type:	30 kW LiBr/Water Absorption chiller
Brand:	EAW
SEER:	~ 4,2



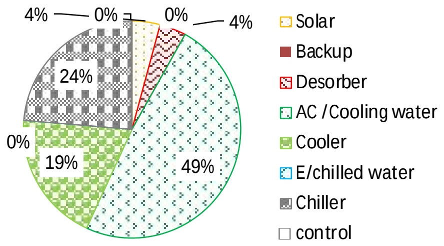
Project: **Sun Master / Xolar**
Location: Rohrbach, Austria
Type: 80 kW LiBr/Water Absorption chiller
Brand: EAW
SEER: ~ 6,7



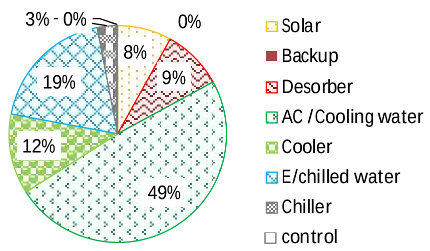
Project: **Gasokol**
Location: Saxen, Austria
Type: 30 kW LiBr/Water Absorption chiller
Brand: EAW
SEER: ~ 4,5



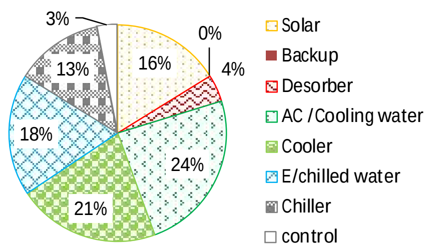
Project: **Feistritzwerke**
Location: Gleisdorf, Austria
Type: 19 kW NH3/Water Absorption chiller
Brand: PINK
SEER: ~ 5



Project: **Caixa Geral de Depositos (CGD)**
Location: Lisbon, Portugal
Type: 545 kW LiBr/Water Absorption chiller
Brand: BINGSHAN
SEER: ~ 5



Project: **SolarHeatCool+PCM**
Location: Garching, Germany
Type: 10 kW LiBr/Water Absorption chiller
Brand: SK Sonnenklima
SEER: ~ 11

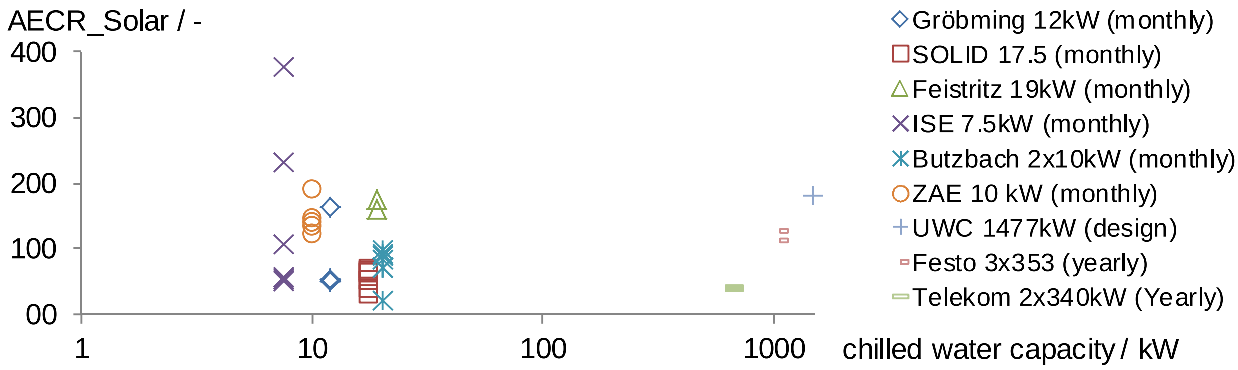


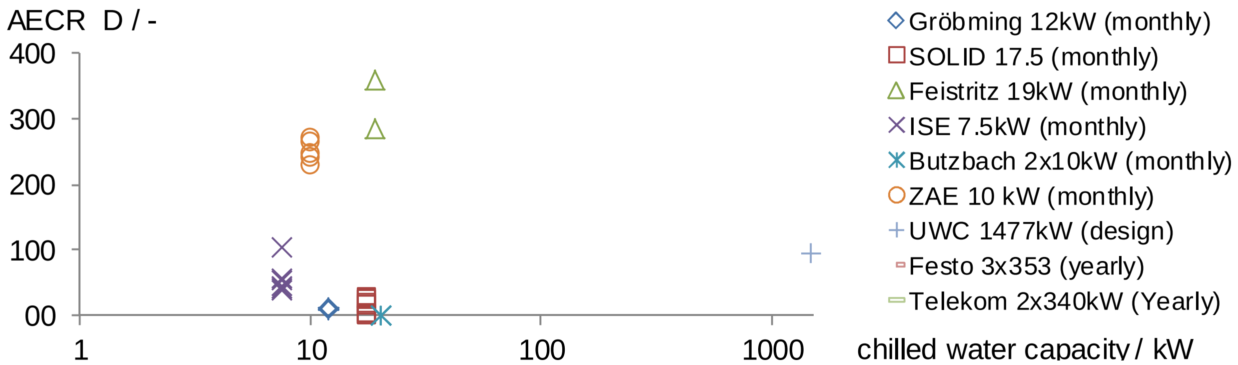
Conclusion:

In most of the SHC-systems the electricity consumption of the heat dissipation system is dominating! More than 50 % of the auxiliary energy is consumed by the cooling water pumps, fans and water make-up and preparation device. In general these Pie diagramms show best saving options for the evaluated systems and preferred application of speed controlled pumps. But on the other hand a direct comparison of system designs among different SHC installations is only possible with AECR values.

Figure 3 shows some selected solar cooling installations with either absorption or adsorption technology in a chilled water capacity range starting from 7.5 kw up to about 1.5 MW. Most of the installations show a good ratio between transferred heat in the hydraulic circuits between solar collector array and buffertank (AECR_Solar) as well as subsequent heat supply to the chiller (AECR_D).

Due to either huge heat quantity and/or small temperature difference the high specific auxiliary energy consumption in the chilled and cooling water circuit requires a proper and efficient subsystem design.





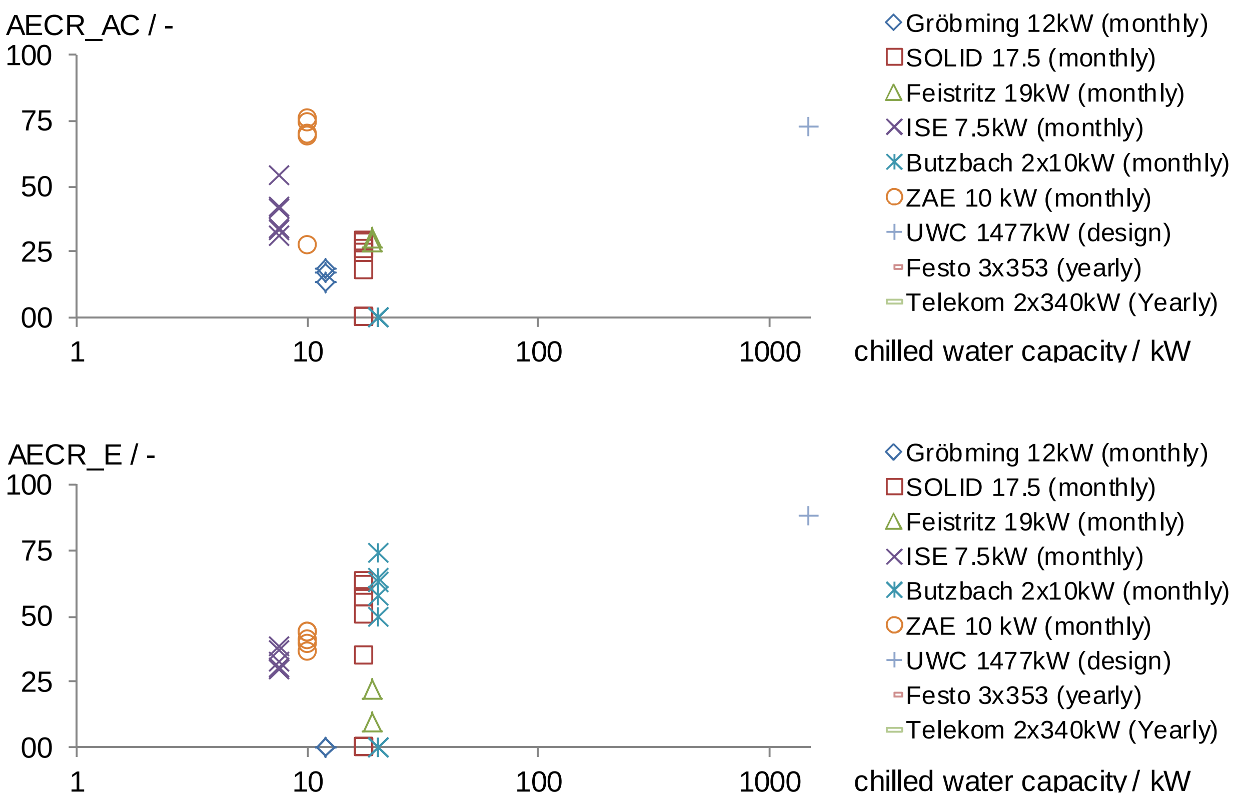


Figure 3: Auxiliary Energy Consumption Ratios in the main hydraulic circuits solar array, Heat supply to chiller, cooling and chilled water of some SHC-systems

3.4. Specific installation costs for pumps

This section identifies the cost-sensitiveness and portion of costs regarding pumps in relation to the total investment of solar cooling applications.

In general, expenses for hydraulic circuits and installation account for one third of the cumulative costs as shown in Figure 5. Another third has to be spent on the solar thermal collectors. The chiller itself reflects in the budget about 15 to 20 %. Electrical wiring and control units cause minor overall costs of about 10 %. This investigation had been performed in the EU-Project ROCOCO [3] for medium cooling capacities between 70 – 180 kW and could be confirmed by several other projects as well.

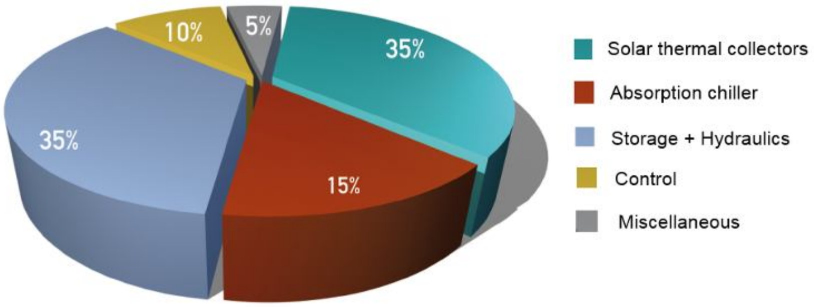


Figure 4: Characteristic cost distribution in SHC systems with absorption chillers.(Source: [3])

Although heat transfer fluid circuit installation contributes to the overall SHC system costs in a large part, the single investment for pumps itself account for less than 5 % in general as shown exemplary in Figure 5 or described for another 5 thermal cooling installations by D. Schall, S. Hirzel [4].

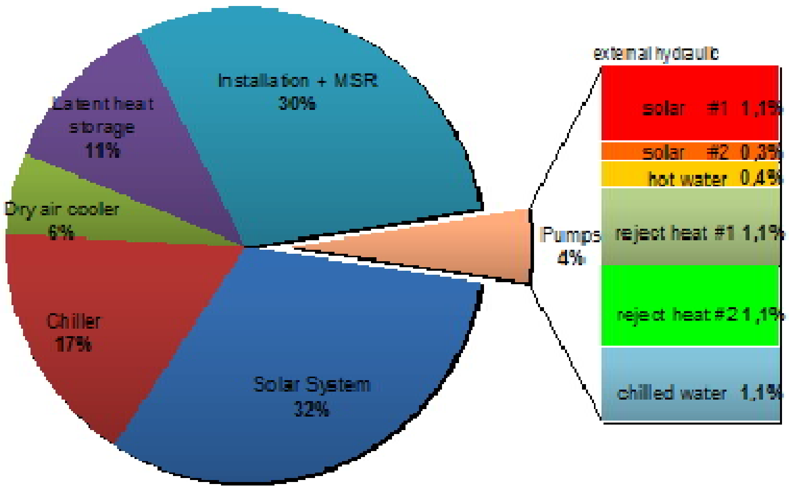


Figure 5: Investment costs in a 10 kW absorption cooling system with dry heat rejection

Conclusion:

A life cycle costs calculation of pumps comprises among other things costs for purchase, installation as well as commissioning, energy consumption during lifetime and maintenance. As prices, operation hours and energy costs vary significantly among SHC-systems and even countries; a comprehensive statement is therefore not possible. Even though investment costs of a variable speed high-efficiency pumps often are twice as high compared to a cheap but inefficient fixed speed pump the overall costs for pumps in SHC systems are rather small or often negligible. But adjusting pump speed to part load conditions has a high impact on system efficiency regarding EER. Nevertheless in most of the cases, especially in installation with high operational hours, the pay back is achieved through auxiliary energy savings in a reasonable period of time.

4. Pump Efficiency - Theoretical -

This chapter gives a short overview about the theory of rotodynamic pumps and crucial design criteria as well as international legislative regulations and shows some real operating points of pumps in solar cooling applications. Figure 6 shows the classification of hydraulic pumps. As almost exclusively kinetic centrifugal volute radial impeller pumps are applied in SHC-Systems this chapter focus only on this pump type.

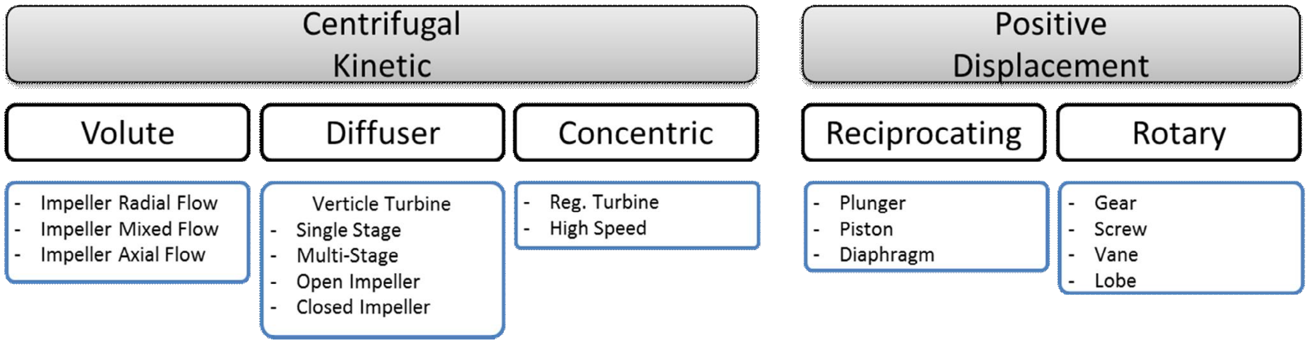


Figure 6: Classification of pump types

In recent years a high increase in the overall efficiency of roto-dynamic pumps had been obtained due to international regulations and standardization such as the EuP-Directive (Directive 2009/125/EC) of 2009, the Energy Efficiency Index EEI involving canned single circulator pumps with less than 2500 W electricity consumption and the new International Efficiency classes of motors (IE) concerning low-voltage three phase asynchronous motors from 0.75 kW to 375 kW. [5]

Table 3: Predominant rotodynamic pump types, canned and shafted, installed in SHC systems

canned rotodynamic pumps	**shafted rotodynamic pumps**
(only single stage)	single and multistage
Millions of units (2005: 120 million installed in residential heating systems)	Multiple thousands especially in industrial processes
up to 15 m head, max. volume ~ 50 m³/h	“No limit”

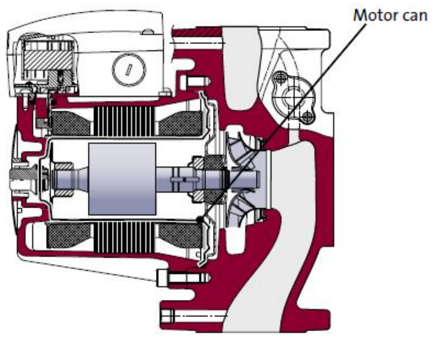


Figure 7: Sectional view of a canned rotodynamic pump. Source: Pump handbook [6]

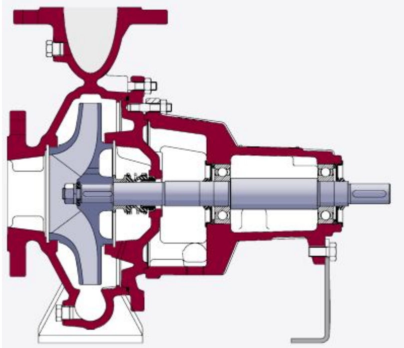


Figure 8: Sectional view of a shafted rotodynamic pump. Source: Pump handbook [6]

Centrifugal pumps perform work by manipulation the velocity of the fluid passing through the impeller of the pump from the inner diameter (ID) to the outer diameter (OD). In the following diffusor dynamic pressure of the fluid is converted into static pressure by flow cross-section expansion.

- The Affinity laws and their effect on part load conditions [7,8,9]

  The affinity laws describe the theoretical dependency of rotation speed, head and power consumption of rotary pumps for ideal, frictionless and incompressible flow. In good approximation they describe the real behavior of speed-controlled rotary pumps.

  Equation 9 describes the proportion relation between flow Q and rotation speed n of the pump.

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1} \tag{9}$$

  Linking equation 9 and 10 shows the quadratic relationship between flow and pump head H which is equal to the pressure drop in the hydraulic circuit.

$$\frac{H_2}{H_1} = \left(\frac{n_2}{n_1}\right)^2 \tag{10}$$

  Referred to the power consumption P of the pump a flow or rotation speed reduction has a cubical influence.

$$\frac{P_2}{P_1} = \left(\frac{n_2}{n_1}\right)^3 \tag{11}$$

Furthermore the overall efficiency and power consumption of pumps is influenced by a lot of factors.

- The impeller velocity
- The impeller diameter (trim limited by 20 %)
- The number of blades on the impeller
- The diameter of the eye of the impeller
- The thickness of the impeller
- The pitch of the blades
- Surface finish of internal surfaces
- Wear ring tolerance (tremendous effect on pumps with low specific speed)
- Mechanical losses (Bearings, seals, packings …)
- Viscosity of the heat transfer medium (Head remains the same, but enhances power input)

As shown prior in Figure 1, current rotodynamic pumps feature excellent performance up to 80 % and more in the high capacity range, but efficiency drops below 50% and less when smaller pumps have to be used.

In general, characteristic pump QH-curves are designed in accordance to ISO 9906 Rotodynamic pumps – Hydraulic performance acceptance tests – Grades 1,2 and3, in which tolerances are specified as following:

Flow	Q	± 9 %	m³/h or l/s or l/min
Head	H	± 7%	m
Electrical Power	P_1	± 9%	W or kW
Efficiency	η	- 7%	-

QH-curve:

This typical graph shows the characteristic QH-curve of a specific pump in terms of flow Q_1 and discharge head H_1 in meter. Low flows allow for high discharge pressure. With decreasing pressure difference between inlet and outlet of the pump the flow increases while available head bottoms out.

The pump characteristic line represents one revolution speed. If a speed converter is applied this line slides down in parallel. A minimum revolution is required for canned pumps to ensure adequate motor cooling by the fluid flowing through the pump. In general a modulation rate of 1:4 in flow is available.

The system characteristic or pressure drop respectively increases quadratic starting at zero in case of a closed hydraulic system. The duty point, the match between pump and system characteristic should be designed next or slightly right to the Best Efficiency Point (BEP) of the rotodynamic pump.

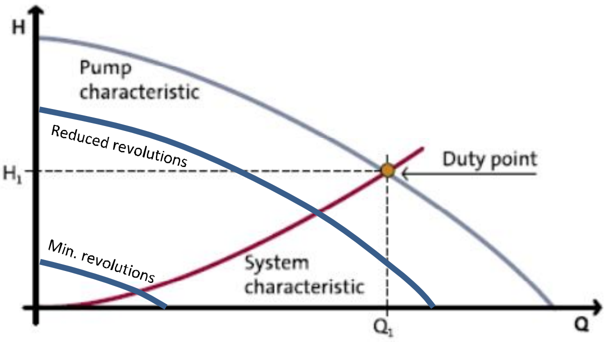


Figure 9: QH map and system characteristic of a pump system. Based on: Pump handbook [6]

Figure 10: QH map and system characteristic of a pump system. Based on:Pump handbook [6]

Efficiency, the η curve

This important value η gives the ratio between hydraulic power P_H delivered to the heat carrier medium and the required either mechanical power at the shaft P_2 (shafted rotodynamic pumps) or electrical power P_1 to the pump (canned rotodynamic pumps). Important fact, if a shafted rotodynamic pump is applied the overall efficiency of the electric motor η_M has to be taken into account additionally.

$$\eta = \frac{P_H}{P_1} = \eta_M \frac{P_H}{P_2} = \eta_{FC} \cdot \eta_M \frac{P_H}{P_2} \qquad (8)$$

Furthermore if a frequency converter is used, its efficiency η_{FC} has to be incorporated as well.

Efficiency reaches its maximum in the so called BEP Best Efficiency Point for a single flow and head for which the pump was conceived. The duty point with most operating hours of the pump should match this point in order to get a good system performance over a wide range of flow conditions.

A general approach or rule of thumb to determine the efficiency of a hydraulic pump quickly is given in equation 8, where Q represents the flow of the heat carrier medium in m³/h and H the pump head in meter or converted pressure loss in the hydraulic loop respectively. Beside viscosity especially the density ρ in kg/m³ of the fluid affects electric power consumption P_1 in Watt and the overall efficiency η.

$$\eta = \frac{Q \cdot H \cdot \rho}{367 \cdot P_1} \qquad (8)$$

Vice versa, the transformed equation 9 allows simple electricity consumption predication P_1 in Watt if flow Q in m³/h, pressure loss in the system p in millibar and overall pump efficiency η is known.

$$P_1 = \frac{Q \cdot p}{36 \cdot \eta} \qquad (9)$$

Electric power consumption, P_1 – curve

Most of the power consumption curves are similar to the given one in Figure 11. There's one very import aspect to be watched out. While for canned pumps in general (but not always) P_1 is shown, shafted always give P_2 as power input to the crank.

P_1 Electricity consumed by whole pumping system including Inverter, circuit box, electric motor a.s.o

P_2 Mechanical power to the shaft of the pump (shaft power)

P_H Hydraulic power performed to the fluid (useful energy)

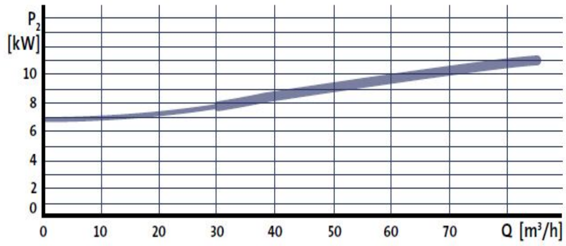


Figure 11: Electric power consumption curve [6]

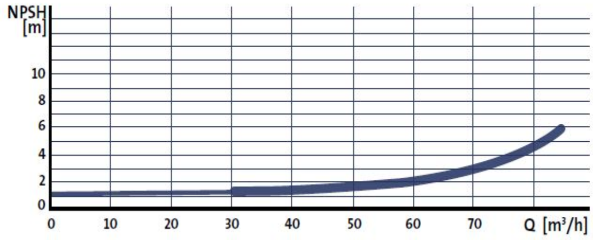


Figure 12: Net positive Suction Head NPSH [6]

NPSH-curve (Net positive Suction Head) and Cavitation

This value is crucial for a durable and safe operation of any pump. Especially at high temperatures e.g. Solar collector or chiller driving heat circuit at high flow conditions (increasing dynamic pressure) the static pressure at the inlet of the volute has to be higher than the boiling temperature of the heat carrier medium to prevent cavitation. Cavitation occurs somewhere at the back of the impeller vanes when the local static pressure is lower than the vapor pressure of the liquid. It results in reduced efficiency on the one hand, but more important might damage the pump over a longer period of time. Sufficient system pressure and a short link with low pressure loss between suction side and expansion vessel are very important design criteria.

5. Pump Adaptability and Control - Practical -

This section deals with advanced control strategies and the adaptability of pumps to part load conditions, in order to improve the overall system performance. As variable speed control is the only choice for efficient and dynamic pump adaption to part load conditions, outdated methods like throttle or bypass control are not discussed.

5.1. Communication and control

A simple way is to switch the power ON and OFF. In some cases older high-efficiency pumps and switching relays have been destroyed due to the high initial current during the first Milliseconds to charge capacitors. In so doing, a big advantage is to avoid standby electricity consumption caused by the electronic of high-efficiency pumps or frequency converters.

If a speed control has to be applied most of the canned high-efficiency pumps allow 0-10 Volt or PWM signals for rotary speed adjustment and a dry contact for start/stop commands. In most of the canned high-efficiency pumps a part load adoption by constant or proportional differential pressure measurement is already implemented.

The next generation of high-efficiency pumps already feature a BUS-Communication module, allowing bidirectional communication and internal pump sensor detection for temperature regulated speed adjustment or heat metering.

Communication with Pumps via bidirectional BUS-Communication e.g. via GeniBUS protocol (provided by GRUNDFOS) or CANopen (provided by WILO)

Main advantages:

+ Reduced Wiring (Only BUS cable and Power Cable needed)
+ Free communication protocol (RS485)
+ Integrated measuring equipment provides additional data (Flow, head, speed, electricity consumption, temperature…) for part load adaption and performance evaluation
+ “Intelligent pumps” might replace most of the measuring equipment needed

Main problems:

- Possible but complex and not recommended for plumbers on-site
- Some measuring values are calculated and therefore not precise under extreme part load conditions
- Standby electricity consumption is increased

By now, intelligent high-efficiency pumps offer a great opportunity for advanced system evaluation at concurrently reduced measuring equipment. Due to its complexness and various interactions between several components, implementation will be reserved for manufacturers or prefabricated system providers.

5.2. Pump control strategies

In this section different control strategies and the adaptation to part load conditions based on operational experience and system simulations are discussed briefly. A plus indicates an advantage either on system performance or simplified design, whereas a minus lists all negative aspects.

- Solar Primary Pump
  - Constant speed
    + simple control signal ON/OFF
    + ensures constant heat transfer values in the heat exchangers and collector
    - decreases EER in part load conditions significantly
  - rotary speed control to assure a constant OUTLET temperature at the solar collectors
    + constant driving heat temperature to chiller
    + increases EER in part load conditions
    + best choice if collector array feeds directly the heat buffer
    - might suffer under oscillation behavior in connection with corresponding solar secondary pump control strategy
  - rotary speed control in relation to the direct insolation on the aperture area
    + direct and independent reference to available solar heat
    + no time delay or dead time
    + ensures constant temperature difference
    - additional insolation sensor needed
- Solar Secondary Pump (obsolete if freeze protection is not in question)
  - Constant
    + simple control signal ON/OFF
    + ensures constant alpha values in the heat exchanger
    - decreases EER in part load conditions
    - variable driving heat temperature leads to inefficient sorption process
  - rotary speed control to assure a constant OUTLET temperature of the solar collectors
    + ensures constant buffer tank and driving heat temperatures
    + increases EER in part load conditions
  - rotary speed control related to mass flow in primary solar collector circuit
    - leads to oscillating behavior

Driving Heat Pump

- Constant
  + simple control signal ON/OFF
  + ensures constant alpha values in the heat exchanger
  - decreases EER in part load conditions
- rotary speed control related to the chilled water capacity
  - Capacity metering needed
  + increases EER in part load conditions)

- rotary speed control related to driving heat temperature
  - oscillating behavior in combination with driving heat mixing valve

➢ Primary Reject Heat Pump

- Constant
  + simple control signal ON/OFF
  + ensures constant alpha values in the heat exchanger
  - decreases EER in part load conditions
- rotary speed control to assure a constant OUTLET temperature of cooling water leaving the sorption machine
  - might interact with cooling tower fan and influence chilled water capacity as well as driving heat regulation
- rotary speed controlled in relation to the chilled water capacity
  - Capacity metering needed
  + increases EER in part load conditions

➢ Secondary Reject Heat Pump

- Constant
  + simple control signal ON/OFF
  + ensures constant alpha values in the heat exchanger
  - decreases EER in part load conditions
- rotary speed control in relation to the control signal of the primary reject heat pump
  + increases EER in part load conditions
  + simple control signal
  - sometimes a mismatch occurs if pump performance curves and hydraulic characteristic differs in both circuits
- rotary speed controlled in relation to the primary rejected heat circuit flow
  + increases EER in part load conditions (best choice)

➢ Chilled Water Pump

- Constant
  + simple control signal ON/OFF
  + ensures constant alpha values in the heat exchanger
  - decreases EER in part load conditions
  - may result in improper chilled water temperatures
- rotary speed control related to differential pressure drop at the cold distributor
  + increases EER in part load conditions
- rotary speed controlled related to chilled water mass flow needed
  + increases EER in part load conditions
  - additional sensor required

- rotary speed controlled to ensure a constant OUTLET temperature
  - may affect the capacity control of the chiller

Basically, if a variable speed control shall be applied, first clarify minimum or required flow-rates through the components in consultation with the manufacturer. Especially the sorption machine often requires a minimum flow in the chilled water circuit. Moreover some open wet cooling towers demand a certain pressure at the distribution nozzles.

5.3. Operating points and overall efficiency under real operating conditions

The deployment of high-efficiency pumps in solar cooling installations does not implicate an efficient pumping automatically. In this section the operating point of rotodynamic pumps in the hydraulic circuits of two real applications are analyzed and discussed.

First SHC system example compromises a 7.5 kW Adsorption chiller with dry heat rejection plus water injection system located in Perpignan, France. A collector aperture surface of 25 m² provides hot water which is fed into a buffer by means of the solar primary pump (Grundfos TP25-90/2). This shafted high-efficiency pump possesses a maximum efficiency of 50 % in its best operating point (BEP) as shown in Figure 13. Due to high pressure losses in the piping and collectors the real operating point is far left towards higher head and lower flow conditions (Q=1.1 m³/h). Thus the real total efficiency η_{total} reaches only 22 % whereby about 110 Watts of electricity are consumed to supply solar heat to the buffer tank. Analogous, another similar circulator provides a hot water flow-rate of 1.6 m³/h to the generator of the adsorption chiller. Higher flow-rate at comparable head induces higher hydraulic power, but pumping efficiency increases in this case to 30 % so that about 120 Watts of electricity are consumed. The cooling tower is linked to the adsorption chiller by short pipework filled with propylene glycol water mixture. Under best conditions the double-stage cooling water pump (Grundfos TP32-150/2) offers 37 % overall efficiency. As real operating point is near the Best Efficiency Point about 370 Watts suffice to drive the heat carrier medium. Finally, another pump (Grundfos TP25-50/2) is needed to overcome the head loss between evaporator and cold water buffer in the chilled water circuit. Therefore about 110 Watts of electricity are consumed.

In contrast, the cumulated required hydraulic power amounts to only 201 Watts.

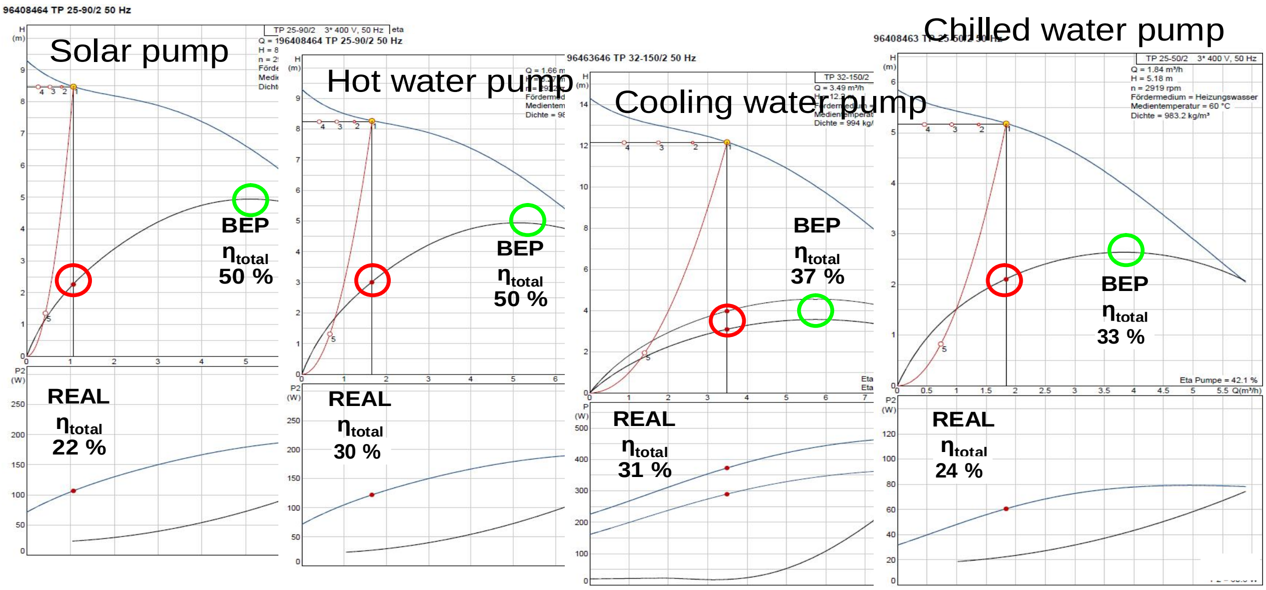


Figure 13: Characteristic HQ and power diagrams with real operating points of circulators implemented in the hydraulic subsystems of a 7.5 chilled water capacity SHC system

Compendious, if the whole electricity consumption for pumping effort is summed up (710 Watts) and referred to the chilled water capacity (7.5 kW) the maximum EER of this scenario is calculated to 10.5 (ignoring other electricity consumers such as control, chiller internally, cooler fan a.so.) Without speed adjustment to part load conditions (e.g. 3.75 kW chilled water capacity) the overall electricity consumption of pumps is unaffected, leading to proportional reduced EER.

The second example comes with canned high-efficiency circulators, mostly adaptable to part load conditions by proportional adjustment of their rotary speed.

In order to drive the 10 kW Absorption chiller a 57 m² flat plate solar collector array provides solar heat to a buffer (2000 Liters) via primary (water propylene glycol mixture) and secondary (water) hydraulic circuit. While the speed of the primary pump (Grundfos Magna 25-100) is adjusted in accordance with the insolation in collector plane, the secondary solar pump ensures a constant supply temperature to the buffer. Figure 14 shows the characteristic HQ and P_1 diagram (light blue section) of the primary solar collector pump and its operating points at design point (100% Load) and minimum part load.

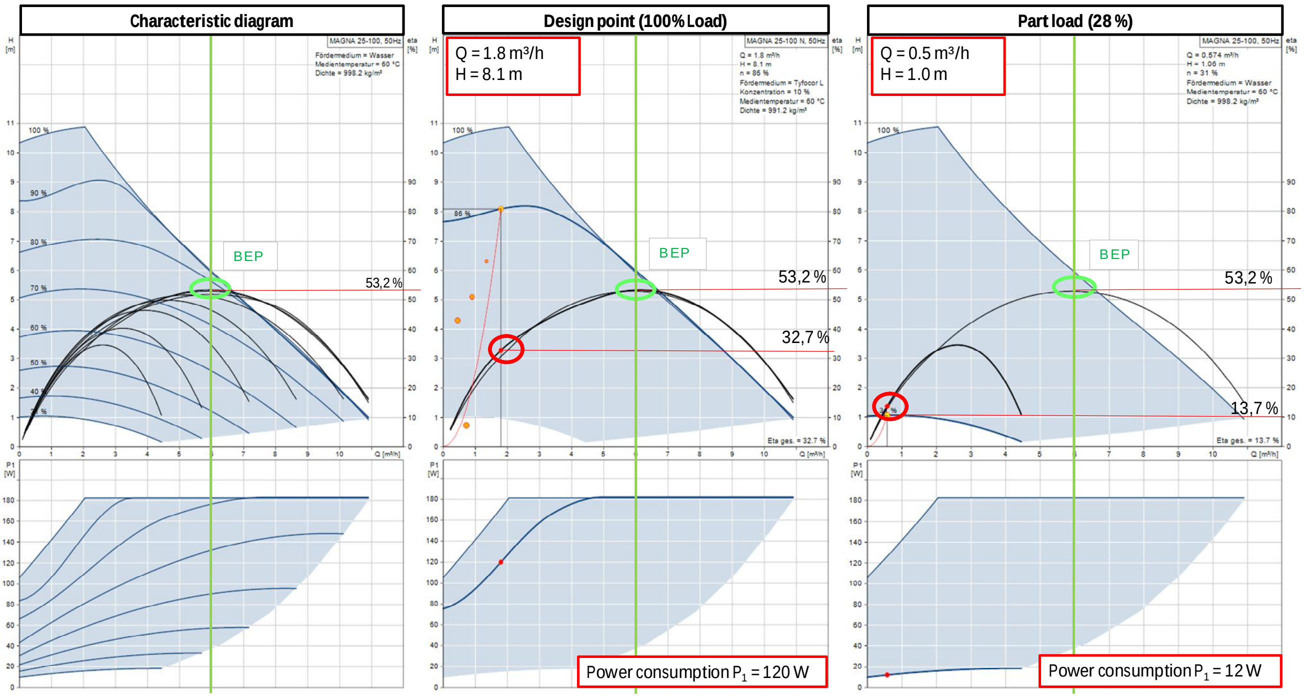


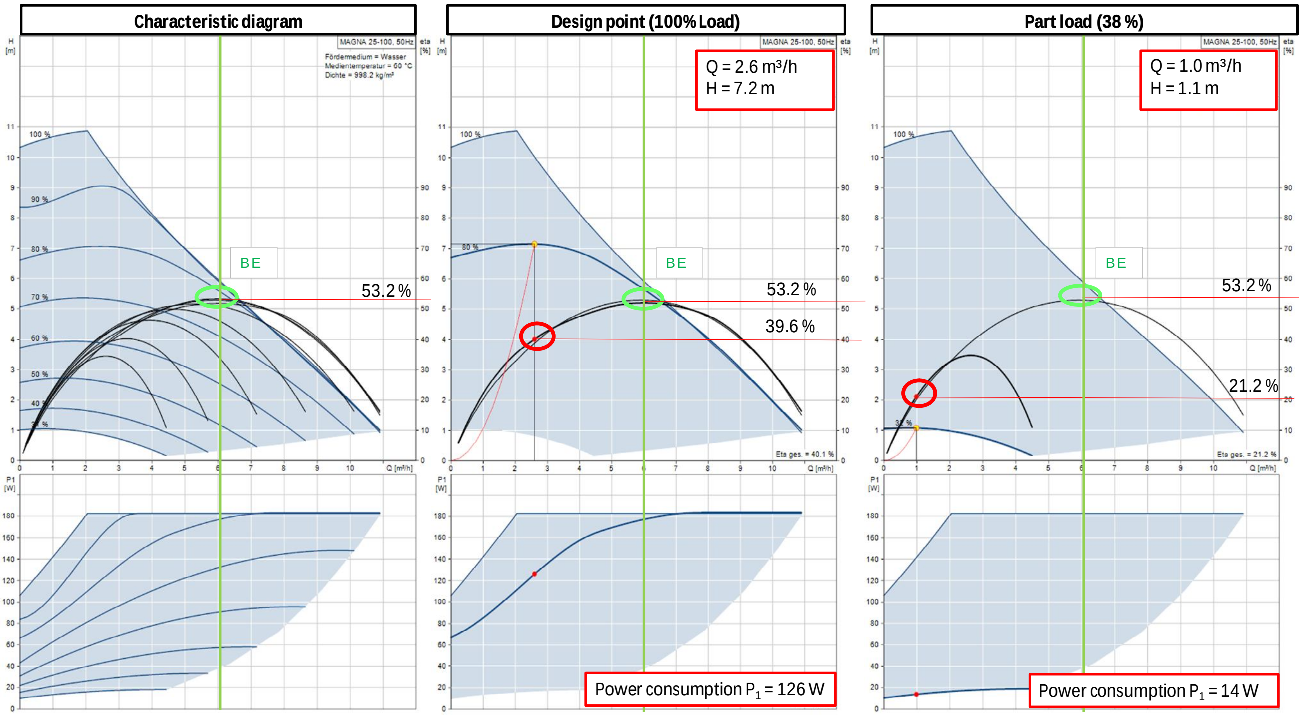
Figure 14: Characteristic HQ and power diagrams of solar primary pump and its operating points

A maximum efficiency of 53.2 % is achievable at Best Efficiency Point (BEP) of this pump. Similar to the prior example the real operating point (Q=1.8 m³/h) is far left resulting in reduced conversion efficiency of only 32.7 %. During minimal part load conditions (Q = 0.5 m³/h) it even drops down to 13.7 %. However, the high dynamic of 3.6 in flow-rate (1.8 compared to 0.5 m³/h) allows a proportional speed adaption between 100 and 28 % load. Despite decreasing pump efficiency the specific electricity consumption is disproportionately reduced from 120 Watts to 12 Watts respectively.

Analogous, the solar secondary circuit pump (WILO-Stratos 25/1-5 BMS) provides a maximum efficiency of 30.9 in contrast to the real value of 25.9 at design point (Q=1.4 m³/h). Likewise, efficiency drops down to 12.5 % in part load conditions at Q=0.9 m³/h. Since this pump is oversized the dynamic range is limited to 1.6, which leads to a constant electricity consumption of 12 Watts even below 62.5 % part load conditions and 25 Watt at design point.

Due to a very low pressure loss through the generator and the pipework between the chiller and heat buffer the pumping effort is relatively small. For this case, a constant speed pump (Grundfos Alpha2 25-50) fits best, operating at almost optimum efficiency of 42.4 % with a constant electricity consumption of 33.5 watt in total.

The cooling water circuit is divided in a primary water and a secondary water-glycol mixture circuit for freeze protection. Both pumps, primary (Grundfos Magna 25-100) and secondary (Grundfos Magna 32-100), actuate depending on the chilled water capacity between 100 % and 38 % of nominal flow.



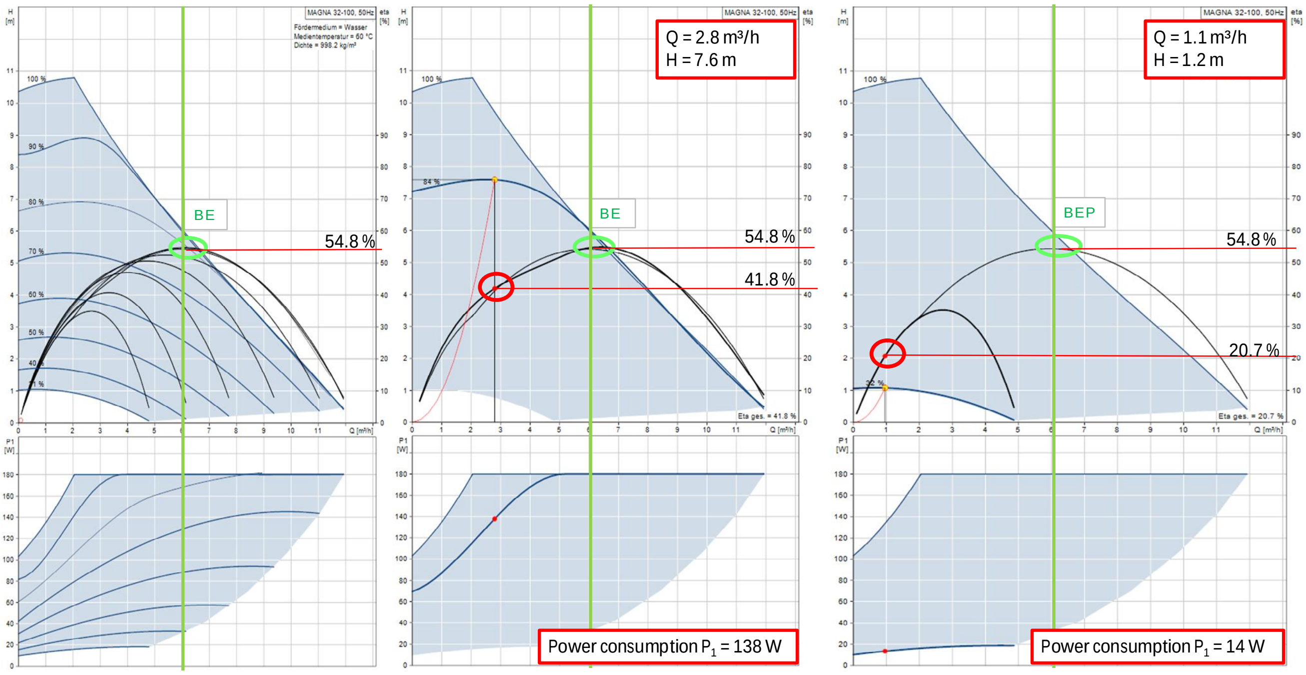


Figure 15: Characteristic HQ diagrams of primary and secondary pump in the cooling water circuit

Consolidated, at design point conditions 264 Watts are to spend for transferring rejected heat from the absorber to the cooling tower and at 38% part load or lower constant 28 Watts, respectively.

The chilled water pump (Grundfos Magna 25-100) operates, differential pressure controlled, between Evaporator and cold distribution of the building. In this way, the pump modulates depending on the chilled water consumption or cooling demand respectively of the building. Furthermore, this reference input allows for speed adaption of cooling water circuit pumps and cooling tower fan to part load conditions. With a high dynamic ratio of 2.7 the chilled water pump operates with satisfying efficiency in a wide range.

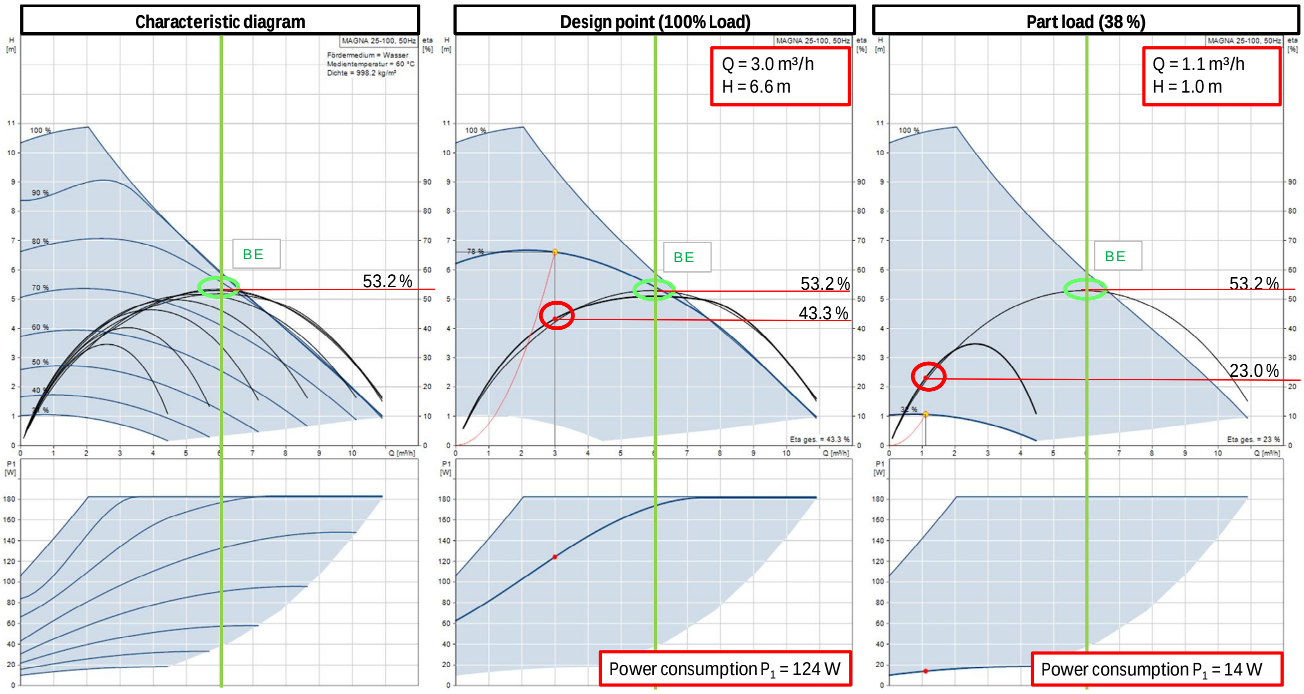


Figure 16: Characteristic HQ and power diagrams of chilled water pump at different operating points

Compendious, the notional overall EER for 10 kW chilled water capacity including the electricity consumption of all pumps at design point (566.5 Watts) figures to about 17.6. Assuming a minimum part load capability of the absorption chiller around 2.5 kW chilled water capacity (25% part load), the overall electricity consumption caused by circulators remains at 99.5 watts. Consequently the EER referred to pump work increases to 25 despite worse conversion factors.

Conclusion:

Although, even small high-efficiency pumps feature high conversion efficiency between electricity and hydraulic power the practical operating point is often far away from that optimum. A proper match of characteristic system and pump curve during design phase is essential. Furthermore if a high adaption to part load conditions is stipulated, a variable speed control at least of cooling water and chilled water pump is recommended.

6. Design Guidelines and Best Practice for Efficient Hydraulics

Notwithstanding various design guidelines, handbooks and simplified system schemes dealing with optimized hydraulic design in SHC systems a proper and well performing installation is still a question of experience or pre-fabrication. The complexity and interaction of many different units cause a high on-site planning effort including multiple error sources. Determinative aspect is a correct pressure loss calculation and review of the overall auxiliary electricity consumption as well as control strategy throughout the whole operating range. Subsequent, some main design aspects as well as recommended standard values for Auxiliary Energy Consumption Ratios are given.

6.1. Guidelines for hydraulic design and pump selection

- Reduce heat carrier flow through the main components, thus
- Prefer high temperature differences in the main circuits especially in the cooling water circuit where most of the electricity is consumed.
- Prefer chiller design with low pressure losses through internal heat exchangers,
- Reduce pipe length of water/glycol circuits to a minimum
- High thermal COP reduces driving and rejected heat quantity simultaneously and consequently pumping effort of heat carrier medium.
- Reduce pressure losses in the pipework (sharp edges, Valves, filters, etc.)
- Design pipe diameter to achieve heat carrier medium flow speed between 1 and max. 1.5 m/s for medium sized SHC-Systems
- Select the operating point of pumps lightly right from Best Efficiency Point in the Sweet or Happy zone to achieve best pump efficiency at part load conditions
- Avoid high head and multi-stage pumps as each additional stage reduces the overall efficiency
- High-efficiency pumps are highly recommended, at least for chilled water and cooling water circuit
- Pump speed related to variable heat flow ensures high performance in a wide operating range
- Continuous review of AECRs during system operation exposes malfunctions promptly

Recommended standard values of AECR in order to achieve a good seasonal system performance.

	AECR at design point and even better under part load conditions
$AECR_{Solar}$	~ 150
$AECR_{Backup}$	~ 250
$AECR_{D}$	~ 250
$AECR_{AC}$	~ 100
$AECR_{Cooler}$	~ 100
$AECR_{E}$	~ 80
$AECR_{chiller}$	~ 150

6.2. Best practice and successful optimization of solar cooling system

Among others, the SHC System at ZAE Bayern could benefit from a detailed scientific supervision and optimization over more than 3 years. This small scale solar cooling and heating installation erected in 2007 comprises a 10 kW cooling capacity absorption chiller with dry heat rejection supported by a latent heat storage. Solar heat is harvested by a common flat plate collector array of 57 m² aperture surface and delivered to a 2000 Liter hot water buffer. Figure 17 lists the whole number of electricity consumers. Starting in 2010 the overall electricity consumption at design point had been calculated to 1345 Watts. Over the following years several optimization phases took place.

As a first step, low efficient circulators of the solar primary and secondary as well as hot water driving circuit had been replaced by high-efficiency pumps. Furthermore oversizing had been eliminated. Thus, an abated overall electricity consumption of about 1117 Watts had been obtained. Subsequent optimization concentrated on fan efficiency and standby auxiliary energy consumption, resulting in a minor effect on system efficiency. In 2012 a revision of the absorption chiller parameters raises thermal COP and temperature difference in the driving hot water circuit, by which means the required flow-rate and heat dissipation to the ambient decreases. Additionally, a revised chilled water pump speed control is applied. Currently the electricity consumption at design point conditions totals to 732 Watts. Because of continual system optimization the Seasonal performance factor SPF_el,tot of this SHC System improved from 7.9 in 2010 to 12.07 in 2012.

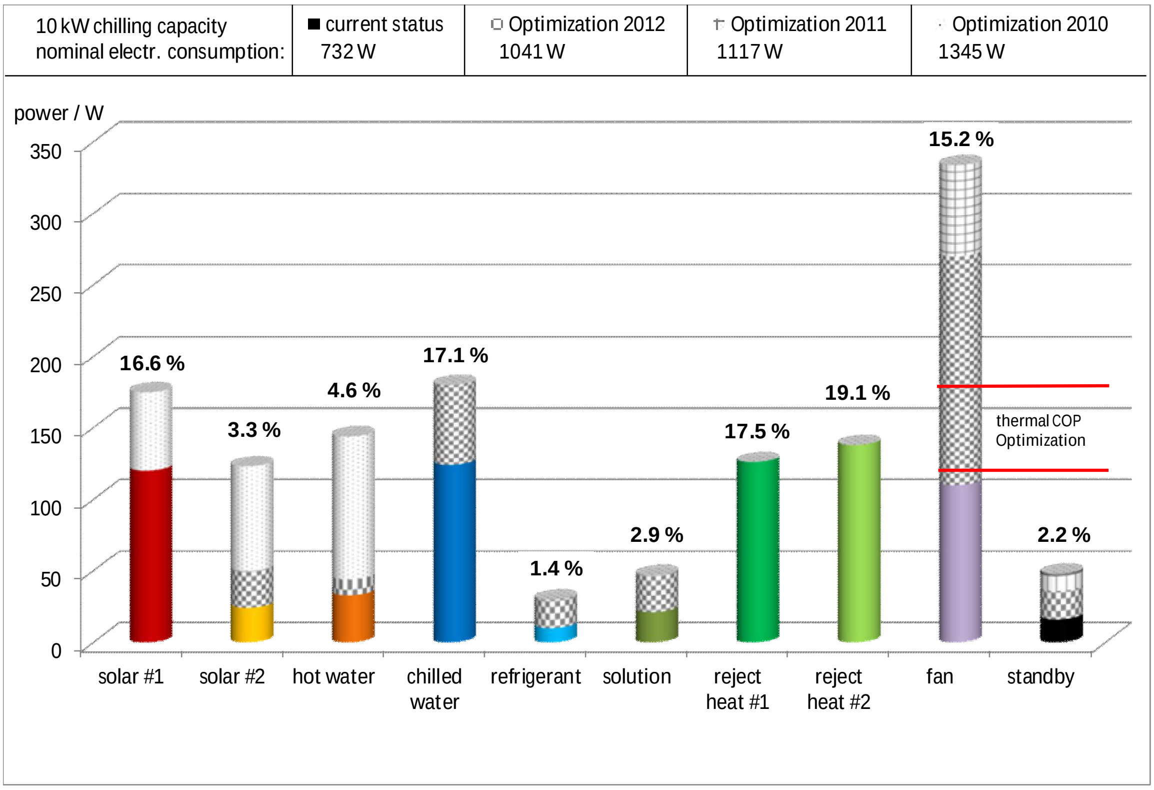


Figure 17: Distribution of auxiliary energy consumption and stepwise optimization results

The following Figure 19 and Figure 20 show the overall system performance key figures thermal and EER for the selected SHC system during the cooling period 2013. In addition to that, relevant boundary conditions such as cumulated daily insolation on collector plane and maximum, minimum and average ambient air temperature are given in Figure 18.

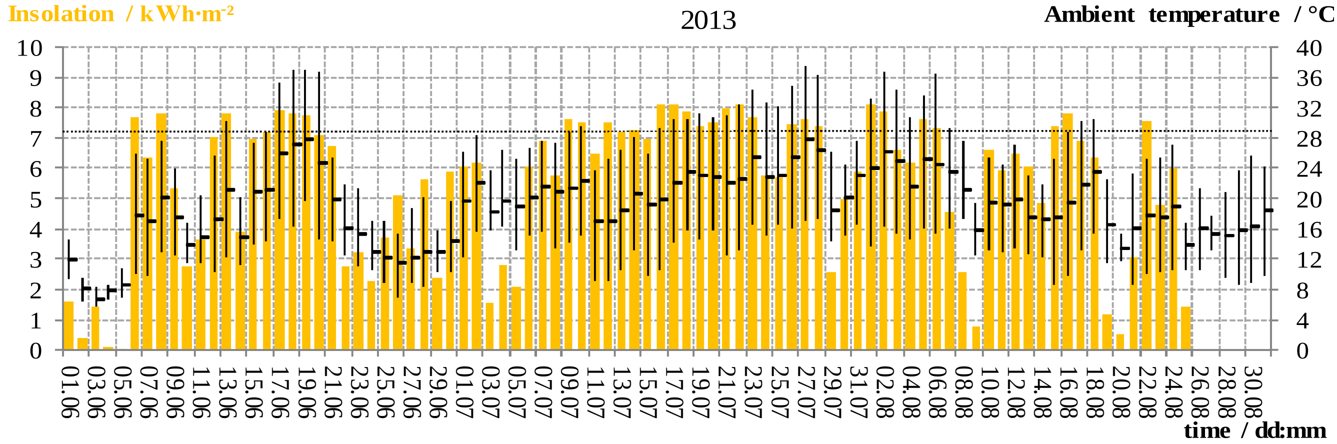


Figure 18: Daily insolation on collector surface and ambient temperatures (max, min, average)

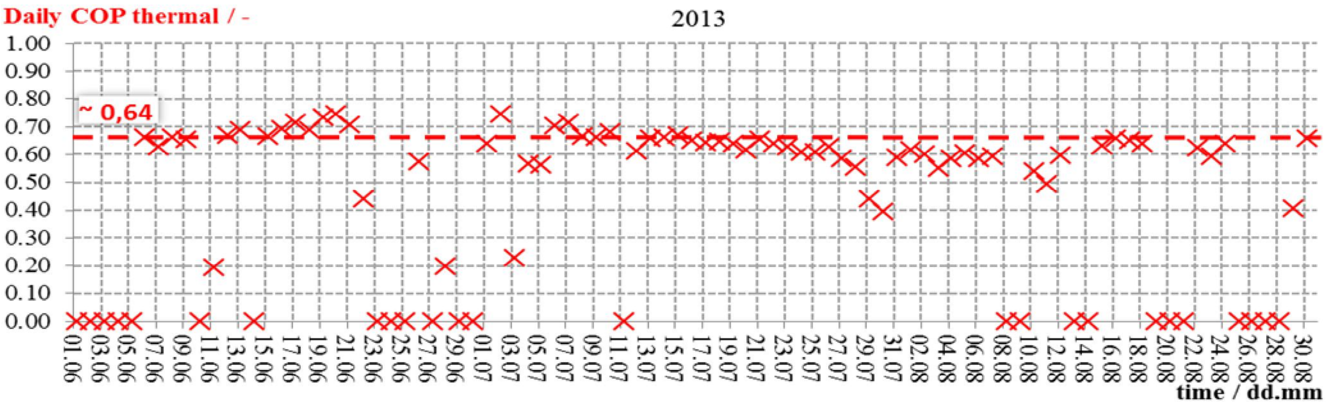


Figure 19: Daily thermal COP including start up and shutdown phases

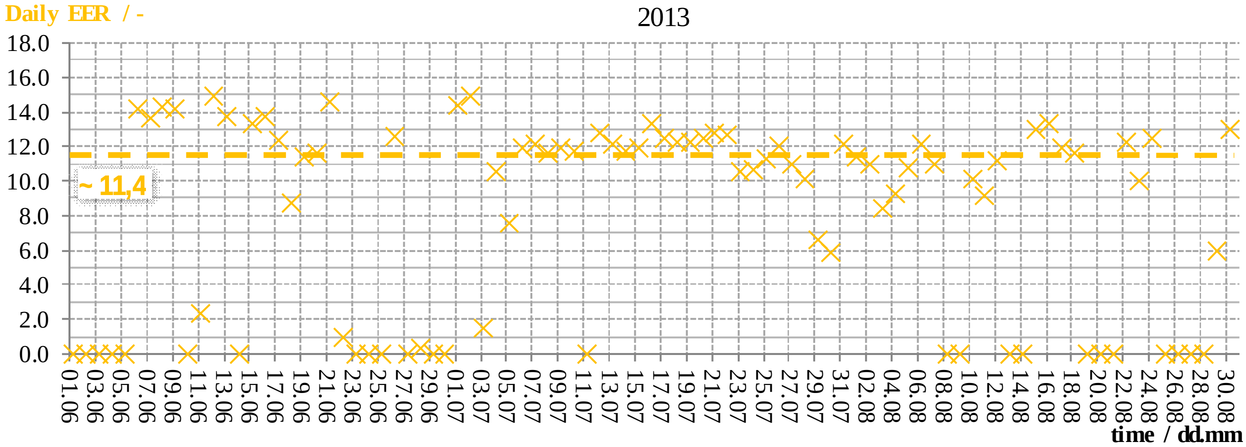


Figure 20: Daily EER for chilled water production including all electricity consumer of the SHC System

Expect for a few days, with little operating hours of the chiller, the thermal COP reaches a seasonal average of 0.64 and the seasonal EER for cooling averages to about 11.4.

Regarding pump and hydraulic circuit efficiency Figure 21 plots the daily AECR Solar as yellow dots in front of the daily cumulated insolation on collector plane and delivered heat to the heat buffer. In most cases with considerable solar gain the AECR Solar exceeds 150.

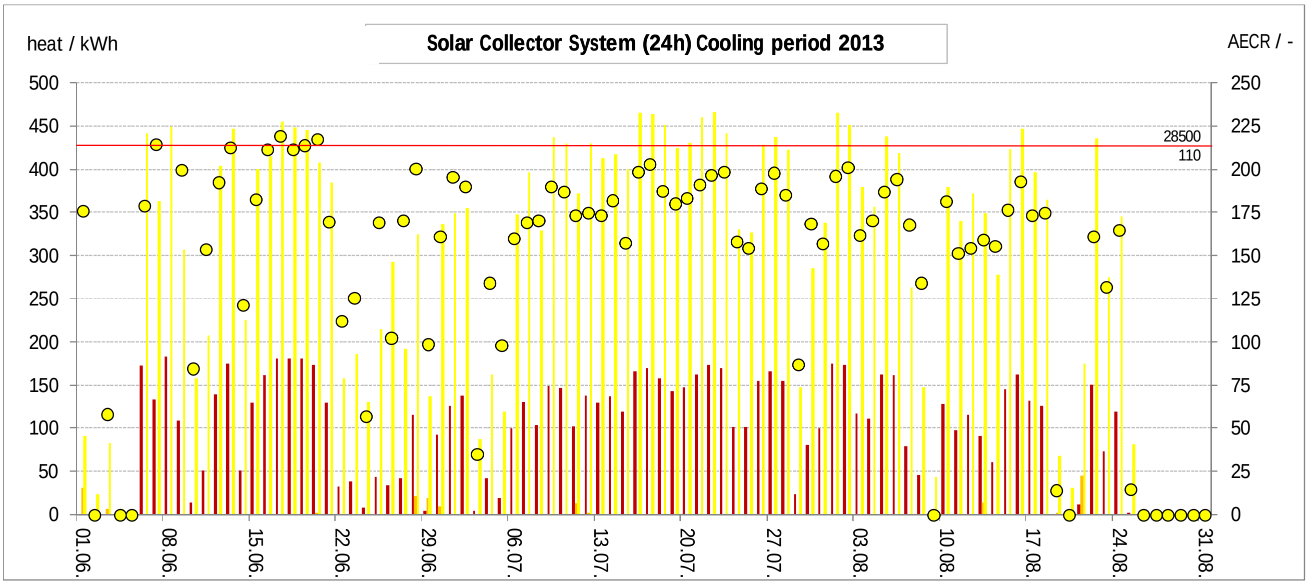


Figure 21: Daily auxiliary energy consumption ratios of the solar collector circuit

Further major auxiliary electricity consumers are primary and secondary pump of the cooling water circuit due to the high amount of rejected heat in single stage absorption chillers. Thus, a high temperature difference between cooling water supply and return reduces flow-rate and pumping effort accordingly. In addition to that, a proper design of the pipework with flow speeds below 1 m/s and minimized length of the secondary circuit containing water propylene glycol mixture, in combination with variable pump speed at part load conditions, ensures an almost constant high $AECR_{AC}$ above 100 as shown in Figure 22.

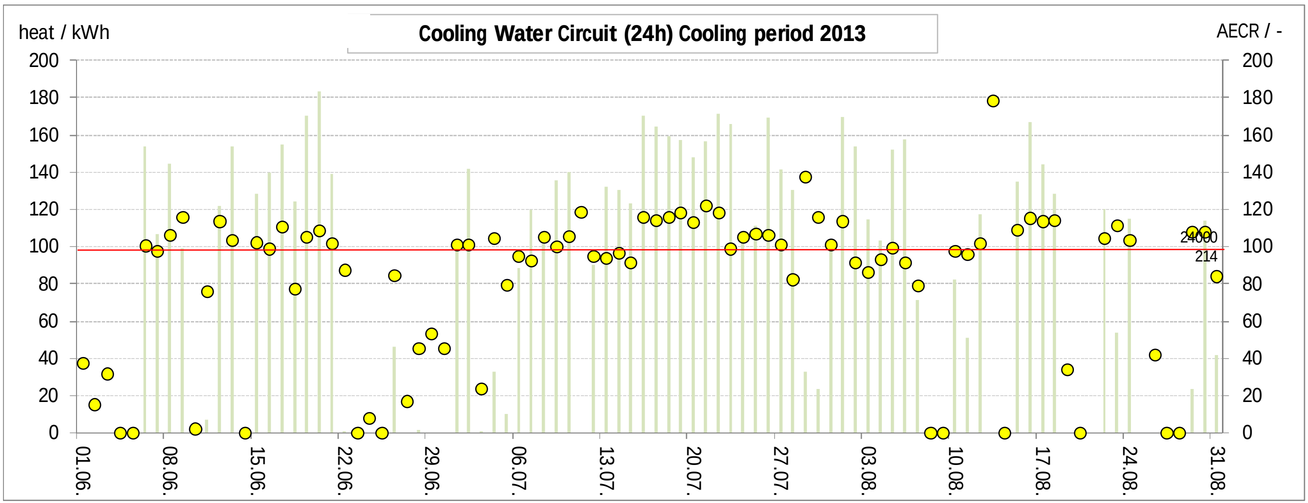


Figure 22: Daily auxiliary energy consumption ratios $AECR_{AC}$ of the cooling water circuit

7. Discussion and Conclusion

Beside the electricity consumption of the cooling tower, auxiliary power in thermally driven SHC Systems is mainly dominated by pumping effort for heat transmission between the components. Since 2010 great improvements in overall pump efficiency have been obtained, in particular due to legislative restrictions and manufacturer initiatives. Depending on size, about 50% to 80% of the consumed electricity is converted into useful hydraulic power. But the utilization of high-efficiency pumps in solar cooling installations does not implicate an efficient pumping automatically. The strong relationship between characteristic pump and system curve demands a proper system design as well as pump selection.

Due to either huge heat quantity and/or small temperature difference the high specific auxiliary energy consumption in the chilled and cooling water circuit are dominating. Even though investment costs of a variable speed high-efficiency pump often is twice as high compared to a cheap, but inefficient fixed speed pump, the overall costs for pumps in SHC systems are rather small or often negligible. But the ability to adjust pump speed to part load conditions, especially in these hydraulic circuits, has a high impact on system efficiency regarding EER.

In addition to that, intelligent high-efficiency pumps featured with internal flow, pressure, temperature and power meter offer a great opportunity for advanced system evaluation at concurrently reduced measuring equipment. Theoretically, the potential savings of auxiliary energy for pumping effort under part load conditions are huge. Due to its complexity and various interactions between several components, implementation will be reserved for manufacturers or prefabricated system providers. Nevertheless, there are some SHC Systems which operate quite fine after some years of optimization.

The way things are an overall SEER for well-designed small scale solar cooling systems of 20 seems to be feasible. However, currently many sorption chiller manufacturers mainly focus on cheap chillers than good seasonal system performance. As the thermodynamic and hydraulic design criteria of the chiller defines the capacities, temperature levels and/or heat transfer medium flows in the entire solar cooling system their thermal design should be revised.

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9. Acknowledgements

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