ANALYSIS OF A GAS-DRIVEN ABSORPTION HEAT PUMPING SYSTEM USED FOR HEATING AND DOMESTIC HOT WATER PREPARATION

Harald Moser, Rene Rieberer, Graz University of Technology, Institute of Thermal Engineering, Inffeldgasse 25 B, A-8010 Graz, Austria, rene.rieberer@tugraz.at

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Abstract: In the building sector the use of thermally driven heat pumps can contribute substantially to energy conservation and the reduction of end-use energy. Within the IEA HPP Annex 34 a medium size absorption heat pump application installed in a storehouse in Graz has been monitored over a period of one year.

The system consists of two directly natural gas-driven ground source ammonia/water absorption heat pumps (each with ca. 40 kW heating capacity). Via a buffer storage space heating of an office building and a storage depot as well as domestic hot water preparation is provided.

During the monitoring period the system showed reliable operation and high energy performance. Based on the lower heating value of the natural gas the seasonal performance of the year 2010 was 1.54 which is approx. 60% higher compared to a condensing gas boiler with a seasonal performance of 96%. However, also room for improvements has been detected, especially at start/stop operation for domestic hot water preparation in summer.

Key Words: demonstration project, monitoring, IEA HPP Annex 34, ground source

1 INTRODUCTION

In the building sector the use of natural gas driven absorption heat pumps can contribute substantially to energy conservation and the reduction of CO_2-emmisions due to the reduction of the end-use energy demand. Especially the small-capacity range of absorption heat pumps for heating purpose are commonly considered as the logical next development step beyond the condensing gas boiler and thus the market potential is enormous. However, small-capacity applications still suffer from relatively low efficiencies and/or high investment cost compared to other technologies.

In the medium capacity range of around 50 kW heating capacity several absorption heat pumps are available on the market. Large scale AHP for heating purpose are relatively rare and mainly special designed systems based on chillers are used (Ziegler 2002).

The Annex 34 “Thermally Driven Heat Pumps for Heating and Cooling” within the IEA “Heat Pump Program” (HPP) aims to the reduction of the environmental impact of heating and cooling systems by the use of thermally driven heat pumps. One of the main objectives of the Annex 34 is to quantify the economic, environmental, and energy performance of integrated thermally driven heat pumps in cooling and heating systems in a range of climates, countries, and applications. Within this framework at Graz University of Technology (Institute of Thermal Engineering) a demonstration project has been started. The aim of the project is to evaluate the seasonal performance of two ammonia/water absorption heat pumps for heating and domestic hot water preparation.

2 APPLICATION DATA

Two “Helioplus 40-S” absorption heat pumps from the company “Helioplus Energy Systems GmbH” are installed in a storehouse of a brewery in Graz, Austria (compare Figure 1 and Figure 2). These ammonia/water AHP are directly driven by natural gas, have a maximum supply temperature of 60°C and a heating capacity of 37.1 kW each. One feature of these AHP is that they include also a flue gas heat exchanger in order to condense water in the flue gas by rejecting condensing heat directly to the return flow of the heating circuit.

Figure 1: Storehouse building where the AHPs are installed

Figure 2: Photo of AHP installation

The AHP supply heat to a 1.2 m³ stratified thermal water storage tank (compare Figure 3). This storage tank supplies heating water to both the space heating distribution system and the domestic hot water tank. The outlet nozzle for the space heating system is located at an intermediate level and for domestic hot water preparation at the very top in order to receive the required temperature level.

The heating area comprises of an approx. 2000 m² storage depot and an office area with approx. 600 m². The storage depot has a nominal heating capacity of ca. 57 kW and is heated to 18°C. The office has a heating capacity of ca. 19 kW and is heated to 22°C room temperature. For domestic hot water (DHW) preparation a 0.5 m³ storage tank is used. It is charged by the heat from the heating water storage tank via a plate heat exchanger.

For space heating the heating water storage is controlled with respect to the ambient temperature and heating curve in order to achieve a temperature level in the storage approx. 2 K in excess of the required supply temperature (ca. 40°C at -12°C ambient temperature) for the heat distribution system.

If the temperature level of the DHW tank drops below a certain threshold the control of the AHP is switched to the DHW temperature set point in order to heat the upper part of the heating water storage to the required temperature level of more than 50°C. Using this heating water the domestic hot water tank is charged via a plate heat exchanger. After the temperature level of the domestic hot water tank has reached the temperature set point of 50°C the control of the AHP is switched back to the current space heating temperature set point.

The low temperature energy source for the AHP is provided by 7 ground probes each with a length of 100 m and a mixture of water and propylene glycol (ca. 20 wt%) is used as heat carrier. During summer the ground probes are used for free cooling of the office area. Therefore a plate heat exchanger is used in order to reject the heat from the heating water to the heat carrier of the ground probes. However, free cooling operation is not discussed in this paper and active cooling by the AHP is not provided.

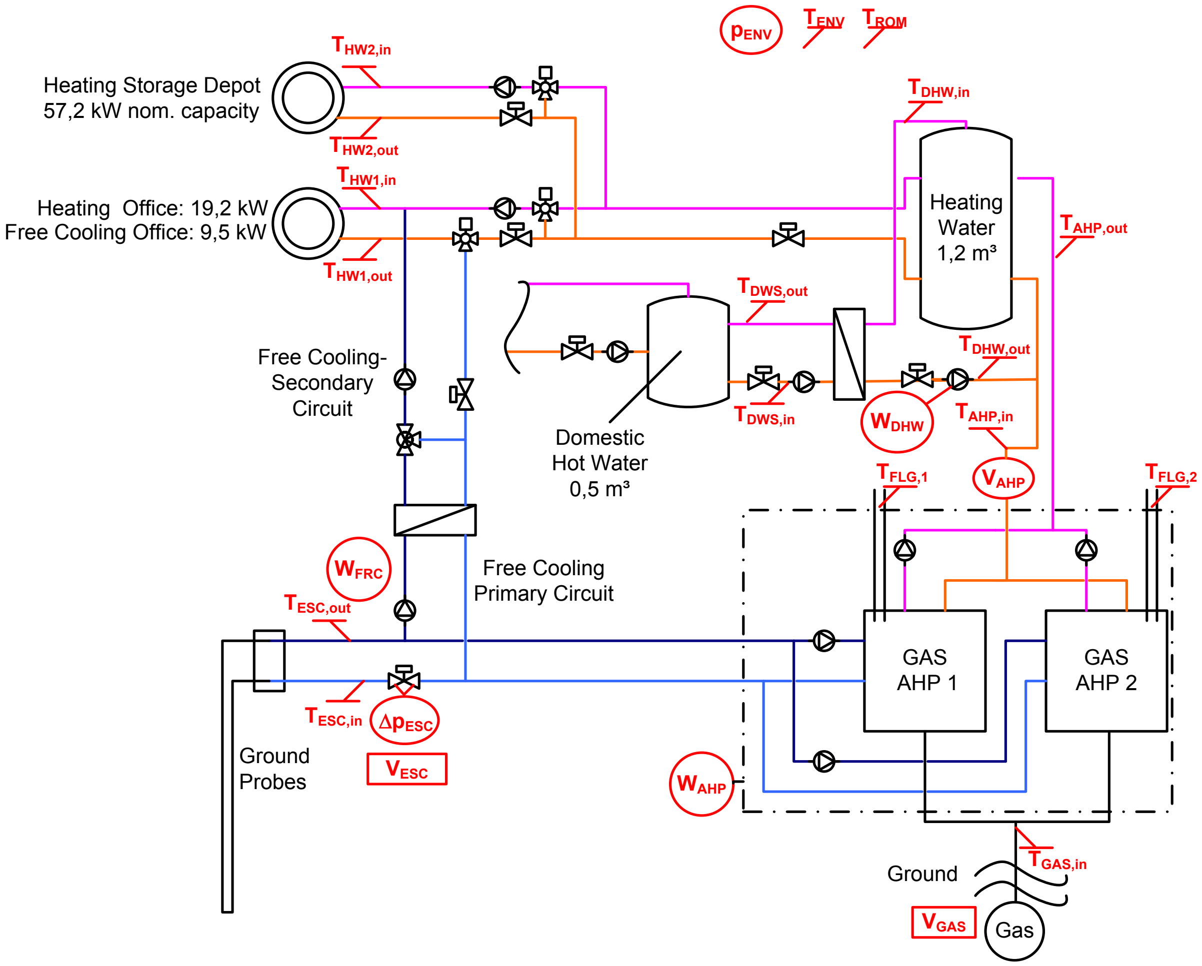


Figure 3: Schematic drawing of the heating system including measurement instrumentation (for legend symbols see Nomenclature)

3 MONITORING CONCEPT

The system has been monitored from 28.12.2009 until 03.01.2011 using a data logger (type "e.reader") and all relevant data have been measured, processed and recorded within a measurement interval of 10 seconds. Due to some problems with the data transfer within the time period of week 17 to 19 and 23 to 25 the data sets are incomplete for these weeks. However, all available data have been used for system evaluation.

3.1 Measurement Instrumentation

The measurement instrumentation contains all relevant temperatures, the energy input of the heat source, natural gas and the electricity consumption as well as the delivered heat to the heating water storage tank. The measurement instrumentation is also indicated in the schematic drawing in Figure 3.

For the natural gas measurement a bellows-type gas flow meter is installed outside the building. In order to calculate the energy input of natural gas Eq. 1 and Eq. 2 have been used. For this the average upper heating value (Hv_{GAS}) was assumed to be 11.176 kWh/Nm³ (AGGM, 2011) as derived from the gas supplier (average Hv_{GAS} value for the period Jan. to Apr. and Sep. to Dec 2010). The ambient pressure has been measured and the gas pressure (p_{GAS}) has been assumed to be 22 mbar above the ambient pressure as controlled by the pressure reducing regulator upstream the gas flow meter. The standard pressure (p_{NORM}) is 1.013 $bar_{a,}$ the standard temperature (T_{NORM}) is 0°C. Because exact measurement of the gas temperature was not possible a temperature of 6°C has been assumed as it is also used for the gas billing procedure.

$$Q_{GAS} = Hv_{GAS} \cdot V_{GAS,NORM} \tag{1}$$

$$V_{GAS,NORM} = V_{GAS} \cdot \frac{T_{NORM}}{T_{GAS}} \cdot \frac{p_{GAS}}{p_{NORM}} \tag{2}$$

The heat coming from the ground probes have been calculated using Eq. 3 and Eq. 4. Due to the fact that the system was already installed and in operation when monitoring was decided within the IEA Annex 34 it was not permitted to insert flow meters in the hydraulic lines. That is why the volume flow has been measured indirectly by measuring the pressure loss via a balancing valve (compare Eq. 4).

$$Q_{ESC} = \rho_{ESC} \cdot \dot{V}_{ESC} \cdot cp_{ESC} \cdot (T_{ESC,out} - T_{ESC,in}) \cdot \tau \tag{3}$$

$$\dot{V}_{ESC} = \frac{Kv_{ESC}}{3600} \cdot \sqrt{\Delta p_{ESC}} \cdot \sqrt{\frac{\rho_{ESC}}{1000}} \tag{4}$$

The flow coefficient (Kv_{ESC} = 19.31 m³/h) of the balancing value has been determined on site using an ultrasonic flow meter. For calculation of the density and specific isobaric heat capacity a temperature depended polynomial curve fitting of the thermodynamic data has been used.

Electricity meters have been used for the electrical energy input to the AHP (which comprises of the heating water pumps, energy source pumps and AHP control), the pump for domestic hot water preparation and the pump for the primary free cooling circuit. The electrically consumption of the pumps for the heat distribution system have not been measured.

In the return flow line of the AHP heating water circuit a heat meter is installed which gives the possibility to use it as volume flow counter. Using this flow counter the heat output of the AHP has been calculated acc. to Eq. 5 for each measurement interval.

$$Q_{AHP} = \rho_{AHP} \cdot V_{AHP} \cdot cp_{AHP} \cdot (T_{AHP,out} - T_{AHP,in}) \tag{5}$$

All temperatures have been measured using PT100 sensors or thermocouples. In order to analyse the efficiency of the absorption heat pump the weekly Coefficient of Performance for heating (COP_H) has been calculated as indicated in Eq. 6. In order to analyse the absorbed energy from the energy source additional the COP for cooling (COP_C) has been calculated using Eq. 7.

$$COP_H = \frac{Q_{AHP}}{Q_{GAS} + W_{AHP}} \tag{6}$$

$$COP_C = \frac{Q_{ESC}}{Q_{GAS} + W_{AHP}} \tag{7}$$

3.2 Estimation of Measurement Uncertainties

In order to evaluate the combined standard uncertainties of the calculated heat flows and COP values the individual uncertainties of the measurement instrumentation have been estimated and combined by Gaussian error propagation as shown in Eq. 8 (ISO 1995).

$$u_y = \sqrt{\sum_i^N \left(\frac{\partial f}{\partial x_i}\right)^2 \cdot u^2{}_{x_i}} \tag{8}$$

For the natural gas energy consumption the measurement uncertainty of approx. 0.3 kWh/Nm³ has been estimated which correspond to the individual uncertainties given in Table 1.

Table 1: Estimated uncertainty for calculation of u(QGAS)

$u(Hv_{GAS})$	$u(V_{GAS})$	$u(T_{GAS})$	$u(p_{ENV})$
[MJ/Nm³]	[%]	[K]	[bar]
0.54	1	4	0.02

The measurement uncertainty for the low temperature heat source (Q_{ESC}), the delivered heat of the AHP (Q_{AHP}) and the electrical energy input (W_{AHP}) has been calculated for each measurement interval using the individual measurement uncertainties as given in Table 2.

Table 2: Estimated measurement uncertainty for calculation of $u(Q_{ESC})$, $u(Q_{AHP})$ and $u(W_{AHP})$

$u(cp_{ESC})$	$u(m_{ESC})$	$u(T_{ESC,out})$	$u(T_{ESC,in})$	$u(cp_{AHP})$	$u(m_{AHP})$	$u(\Delta T_{AHP})$	$u(W_{AHP})$
[kJ/kg K]	[kg/h]	[K]	[K]	[kJ/kg K]	[%]	[K]	[%]
0.05	350	0.1	0.1	0.02	1.5	0.4	1

It should be noted, that the temperature sensors of the low temperature heat source have been wet installed through drain nozzles, thus the process temperatures could be measured quite accurately. Unfortunately this was not possible at the heating water pipes of the AHP, thus these temperature sensors have been installed on the outside surface of the pipes. Subsequently these piping sections have been well insulated in order to limit the effect of the ambient air temperature. However, the measured temperature possibly deviates from the process temperature, especially at transient operating conditions. Because the temperature difference ($\Delta T_{AHP}=T_{AHP,out}-T_{AHP,in}$) which is relevant for calculation of the capacity can be measured with higher accuracy this difference has been used for uncertainty propagation.

For all data sets of every week the uncertainties of the COP have been calculated acc. to Eq. 9 and Eq.10.

$$u_{COP_{H,AHP}} = \sqrt{\left(\frac{1}{Q_{GAS}+W_{AHP}}\right)^2 \cdot u^2{}_{Q_{AHP}} + \left(\frac{-Q_{AHP}}{(Q_{GAS}+W_{AHP})^2}\right)^2 \cdot u^2{}_{Q_{GAS}} + \left(\frac{-Q_{AHP}}{(Q_{GAS}+W_{AHP})^2}\right)^2 \cdot u^2{}_{W_{AHP}}} \tag{9}$$

$$u_{COP_{C,AHP}} = \sqrt{\left(\frac{1}{Q_{GAS}+W_{AHP}}\right)^2 \cdot u^2{}_{Q_{ESC}} + \left(\frac{-Q_{ESC}}{(Q_{GAS}+W_{AHP})^2}\right)^2 \cdot u^2{}_{Q_{GAS}} + \left(\frac{-Q_{ESC}}{(Q_{GAS}+W_{AHP})^2}\right)^2 \cdot u^2{}_{W_{AHP}}} \tag{10}$$

4 MONITORING RESULTS

The system behavior for heating and domestic hot water preparation is discussed hereafter within the time frame of 07.03.2010 from 4:30 to 10:30 as shown in Figure 4a - 4d.
Figure 4a show the temperatures of the AHP, i.e. heating water and heat source in- and outlet. The temperature levels during domestic hot water preparation and the operation status information of the DHW pump are indicated in section b. The relevant temperature level of the two heat distribution systems (office building and storage depot) are shown in section c, and section d displays the operation status of the two AHP and the mass flow for heating water and heat source of the AHP.

At 4:30 one AHP is in heating operation. The AHP absorbs the heat from the cold water (6 to 2°C) which is the heat carrier from the ground probes to the AHP. The heating water enters the AHP with a temperature of 23°C and is heated up to approx. 35°C (compare Figure 4a). The supply flow temperature of both heat distribution systems is 32°C and the return flow is ca. 27°C for the office building and ca. 20°C for the depot area (compare Figure 4c).

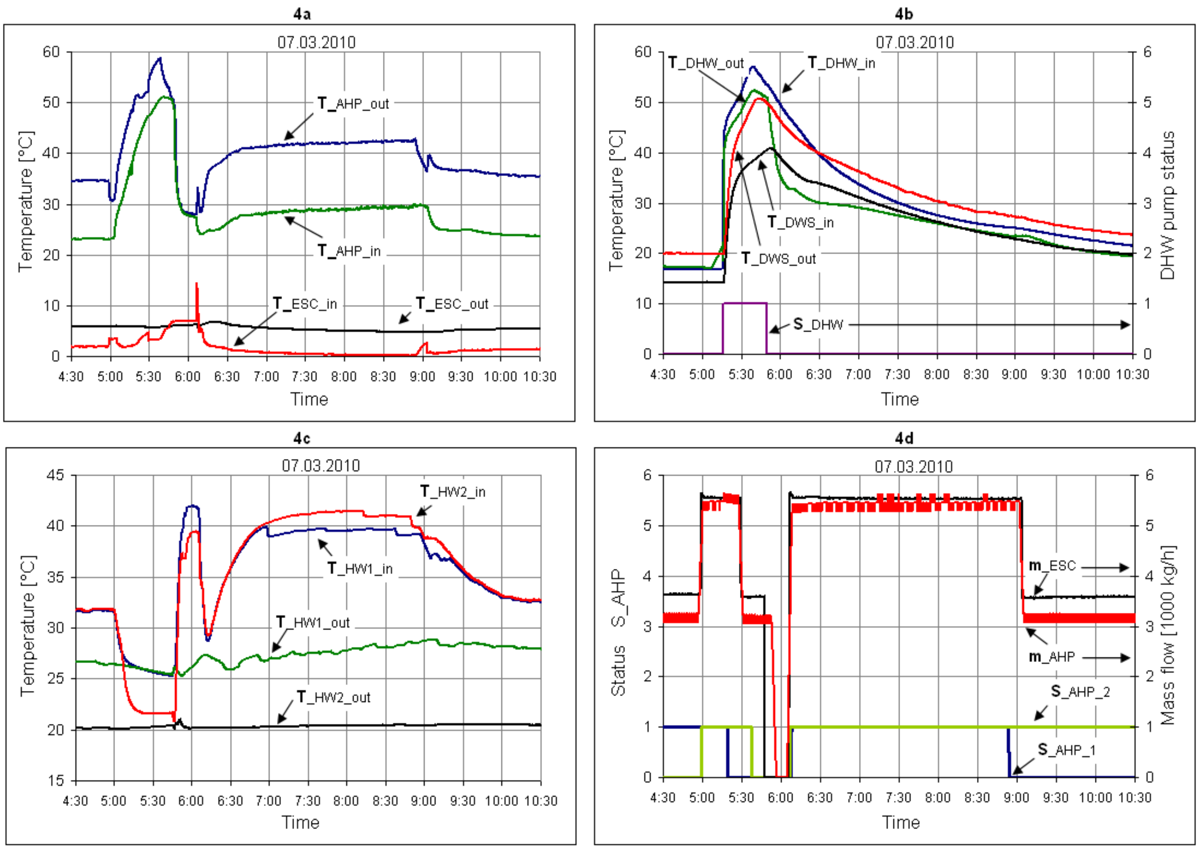


Figure 4: Temperature level, mass flow and operation status information for system operation during space heating and DHW preparation.

At approx. 5 o'clock the control is switched to DHW preparation mode, the second AHP is put into operation (Figure 4d) and the heat distribution for space heating is switched off (Figure 4c). Since the temperature level of the heating water tank rises, at approx. 05:15 the DHW pump is put into operation (Figure 4b) in order to deliver the heat via the heat exchanger to the DHW tank. At ca. 5:20 the first and at ca. 5:37 the second AHP is switched off while the pumps for the heating water and cold water remain in operation for approx. 10 more minutes. The DHW pump stops at 5:50 the heat delivery at a supply temperature to the DHW tank of ca 50°C (Figure 4b) and the space heating distribution systems start operation again at the same time (Figure 4c).

Due to heat supply for space heating without recharging the storage by the AHPs the temperature of the heating water storage tank and consequently the supply temperature for space heating drops (Figure 4c), and because of that at 06:07 both AHPs are put into operation. Then the heating water temperature of the AHP and the space heating distribution system rises to ca. 42°C. At approx. 8:50 AHP_1 is switched off and the supply temperature level drops to ca. 32°C. Subsequently one AHP remains in operation for space heating which was also the starting point of this discussion at 4:30.

As discussed above, the system has been monitored from 28.12.2009 until 03.01.2011 which are 53 weeks of operation. Figure 5 (left) shows the average values of the energy input to the AHP: natural gas (Q_GAS), low temperature heat source (Q_ESC) and electrical energy (including the AHP control, heating water pumps and heat source pumps - W_AHP) for every week. Figure 5 (right) shows the overall energy balance of the monitoring period. The energy consumption of the system ranged from ca. 1 GJ in week 29 where only DHW preparation has been used to nearly 45 GJ in week 51 where the main energy demand was space heating. The yearly energy balance shows a total energy input of 870 GJ (calculated with the upper heating value of natural gas) and a total energy output of 800 GJ. The difference between in- and output amounts to approx. 8.8% of the energy input. Apart from errors of measurement this can be dedicated to different kind of losses, i.e. heat losses, flue gas losses, start/stop losses and stand by losses. These comparably small losses (compared to e.g. 22% reported by Bakker and Sijpheer 2008) may likely be achieved because of the heat exchanger for flue gas condensation combined with the very low return temperature level of the heat distribution system between 18 and 32°C (compare Figure 4c).The electricity demand for the AHP amounts to approx. 3% of the heat output of the AHP which is a reasonable low value.

Figure 6 (left) shows the weekly overall energy balance (energy input - output) for the monitoring period and Figure 6 (right) shows the energy balance of week 29 in detail, subdivided into the different kinds of energy. In winter the energy balances are almost consistent (the deviation is below 10%). During summer the deviation rises up to over 70%, when only DHW preparation is in use as also shown in Figure 6 for week 29. This is because the operation period for DHW preparation is only ca. 40 min with long standby periods and therefore the stand by losses and start/stop losses are large. Especially the electricity demand becomes significant in summer and amounts to approx 21% which can mainly be dedicated to standby losses of the AHP control. However, as already discussed and shown in Figure 5 (right) due to the much higher energy consumption during the heating season the effect of summer operation on the total yearly energy consumption is minor.

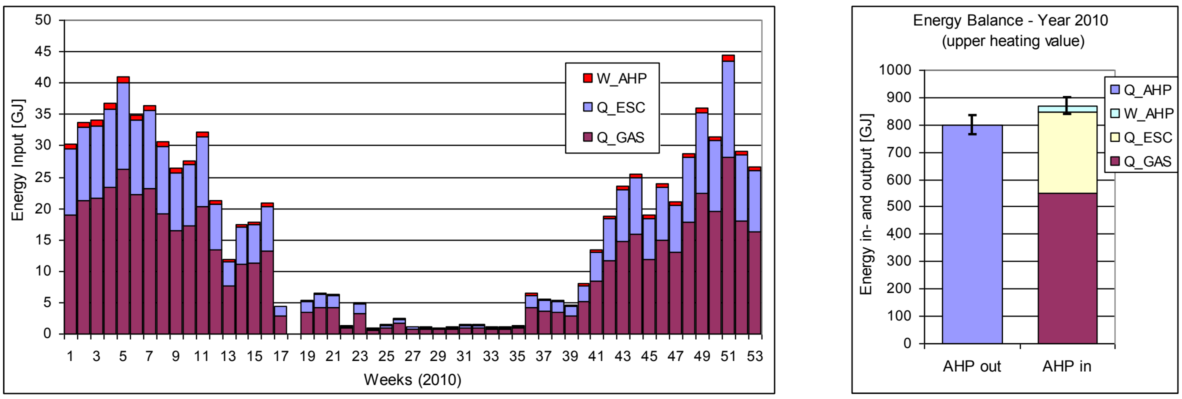
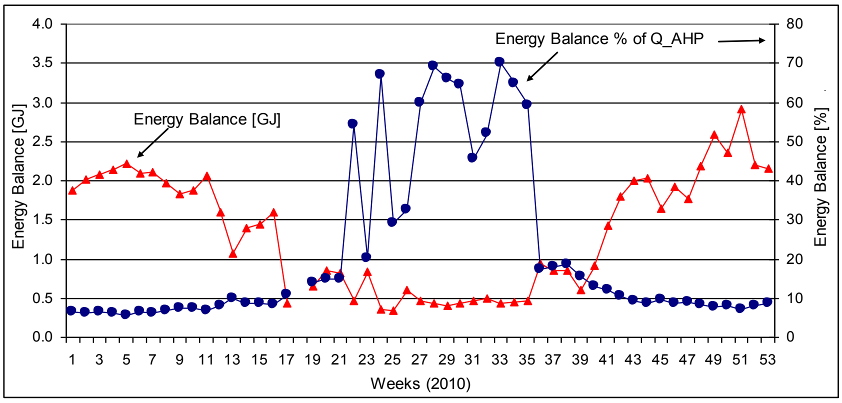


Figure 5: Average weekly energy input to the AHP (left) and energy balance of the AHP (right) for the year 2010



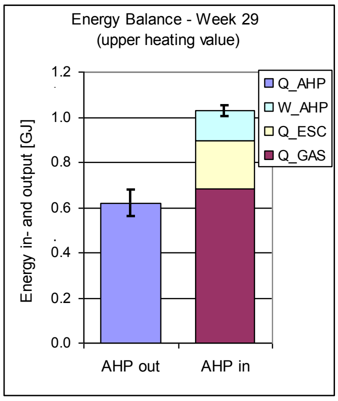
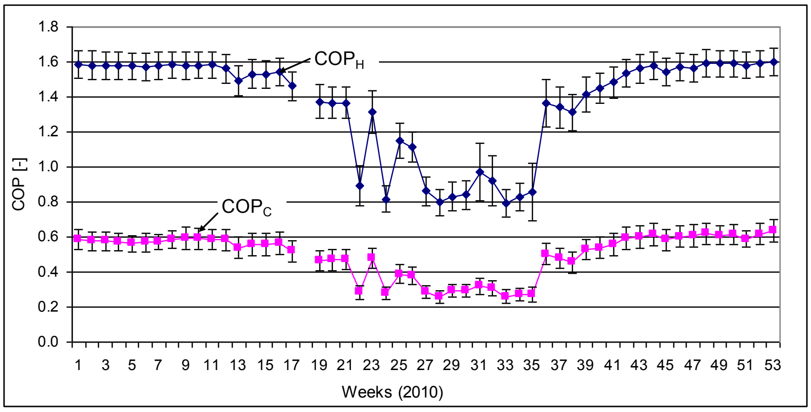


Figure 6: Average weekly energy balance based on upper heating value (input - output) for the year 2010 (left) and energy balance for week 29 - DHW preparation (right)

The average weekly COP for heating (COP_H) and cooling (COP_C) calculated with the lower heating value of the natural gas are shown in Figure 7 (left) where the calculated measurement uncertainties are shown with vertical bars. In winter, when the system is in continuous operation the COP_H is approx. 1.58 and the COP_C ca. 0.59. In summer when the system is mainly in start/stop operation for DHW preparation the COP_H drops to approx. 0.9 and the COP_C to ca. 0.3.

Even though the system is used for heating purpose only the COP_C shall be discussed in order to analyze the heat uptake from the heat source. If only the COP_H is analyzed one could come to the conclusion that the AHP process doesn't work correctly for DHW preparation in summer. The COP_C shows clearly that mainly losses are responsible for the low COP_H because even in summer the AHP absorbs approx. 30% of the energy input from the low temperature energy source.

The right diagram of Figure 7 shows the overall COP values for the year 2010 (which are equal to the yearly seasonal performance factors - SPF) for both calculated with the lower and upper heating value of natural gas. The overall COP for heating was 1.54 based on the lower heating value which is approx. 60% higher compared to a condensing gas boiler with a seasonal performance of 96%. The combined standard uncertainties for the seasonal performance factors for heating have been calculated to approx. ± 5.2% as shown in vertical bars.



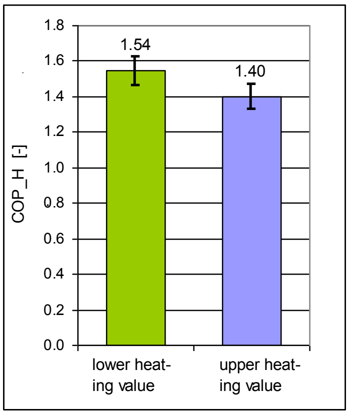


Figure 7: Average weekly COP (lower heating value) for heating and cooling (left) and overall COP for heating for the year 2010 calculated with the lower and upper heating value

5 CONCLUSION

The monitored medium size absorption heat pumping system for space heating and DHW preparation showed reliable operation and high energy performance over the monitoring period of one year. The energy consumption of the system ranged from ca. 1 GJ per week in summer when only DHW preparation has been used to nearly 45 GJ per week in winter when the main energy demand was space heating. In winter the energy balances are almost consistent (deviation below 10%). Apart from errors of measurement this can be dedicated to different kind of losses. The rather small losses in winter may likely be achieved because of the integrated heat exchanger for flue gas condensation combined with the very low return temperature level of the heat distribution system of max. 32°C.

In summer the deviation of the energy balances rises up to over 70%, when only DHW preparation is in use. These high losses are caused by relatively short operation periods for DHW preparation with long standby periods. Especially the electricity demand becomes significant in summer and amounts to approx 21% which can mainly be dedicated to electrical standby losses of the AHP control. However, due to the much higher energy consumption during the heating season the effect of summer operation on the total yearly energy consumption is minor.

Based on the lower heating value of the natural gas the seasonal performance for heating of the year 2010 was ca. 1.54 which is approx. 60% higher compared to a condensing gas boiler with a seasonal performance of 96%.

6 NOMENCLATURE

Symbols

COP	Coefficient of Performance	
c_p	Specific heat capacity	[kJ/(kg K)]
Δp	Pressure difference	[bar]
f	Functional relationship	
Hv	Heating value	[MJ/Nm³]
ρ	Density	[kg/m³]
Kv	Flow coefficient	[m³/h]
m	Mass	[kg]
$\dot{m}$	Mass flow rate	[kg/s]
p	Pressure	[bar]
Q	Thermal heat	[KJ]
$\dot{Q}$	Thermal capacity	[kW]
S	Status	
T	Temperature	[°C]
τ	Measurement interval	[s]
u	Standard uncertainty	
V	Volume	[m³]
$\dot{V}$	Volume flow	[m³/s]
W	Electrical energy	[KJ]
x	Variable x	
y	Variable y	

Subscripts

AHP	Absorption heat pumps
C	Cooling
DHW	Domestic hot water
DWS	Domestic hot water to supply tank
ENV	Environment
ESC	Low temperature energy source
FLG	Flue gas
FRC	Free cooling
GAS	Natural gas
H	Heating
HW1	Heating water distribution office
HW2	Heating water distribution depot
ROM	Room
i	index
in	inlet
out	outlet

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