Experimental Investigation of a Solar-Gas Driven HVAC System for Temperate Climate

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ABSTRACT

In a pilot installation at Hamburg University of Technology a solar-gas driven HVAC system is investigated experimentally. The coupled operation of a desiccant assisted air handling unit and an adsorption chiller enables a mainly heat driven air conditioning process. This paper presents an experimental evaluation of the system based on measurement data recorded during 2013. A performance comparison with a conventional system in terms of heat and electricity demand is conducted. It is found that through the investigated process up to 54.81 % of the electricity demanded by the reference system can be saved while the heat demand of the pilot plant is 4.8 times higher. Regarding the primary energy demand of both systems, the heat supply of the pilot plant is crucial. If a common gas boiler is used to support the solar thermal system, a solar fraction of at least 65 % is required to achieve primary energy savings. A modern small scale CHP engine leads to advantages for a well achievable solar fraction of 30 %. A bigger CHP engine leads to increased primary energy efficiency, even if no solar energy is used at all. The advantages of CHP engines can therefore be extended to the summer period.

1. INTRODUCTION

In order to ensure an acceptable indoor air quality, air conditioning systems have to cool down and dehumidify the outside air during warm and humid days. In a conventional system the air is cooled below dew point temperature (e.g. 283 K) to remove latent loads. Condensing out water requires large cooling capacities, which are usually provided by an electric motor driven vapor compression cycle and therefore causes a high demand for electric energy. In an open cycle desiccant assisted air conditioning system dehumidification and cooling can be separated within the process. First, moist air is dehumidified by using a solid or liquid desiccant. Afterwards, the dried air can be cooled by a heat sink at a higher temperature level (e.g. 290 K). For the regeneration of a desiccant wheel energy in form of heat at a temperature level of 310 K up to 340 K is sufficient. This enables the efficient use of alternative heat sinks and solar thermal energy. Furthermore, mainly energy as heat instead of electricity is required to run such kind of systems.

Several hybrid systems, relying on desiccant dehumidification and closed cycle cooling, have been investigated and proofed to be advantageous (Al-Alili *et al*., 2012; Fong *et al.*, 2011; Wrobel *et al.*, 2013; Angrisani *et al.*, 2011). While most systems rely on vapor compression chillers, Schmitz (2005) investigated the use of shallow geothermal energy to further reduce the electricity consumption. Another alternative heat sink, enabled through the desiccant wheel, can be an adsorption chiller, as proposed and simulated by Fong *et al.* (2010). Adsorption chillers achieve their best performance for higher temperature levels of the chilled water circuit and the energy in form of heat required to run the chiller can be supplied on similar temperature levels as the energy required for regeneration of the desiccant wheel.

A test facility at Hamburg University of Technology combines the mentioned aspects. Through the coupling of an open cycle desiccant assisted air conditioning system to an adsorption chiller the electricity demand of air conditioning is reduced further and a mainly heat driven process is achieved. In combination with the well-known advantages of CHP engines and adsorption heat pumps during the winter months, an efficient HVAC process for the whole year is ensured. This paper presents experimental results of the summer period 2013. The investigated system is compared to a conventional reference system regarding the electricity, heat and primary energy demand.

2. PILOT INSTALLATION

The test facility shown in Figure 1 consists of eight 20 ft containers. The four containers at the ground level contain the air handling unit as well as the further technical installations. The upper four containers serve as office room and are used as reference for the air conditioning system.

2.1 System Layout

The layout of the pilot installation is shown in Figure 2. It can be divided into two major parts, the air-handling unit and the hot and cold water circuit.

The air-handling unit is designed as a hybrid system, similar to the facility presented by Wrobel and Schmitz (2012). The outdoor air is dehumidified in a desiccant wheel (1-2) and precooled by the sensible heat regenerator (2-3). Both wheels exhibit a diameter of $0.65\ \mathrm{m}$, lithium chloride is used as desiccant. The air is then cooled down by a water/air heat exchanger (3-4) to achieve the desired supply air temperature. The extract air is preheated by the heat recovery wheel (5-6) before it is heated further to the required regeneration temperature (6-7). The air finally regenerates the desiccant wheel before it exits the installation. In comparison to a conventional air conditioning unit, the usage of a desiccant wheel to separate dehumidification and cooling reduces the cooling load and enables the coupling to heat sinks above the dew point temperature to the air handling unit (Wrobel and Schmitz, 2012; Angrisani *et al*., 2011).

The hot water required to run the adsorption chiller and regenerate the desiccant wheel is provided by a $20\ \mathrm{m}^2$ flat panel solar thermal system as well as a small scale CHP engine (Figure 1, nominal data: $P_{\mathrm{el}} = 5\ \mathrm{kW}$, $\dot{Q}_{\mathrm{th}} = 12.5\ \mathrm{kW}$). A 1000 l stratified thermal storage tank is used to decouple heat supply and demand. Another 800 l storage tank is used on the cold water side to supply the cooler and the cooling ceilings. The adsorption chiller (nominal data: $\dot{Q}_{\mathrm{ch}} = 10\ \mathrm{kW}$), shown in Figure 1, is connected with both storage tanks. Detailed manufacturer information about the chiller is provided in the data sheet (Invensor GmbH, 2013). A dry cooler is used for heat rejection, it is operated without a spraying kit during the whole year. The heat transfer fluid within the hot and cold water circuit is a mixture of water and ethylene glycol (21 Vol.-%).

Figure 1: Test facility, adsorption chiller and the CHP engine

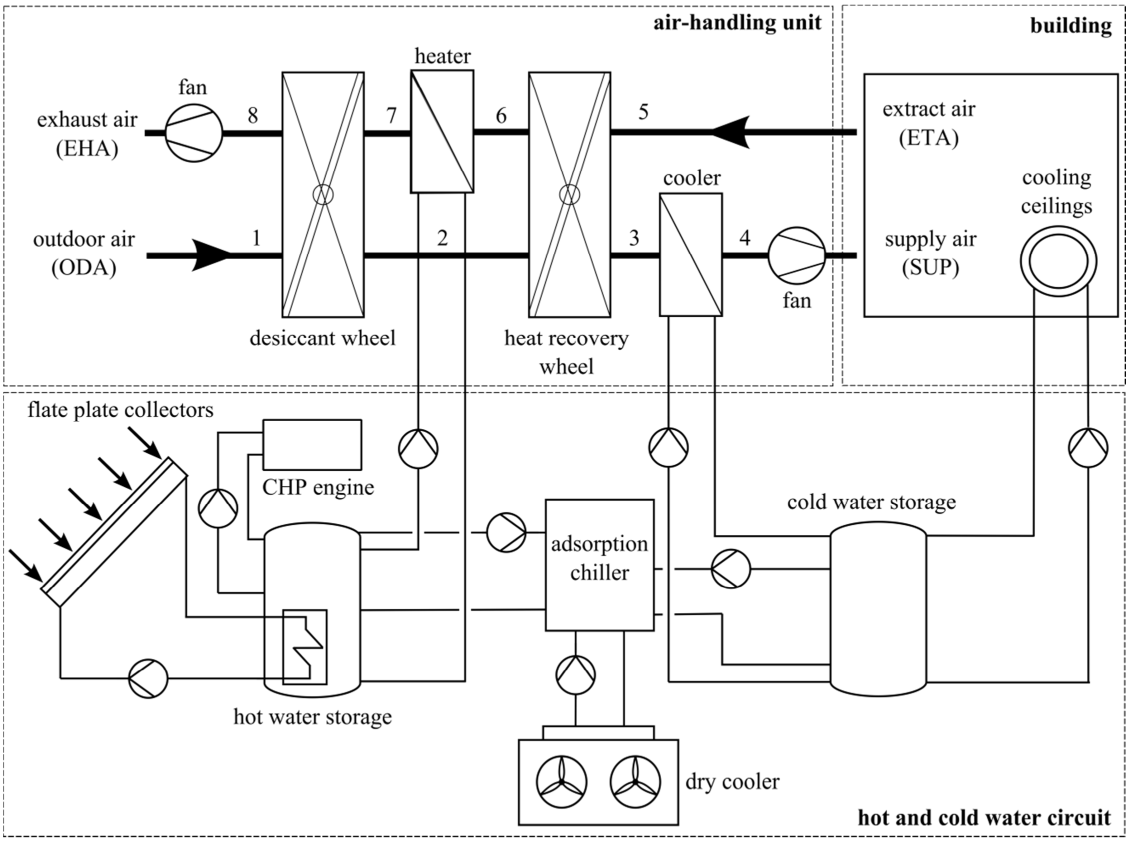

Figure 2: Layout of the pilot installation

2.2 Measurements and Data Acquisition

In order to investigate the transport and conversion of energy within the investigated system, the usage of adequate measurement devices is crucial. To evaluate the air handling unit and the connected reference room, temperature and humidity are measured at all stages of the process and return air, marked with numbers in Figure 2. The volume flow is measured at positions 4 and 8. Pressure drops are measured across all components of the air handling unit. The cold and hot water loops are evaluated by one volume flow measurement and two temperature measurements per component. Furthermore, the electricity demand of all devices connected to the grid is measured. Table 1 shows the selected types of sensors and their uncertainties.

Table 1: Measurement equipment used in the pilot plant

Measured property	Sensor Type or Principle	Uncertainty
Temperature, T	Resistance thermometer (PT100)	$\pm 1/3 \cdot (0.3+0.005 \cdot \boldsymbol{T})$
Relative humidity, φ	Capacitive humidity sensor	± 2 % r.h. for 10 % < φ < 90 %
Pressure difference, Δp	Ceramic fulcrum lever technology	± 2 % FS, (range: 0-300 Pa & 0-1000 Pa)
Volume Flow (air), $\dot{V}$	Differential pressure	± 10 %
Volume Flow (water), $\dot{V}$	Electromagnetic flow meter	± 0.5 % ± 1 mm/s
Electric Energy, W	AC energy meter	± 2 %

The volume flow of the air is determined by comparing the static pressure in front of the fan's inlet ring with the static pressure measured in the narrowest point of the inlet ring. A calibrated nozzle factor is provided by the manufacturer. To increase the quality of the volume flow measurement of the air and check the factor provided by the manufacturer, an onside calibration with an orifice plate measurement according to DIN EN ISO 5167-2 (2004) has been conducted. Relative humidity of the air is measured using capacitive sensors, which are usually subjected to a drift and can cause high inaccuracies in the evaluation of air conditioning systems. To take care of these effects, all sensors used in the following analyses have been calibrated at eight different values of relative humidity for four different temperatures, using a dew point sensor as reference. The correction of each measured value is then achieved by a linear interpolation between the respective 32 values. Furthermore, to ensure a reliable evaluation of the adsorption chiller, all six temperature sensors located in its hydraulic loops were checked in a reference bath, each two of them exhibiting similar characteristics were used for measurements in one hydraulic loop. Due to the small occurring temperature differences an accurate temperature measurement is crucial to evaluate an adsorption chiller. All measured data of the air handling unit and the hydraulic cycles are recorded every 60 seconds. Taking into account the dynamic behavior of the adsorption chiller, all data associated with the chiller are recorded every second.

3. SYSTEM PERFORMANCE

To evaluate the performance of the pilot plant, monitoring results of the cooling period 2013 are presented. The whole system is evaluated by comparing its heat and electricity demand to the respective demands of a reference system relying on dehumidification through condensation, which is achieved using a vapor compression chiller. Finally, the primary energy consumption of both systems is compared for different efficiencies of electricity and heat supply. Especially, the utilization of different CHP engines is investigated.

3.1 Test Period and Monitoring Results

Due to construction works, influencing the pilot plant, feasible data could not be recorded during the whole summer of 2013. Table 2 shows the ambient and supply air conditions for the days evaluated in this study, averaged over the operation time t_{o} of the air conditioning system. Furthermore, the chilling energy $Q_{\mathrm{ch,d}}$ provided by the adsorption chiller during the respective days is included in Table 2. All quantities are presented with the occurring measurement errors.

Table 2: Selected days for the system evaluation

	t_{o}	$T_{\mathrm{out}}/T_{\mathrm{out,max}}$	T_{sup}	w_{out}	w_{sup}	$\dot{V}_{\mathrm{sup}}$	$Q_{\mathrm{ch,d}}$
	min	°C	°C	$\mathrm{g_w/kg_{air}}$	$\mathrm{g_w/kg_{air}}$	$\mathrm{m^3/h}$	kWh
July 19th	581	22.0 / 23.8	22.4	10.5 ± 0.4	7.2 ± 0.4	826 ± 83	18.4 ± 3.6
July 25th	464	25.9 / 29.1	22.2	11.7 ± 0.5	7.2 ± 0.4	817 ± 82	29.6 ± 6.1
July 29th	529	25.3 / 26.8	22.1	11.9 ± 0.5	7.2 ± 0.4	817 ± 82	36.2 ± 6.4
August 1st	619	24.5 / 29.3	22.3	12.4 ± 0.5	7.2 ± 0.4	816 ± 82	31.5 ± 6.5
August 2nd	633	29.8 / 33.3	22.2	12.1 ± 0.6	7.6 ± 0.4	811 ± 81	45.2 ± 10.7
August 6th	639	24.9 / 26.1	22.1	10.5 ± 0.5	6.6 ± 0.4	819 ± 82	36.5 ± 6.8
August 8th	660	19.6 / 21.1	22.1	10.7 ± 0.3	6.5 ± 0.4	820 ± 82	19.9 ± 3.8

A detailed performance evaluation of the adsorption chiller during the respective days can be found in Speerforck and Schmitz (2014).

3.2 System Comparison

In order to evaluate the presented solar-gas driven air conditioning system, a suitable reference process is required. In this study the pilot installation is compared to a conventional air conditioning system relying on dehumidification through condensation and a vapor compression chiller. The Air Handling Unit of the reference system is shown in Figure 3.

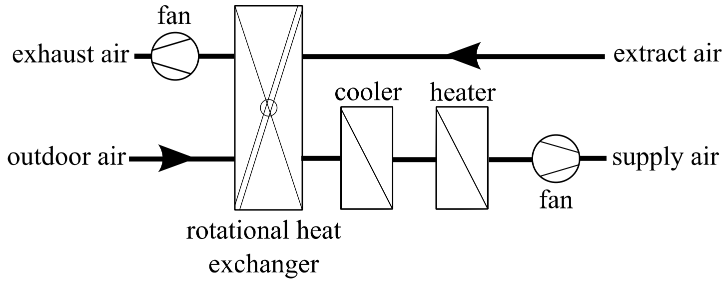

Figure 3: Air handling unit of the reference system

The key assumptions made regarding the reference process are:

- Supply air water content and mass flow rate: measured data of pilot installation,
- Supply air temperature: 16 °C,
- Seasonal Performance Factor (SPF = $\frac{Q_{\text{sea}}}{W_{\text{sea}}}$) of the vapor compression chiller including a dry cooling process for heat rejection, SPF: 3.

The Seasonal Performance Factor of the vapor compression chiller is chosen slightly higher than recommended by Napolitano *et al*., (2011) and in good accordance with simulation and measurement results provided by Angrisani *et al*., (2011). Due to the lower supply air temperature level of the reference system the sensible load removed by the air stream increases. The respective difference is credited to the conventional process, in order to ensure a meaningful comparison. The parasitic energy consumption of the reference system is calculated based on measured data of the pilot plant. Furthermore, the lower electricity consumption of the fans due to the absence of the desiccant wheel and the regeneration heat exchanger is taken into account.

Figure 4 shows the demands of electricity and energy as heat of both systems summed up for the whole investigated period. The displayed amounts are based on 24 h measurements recorded during the days listed in Table 1. Standby consumptions of all electric components are included.

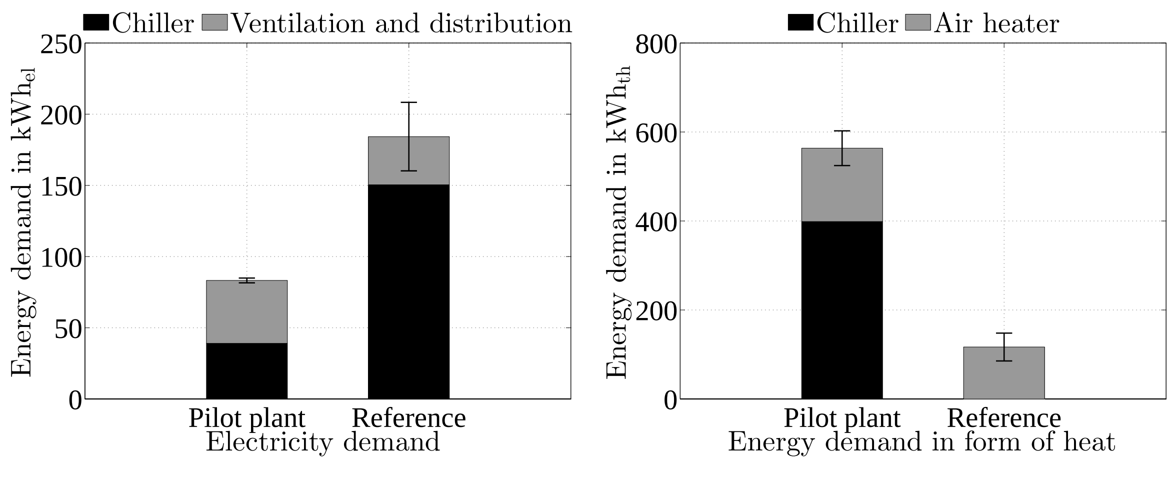

Figure 4 : Energy demands of the pilot plant and the reference system for the whole period

Regarding the electricity demand, 54.81 % of the amount required to run the reference system can be saved by the pilot plant. However, the adsorption chiller, the associated pumps and the dry cooler still account for nearly half (47 %) of the electricity demand of the whole plant. Even though, the chiller itself might almost exclusively rely on heat as energy input, the components required to run it still consume a considerable amount of electrical energy. Another important factor within the electricity demand of the pilot plant is the standby consumption of the used components, accounting for 9.2 kWh_{el} (11 %). While the electric energy demand of the pilot plant is measured directly with accurate sensors, the demand of the reference system is calculated using the assumptions listed above. As the volume flow and the water content of the supply air are important quantities for the comparison, their measurement error is influencing the electric energy demand of the reference system.

Regarding the heat demand, the result is *vice versa*. The amount of energy as heat demanded by the pilot plant (563.5 kWh_{th}) is 4.8 times bigger as the amount required to run the reference system (116.8 kWh_{th})[1]. The biggest part of the difference regarding the heat demand is caused by the adsorption chiller, which is accounting for 70 % of the overall demand for the pilot plant. As the energy in form of heat required for the regeneration of the desiccant wheel and to run the adsorption chiller are determined by flow and temperature measurements, the low temperature difference at the hot water loop of the adsorption chiller is mainly influencing the measurement uncertainty. The relatively high uncertainty for the heat demand of the reference system is caused by the uncertainty of the airflow measurement as well as the supply air water content of the pilot plant.

According to Figure 4, the electricity saved by through the pilot plant is accompanied by a considerable amount of energy in form of heat required to regenerate the desiccant wheel and to drive the adsorption chiller. Whether the pilot plant or the reference system is more suitable from a primary energy perspective is highly dependent on the type of heat generation. In order to achieve a suitable comparison of both systems, Figure 5 shows the primary energy savings Δ_{PE} generated by the pilot plant for different types of heat generation. In all cases solar energy is used to generate a fraction of the heat required to run both systems. The remaining fraction is provided by a boiler or CHP engines with different efficiencies for electricity and heat generation. The efficiency of electricity generation is chosen according to the boundary conditions prevalent in Germany, $f_{p,el} = 2$ (DIN V 18599-1, 2011)[2].

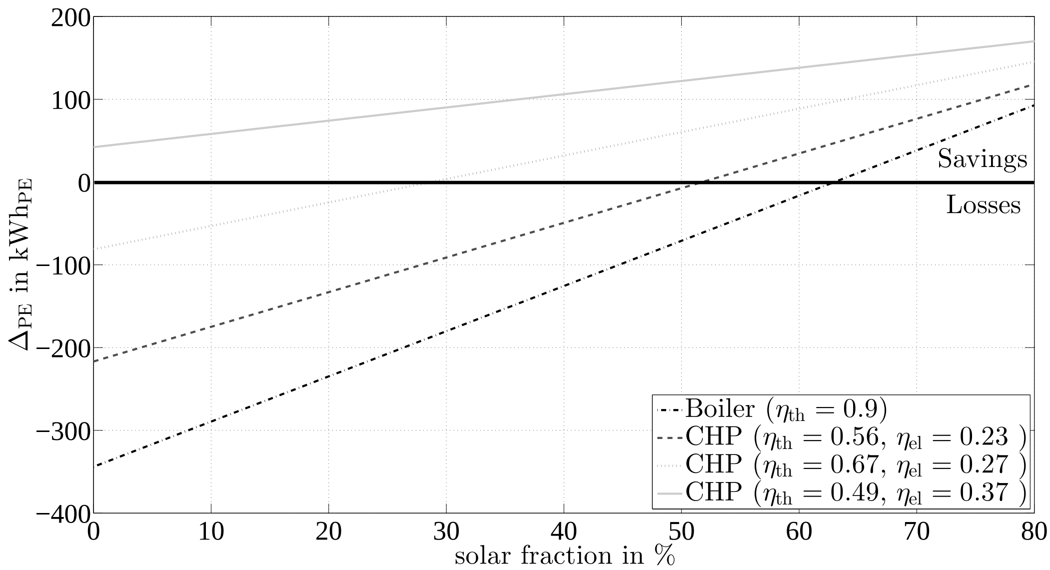

Figure 5: Primary energy savings achieved by the pilot plant in comparison to the reference system for different types of heat generation

As shown by Figure 5, the highest solar fraction (65 %) is required if a common gas boiler is used to support the solar thermal system. A well sized storage tank and a sufficient area of collectors are therefore mandatory to ensure primary energy savings through the mentioned system. In combination with a relatively inefficient CHP engine ($\eta_{el} = 23$ %, $\eta_{th} = 56$ %, $P_{el} = 5$ kW, as used in the pilot plant) the system leads to primary energy savings if at

[1] Note that the exergy content of the energy as heat is not taken into account in this comparison.
[2] According to DIN V 18599-1 (2011) the electricity provided by the CHP engine is credited with a factor of $f_{p,c} = 2.5$. The factor for transportation and extraction of natural gas is $f_{p,G} = 1.1$. The electricity demand of the required pump for the solar thermal circuit is neglected.

least a solar fraction of 53% is achieved. While a state of the art small scale CHP engine ($\eta_{\mathrm{el}} = 23$ %, $\eta_{\mathrm{th}} = 56$ %, $P_{\mathrm{el}} = 6$ kW) leads to primary energy savings as soon as a solar fraction of 30 % is realized, a modern bigger scale CHP engine ($\eta_{\mathrm{el}} = 37$ %, $\eta_{\mathrm{th}} = 49$ %, $P_{\mathrm{el}} = 200$ kW) leads to primary energy saving even if no solar energy is used at all. Therefore, the efficiency of the heat generation process used to support the solar thermal system is crucial. Used in combination with modern CHP engines the process enables considerable primary energy savings and a mainly gas driven air conditioning process. The high utilization of electricity grids due to air conditioning processes during the summer months can be reduced and the rate of utilization for CHP engines can be increased significantly.

4. CONCLUSIONS

A solar-gas driven HVAC system for dehumidification and cooling of outside air can be achieved through the coupling of an open cycle desiccant assisted air conditioning system and an adsorption chiller. Compared to a conventional reference system based on dehumidification through condensation the electricity demand can be reduced by 54.81 %. However, the heat demand is 4.8 times higher for the investigated system. Primary energy savings achieved through the desiccant assisted system are highly dependent on the boundary conditions and the efficiency of heat supply. While the utilization of a common gas boiler requires a solar fraction of 65 % in order to save primary energy, a modern small scale CHP engine leads to advantages for a well achievable solar fraction of 30%. A bigger engine leads to increased primary energy efficiency, even if no solar energy is used at all. The presented system is therefore a promising possibility to increase the rate of utilization for CHP engines during the summer months and to lower electricity peak demands. Combined with the well-known advantages of CHP generation and adsorption heat pumps during the winter months, an efficient HVAC process during the whole year is ensured.

NOMENCLATURE

COP	Coefficient of Performance	(-)
f	primary energy factor	(-)
P	electrical power	(W)
p	pressure	(Pa)
Q	energy as heat	(J)
$\dot{Q}$	power as heat	(W)
SPF	Seasonal Performance Factor	(-)
T	temperature	(°C)
t	time	(min)
$\dot{V}$	volume flow	(m^3/h)
W	electrical energy	(J)
w	air humidity ratio	(g/kg)
Δ	difference	(-)
φ	relative humidity	(%)

Subscripts

c	credit
ch	chill
d	day
el	electrical
G	gas
o	operation
out	outside
p	primary energy
sea	season
sup	supply
th	thermal
w	water

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