# Abstracts of Heat Powered Cycles 2009

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THE USE OF PINCH ANALYSIS IN THE DESIGNING OF A DISTRIBUTED TRIGENERATION APPLICATION

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Abstract

Miscellaneous thermodynamic and thermo-economic theories are used to analyze distributed trigeneration systems to reveal the advantages compared with conventional systems. However, these theories cannot help practical designing, as it is almost impossible to enumerate detailed plans to analyze them all. Thus, a methodology for distributed trigeneration designing is needful.

In this paper, improved Pinch Analysis is applied. In the winter mode of a trigeneration system, the problems in designing as discontinuous composite curve, large minimum temperature approach and threshold problem are solved to obtain an optimized heat exchanger network. As for the summer mode, the placement of heat engine and chillers are discussed; the availability and feasibility of absorption chiller and electric chiller are compared. Based on the designing with improved Pinch Analysis, the new trigeneration with a gas engine, exhaust driven absorption chiller, heat exchangers and vacuum boiler can save 10-30\% prime energy consumption, which substitutes a conventional system with the grids and coal-fueled boilers in an 18-storey hotel and office building complex.

References

Figures and Tables

Figure 1: Demand forecasting of the new system in summer

Figure 4: Grid diagram of the original HEN in winter

Figure 9: Composite curve with absorption chiller in summer

Figure 10: Schematic of the new system plan
ABSTRACT

Among the newly developed heat powered cycles, rotary desiccant cooling, which works in open cycle and is advantageous in using low grade thermal energy, controlling humidity independently and only using free water and air, has been considered as a potential cooling technology. However, such method, which normally can produce efficient dry air, can not cool the process air to a satisfied state for some times. To further improve the performance of desiccant cooling, isothermal dehumidification and regenerative evaporative cooling are of most importance, since the measures can minimize the irreversibility of dehumidification and cool the process air sufficiently.

In this paper, an innovative open cycle cooling method, which combines the technologies of two stage rotary desiccant dehumidification and regenerative evaporative cooling, has been proposed and investigated in this paper. Here, the two stage desiccant dehumidification with internal cooling is a feasible way to minimize the adsorption heat and approach the isothermal dehumidification. This method can simultaneously dehumidify the fresh process air and produce chilled water for space cooling. A mathematical model has been established to predict the performance. Case study has also been conducted under given conditions in order to check the feasibility and energy saving potential. The effects of chilled water flow rate, dry air distribution, inlet air conditions, regeneration temperature and solar radiant intensity, etc., are analyzed. The proposed method is more environmental friendly, as it works on thermal energy and realizes the independent temperature and humidity control just using free water and air, and without any assistance from traditional vapor compression unit.

DESCRIPTION OF THE CYCLE

Figure 1 illustrates the working principle of the proposed method. As seen, the unit generally consists of two parts, namely, desiccant dehumidification and regenerative evaporative cooling.

Desiccant dehumidification

The process air cycle is as follows: Process air at state point 1 passes through section I of the desiccant wheel (DW), where it is dehumidified and heated due to the adsorption heat effect. Then this hot dry air is sensibly cooled from state point 2 to 3 in a cross-flow heat exchanger (CHE). Afterwards, the process air passes through section II of the DW, where its moisture is further removed. And then in another CHE, the process air (state 4) is sensibly cooled with a dry air output at state point 5.

The regeneration air cycle is as follows: The regeneration air is first cooled and humidified in a direct evaporative cooler (DEC) from state point 11 to 12. Then it is divided into two groups, which work in parallel (state 12-13-14-15; state 12-16-17-18). These two air streams are sensibly preheated to state points 13 and 16 in the CHEs. The warm air streams are then further heated to state points 14 and 17 in the solar air collectors, with two thirds of them bypassing the collectors each to reduce regeneration heat consumption without substantial influence on dehumidification capacity. Whereafter, these hot air streams flow through section III and section IV of the DW to desorb water vapor and regenerate the desiccant, and exit at state points 15 and 18.
Regenerative evaporative cooling

After being split at state point 5, the dehumidified air flows in two different paths (state 5-6-7-8-9-10; state 5-9-10). The part used for making chilled water is first pre-cooled to state point 6 in a CHE. Then it passes through a cross-flow evaporative cooler (CEC). With the pressure difference between the water vapour pressure of air and the saturation water vapor pressure, the water is cooled due to the evaporative cooling effect, thus chilled water can be produced. At the same time, the air is humidified and cooled to state point 7. This cold humid air is redirected to the CHE to cool the air supplied to the CEC and exits at state point 8. Then, this part of air is reused by mixing with the remaining part of dehumidified air from the desiccant dehumidification subsystem partly or totally. The process air (state 9) is then sensibly cooled to state point 10 by the chilled water in the air-to-water heat exchanger (HE).

CONCLUSIONS

Conclusions generated by this investigation are as follows: (1) the proposed method can achieve a thermal COP above 1.0 and an electric COP about 8.0, while corresponding temperature of produced chilled water is around 14-20 °C, thereby realizing separate temperature and humidity control without the assistance of vapor compression system; (2) although the impacts of inlet air conditions and regeneration temperature on performance are found to be significant, the thermal COP of the system is sounded in most cases; (3) there exists an optimal chilled water flow rate and process air distribution ratio to obtain the biggest cooling capacity of chilled water; (4) For solar air conditioning application, theoretical analysis shows that the system can convert more than 50% of the received solar radiation to the capability of air conditioning in sunny days.
A SOLAR COOLING/HEATING SYSTEM FOR A LABORATORY BUILDING

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Abstract

This paper presents the original design concept of a solar cooling/heating system which will be installed in the mechanical engineering laboratories of the Cyprus University of Technology. The system will consist of a LiBr-water absorption unit, a stationary solar collector system, storage tanks, cooling towers and the necessary control and distribution system. As the area where the laboratories are situated has a lot of ground water at a shallow depth, an attempt will be made to relieve part of the condenser load using the ground water as a means of heat dissipation.

Absorption machines are thermally activated and they do not require high input shaft power. Therefore, where power is unavailable or expensive, or where there is waste, geothermal or solar heat available, absorption machines could provide reliable and quiet cooling. Absorption systems are similar to vapor-compression air conditioning systems but differ in the pressurization stage. In general an absorbent, on the low-pressure side, absorbs an evaporating refrigerant. The most usual combinations of fluids include lithium bromide-water (LiBr-H2O) where water vapor is the refrigerant and ammonia-water (NH3-H2O) systems where ammonia is the refrigerant.

The NH3-H2O system is more complicated than the LiBr-H2O system, since it needs a rectifying column that assures that no water vapor enters the evaporator where it could freeze. The NH3-H2O system requires generator temperatures in the range of 125 to 170°C with air-cooled absorber and condenser and 95 to 120°C when water-cooling is used. LiBr–H2O systems are also friendlier to the environment as they work at sub-atmospheric pressures and when there is a leakage no contamination occurs. For these reasons the LiBr-H2O system is preferred in this design.

Lithium bromide-water chillers are available in two types, the single and the double effect. The single effect absorption chiller is mainly used for building cooling loads, where chilled water is required at 6-7°C. The coefficient of performance (COP), which is defined as the ratio of the cooling effect to the heat input, varies to a small extent (0.65-0.75) with the heat source and the cooling water temperatures (Florides et al., 2002a). Single effect chillers can operate with a hot water temperature ranging from about 80°C to 120°C when water is pressurized, whereas for the double effect much higher temperatures are required. For this reason the single effect unit is preferred since the lower temperatures are easily obtained with stationary solar collectors.

It should be noted that the refrigerant in the water-lithium bromide system is water and the LiBr acts as the absorbent, which absorbs the water vapor thus making pumping from the absorber to the generator easier and economic. A single-effect, two shell, LiBr -water chiller is illustrated in Figure 1, where typical temperatures are shown. At point (1) the solution is rich in refrigerant and a pump forces the liquid through a heat exchanger to the generator (3). The temperature of the solution in the heat exchanger is increased. In the generator thermal energy is added and refrigerant boils off the solution. The refrigerant vapour (7) flows to the condenser, where heat is rejected as the refrigerant condenses. The condensed liquid (8) flows through a flow restrictor to the evaporator (9). In the evaporator, the heat from the load evaporates the refrigerant, which flows back to the absorber (10). A small portion of the refrigerant leaves the evaporator as liquid spillover (11). At the generator exit (4), the steam consists of absorbent-refrigerant solution, which is cooled in the heat exchanger. From points (6) to (1), the solution absorbs refrigerant vapour from the evaporator and rejects heat through a heat exchanger.

The complete solar system that will be used in the laboratories is shown schematically in Figure 2. The design temperature of the system will be 70 to 90°C so as to avoid having a pressurized storage. The solar collectors to be used are of the evacuated tube type, which have a good
efficiency at this working temperature. The total area of the laboratories is 1400 m$^2$. The building is an existing one that will be renovated and has limestone walls, 50cm in thickness. The roof will be reconstructed and covered with 15cm insulating material and the existing doors and windows will be replaced with double glazed ones. The maximum cooling load of the laboratories is of the order of 250 kW. A total of 12,000 liters storage cylinders will be used. Such a system was evaluated in the past (Florides et al., 2002b) and found to have a low total equivalent warming impact (TEWI).

The scope of this solar application is to evaluate the technology in the Cyprus environment and for this reason the system will be fully monitored for a number of years. This will lead to conclusions whether the Department of Energy will partly subsidize these types of systems in the future and at what extent. Electronic display boards will also be installed showing at any time the working condition of the various parts of the system and their contribution to the load. This will be done for the purpose of demonstrating to the general public the benefits of this solar application.

References


OPERATING EXPERIENCE OF ABSORPTION COOLING
FROM LOW GRADE HEAT SOURCES IN THE RANGE 25 kW-40 kW

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ABSTRACT

In 2006 an ammonia/water absorption system was described technically and evaluated thermodynamically and economically. It can provide a refrigeration load between 25 kW to 40 kW with a driving heat input at temperatures down to 90 °C. Present applications for cooling temperatures at 0°C and below are of complex design and restricted to cooling capacities above 100 kW. The decisive cost advantage with increasing unit capacity and the resulting high efficiency enables economic operation. Sorption refrigeration systems with cooling capacities of less than 30 kW, low cooling temperatures, and low-grade heat as the driving energy are currently not available. In this range they could contribute considerably to rational energy supply in decentralised energy systems, especially combined with solar energy, trigeneration, or ice-storage.

The aim of this project was to develop a sorption refrigerator, of minimal complexity, to provide cooling in an efficient manner as possible which would utilise low grade heat. Systems of this type tend to be less efficient due to the use of low-grade heat. The technical specification is representatively quantified by a cooling capacity of 25 kW at -1 °C mean coolant temperature, and 90 °C mean temperature of the heating fluid. These assumptions imply ammonia-water absorption technology to achieve the low coolant temperature, while stainless steel should prevent corrosion and fully welded heat exchangers should ensure long mean time between failures. A single-stage system was selected. The data gained from a detailed calculation of the real process was verified experimentally by investigations on an established pilot plant. With the results of the system flow scheme an optimised system could be defined. A Second law analysis determined the deviation from the ideal and specifies the exergy losses of the individual components. The findings led to a particularly simple system without a rectifier, reflux condenser, or refrigerant subcooler. The energy efficiency obtained the so-called coefficient of performance, amounts to 47%. This value appears to be comparatively modest, but the efficiency is of minor importance if low cost waste energy can be employed and the low temperature levels are taken into account. From the thermodynamic point of view the exergy input is transferred to the coolant at a rate of 28%. This magnitude represents an exergy efficiency comparable to complex absorption refrigeration units driven by high-grade heat.

The optimised small scale ammonia-water absorption system constitutes a valuable complement for decentralised energy systems. Adapted to the specific local demand with respect to permanent operation, and making use of available heat sources, a competitive economic solution is achieved.

In 2008 the first implementation of the proposed advanced absorption refrigeration system was completed. It is utilised for air-conditioning of an administration building. The system is driven by district heat, while additional ice storage allows to shave peak loads during summertime. The process data to be gained can then be compared to available data of other heat driven refrigeration systems.
THE THERMODYNAMIC EQUILIBRIUM DROP OF REACTIVE MEDIUM IN RESORPTION SYSTEM


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Abstract

Resorption systems are similar to conventional chemisorption systems used in heating and cooling applications. Both types of system have cyclic operation, where a metallic salt placed inside a reactor may absorb or desorb refrigerant vapor, depending on the phase of the cycle. However, in the former type of system, a second reactor, which contains another type of salt, replaces the evaporator and the condenser. Under the same pressure, the salt inside the first reactor has higher equilibrium temperature than the salt in the second reactor; thus, they are called high temperature salt (HTS) and low temperature salt (LTS), respectively. Moreover, the resorption cycle has one high-pressure phase (HPP), when the HTS desorbs refrigerant and the LTS absorbs it; and one low-pressure phase (LPP), when the reaction in each salt reverse its direction.

The synthesis and decomposition reactions can only occur when the salts are removed from their equilibrium. The difference between the working temperature and the equilibrium temperature is called temperature equilibrium-drop ($\Delta T_{eq}$), and which is the driving force for the reaction. Because the kinetics of the reaction of different salts are not affected in the same way by $\Delta T_{eq}$, we studied how the degree of conversion in the synthesis and in the decomposition of ammoniates inside the reactors of three different resorption systems varied with the equilibrium drop, during specified periods. Such a type of information is useful in the choice of the proper salts, equilibrium drops and working conditions that may lead to minimization of the reaction time and maximization of the coefficient of performance.

Experiments and results

Each resorption bench-scale prototype was composed by two reactors. The diameter of the reactors was 60 cm and the height varied between 88 and 147 cm. Manganese chloride was the HTS in all the machines, but the LTS was different for each machine. The LTS used were NH$_4$Cl, NaBr and BaCl$_2$. The synthesis and decomposition reactions were studied independently, and the initial degree of conversion in all experiments were $0.00 \pm 0.01$.

The experiments mainly consisted in assess with an electronic balance, the mass of ammonia that the reactors exchanged at different working conditions. Thermoresistances Pt100 indicated the temperature inside the reactors, and a piezoelectric pressure sensor indicated the pressure of the system.

Figure 1 shows how the conversion of the different LTS changed with the $\Delta T_{eq}$ during the HP and LP phases. The conversion in the HP phase was higher than that in the LP phase for all the salts, although the $\Delta T_{eq}$ and the reaction time were smaller. Such a result indicated that for the operation conditions studied, the critical temperatures are those of the LP phase. NH$_4$Cl had the highest conversion among the LTS tested, even in a condition where its $\Delta T_{eq}$ was smaller than that of the other salts. In the conditions studied for the HP phase, the conversion increased
linearly with the $\Delta T_{eq}$ for NH$_4$Cl and NaBr; however for BaCl$_2$, it reached 100% when $\Delta T_{eq}$ was 8°C, even though the next working condition could provide higher equilibrium drop but that was a energy waste when the degree of conversion reached 100%.

Regarding the LP phase, for a same $\Delta T_{eq}$, the reaction of the ammoniate of NH$_4$Cl had higher conversion than the reaction of the ammoniates of NaBr and BaCl$_2$. In the range of $\Delta T_{eq}$ between 10 and 20 °C, the degree of decomposition of the NH$_4$Cl ammoniate was constant; whereas the degree of decomposition of the NaBr ammoniate increased linearly from 77 to 87%, in the range of $\Delta T_{eq}$ between 10 and 15 °C. The worst conversion was obtained with BaCl$_2$: In the range of $\Delta T_{eq}$ between 17 and 27 °C, the conversion only increase from about 5 to 45%.

![Figure 1: The degree of conversions vs. $\Delta T_{eq}$ of LTS coupled to MnCl$_2$ as HTS. (■)=NH$_4$Cl(HPP); ◆=NaBr(HPP); ●=BaCl$_2$(HPP); □=NH$_4$Cl(LPP); ◇= NaBr(LPP ); ○=BaCl$_2$(LPP))](image)

**Summary conclusions**

Among the three resortion systems tested, the one using NH$_4$Cl is the most suitable for applications requiring temperatures below 0°C. The system with NaBr would be the second option, whereas the system with BaCl$_2$ would not be feasible.
A STUDY OF A DIESEL GENSET BASED TRIGENERATION RUNNING WITH RAW JATROPHA OIL

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ABSTRACT

The performance and the efficiency of a trigeneration system fueling with pure diesel and with raw jatropha oil are investigated using the ECLIPSE software. The study is based on a diesel engine generating set. The genset is used for electrical power generation only, acting as a single generation (SG). The trigeneration system consists of the genset, a waste heat recovery system and an absorption refrigerator. The genset is used to generate electricity; the waste heat system is used to collect the waste heat from the cooling system and the exhaust gas, to supply heating/hot water; and the absorption refrigerator is used to supply cooling/refrigeration, which is driven by the waste heat from the engine instead of electricity. Therefore, it forms a ‘trigeneration’. The trigeneration generates electricity, heat and refrigeration simultaneously with only a fuel energy input. Using the renewable jatropha oil, the trigeneration system may be seen as a ‘carbon’ neutral system, which will benefit the environment. A comparison of the thermal efficiencies and the CO₂ emissions of trigeneration with single generation and cogeneration are carried out (see Fig 2 and Fig 3). The results from the study show that the thermal efficiency of trigeneration is higher than that of single generation; the CO₂ emissions of trigeneration are lower than that of single generation. The results also show some performance differences between the trigeneration and single generation; and the differences between trigeneration and cogeneration.

Fig 1 The Schematic Diagram of the proposed trigeneration system
Fig 2  Comparison of Efficiencies Using Diesel/Jatropha Oil

Fig 3  Comparisons of CO₂ Emissions for Different Generations Using Jatropha Oil & Diesel
Abstract

Several investigations have pointed the operation behavior of passive thermal control devices such as loop heat pipes (LHPs) and pulsating heat pipes (PHPs) in the past, which have cleared many points related to their design. However, an interesting aspect related to passive thermal control devices that operated by means of capillary forces to pump the working fluid have gained attention during the last years, which is in regard to the use of nanofluids on such devices. Nanofluids are known as regular fluids with addition of solid nanoparticles with sizes (diameter) below 40 nm, which are used to enhance the working fluid’s thermal performance by enhancing its thermal conductivity. Previous works have demonstrated that the liquid’s thermal conductivity can be enhanced by 20% if nanoparticles are added on a concentration of 5% by mass. PHPs operate by the dynamics of slug/plug formation removing heat from a high temperature source and dissipating in a low temperature sink and are highly influenced by the bubble critical diameter related to a specific working fluid. Thus, an experimental open loop PHP (OLPHP) was tested with water-copper nanofluid, with an addition of 5% by mass of copper nanoparticles. Improvements on the overall device’s operation have been observed when using the nanofluid with lower temperatures, as well as a direct influence on the thermal resistances throughout the PHP. Further analysis has shown that the addition of solid nanoparticle on the working fluid has directly contributed for the improvement of the OLPHP thermal performance.

Introduction

Nanofluid application is a recent area of investigation with promising results for thermal control systems. Basically, nanofluids are working fluids that have just started been applied in thermal system devices, using nanoparticles of solid materials used to improve the fluids’ thermal conductivity. By adding 5% of the working fluid mass with nanoparticles, the liquid thermal conductivity can be increased by up to 20% (Koo and Kleinstreuer, 2004). Some researches have already presented important contributions using nanofluids usually composed of water and copper nanoparticles with sizes around 25 nm (Koo and Kleinstreuer, 2004; Chein and Huang, 2005), which all represent the recent advances on this new and innovating technology. Investigations performed so far have pointed to the potential in using nanofluids in several thermal control applications with great improvement on the heat transfer coefficient, especially when liquid single-phase thermal control has been used. It is important, however, to mention that investigations performed so far utilizes regular pumping devices to transport the nanofluid throughout the loop (Park and Jung, 2007). However, very little is known about nanofluids application in devices such as loop heat pipes (LHPs) that require the generation of capillary forces to drive the working fluid (. In this last case, the capillary evaporator presents a porous wick structure with fine pores and the interaction with the nanofluid needs to be better investigated. The nanofluid is composed of a pure substance, like water, with solid nanoparticles usually mixed with mass fractions from 1 to 5 %. The nanoparticles are materials with size below 100 nm in diameter and should be as pure as possible to avoid any kind of chemical reaction of the substance that has been mixed. Investigations have already presented the increase on the liquid thermal conductivity of the nanofluid when compared with the pure substance by as much as 20% (depending on the nanoparticle material) when a mass fraction of up to 5% of nanoparticle was added (Koo and Kleinstreuer, 2004).

Experimental results and discussion

The PHP was operated without pre-conditioning procedures. Prior to start the skin heater placed on the evaporation section, the fan on the condensation section was turned on and kept this way until
the end of the test. The test rig was placed on a controlled temperature environment, being kept between 18 and 20 °C.

Figure 1 - Experimental setup for the open loop PHP.

Figure 2 presents a comparison between the experimental tests with the PHP operating with pure deionized water and the nanofluid at horizontal orientation. It is important to observe that with the pure water, the PHP did not present the great pulsations that are characteristics of this device, even though it was fully operational. However, with the addition of solid copper nanoparticles, the pulsations started appearing at 40 W and became more evident at 50 W with amplitudes around 5 °C on the evaporation section, 22 °C on the adiabatic section and 23 °C on the condensation section. It could also be observed that the mean evaporation section temperature when using the nanofluid stabilized around 90 °C for 50 W while for the same operation condition when using pure water was around 118 °C. In general, the evaporation temperatures when using the nanofluid were lower than when using deionized water.

Figure 2 - Experimental results for the PHP at horizontal orientation: (a) pure water and, (b) water-copper nanofluid.

References


RE-EVALUATION OF THE HONIGMANN-PROCESS: THERMO-CHEMICAL HEAT STORE FOR THE SUPPLY OF ELECTRICITY AND REFRIGERATION

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Abstract

In 1883 the ‘Honigmann-Process’ was invented and filed for patent by Moritz Honigmann (1). In the following two years several so-called ‘fireless locomotives’ were built and successfully used for public transportation in Berlin and Aachen. Obviously, the Honigmann-Engine was not able to compete against the steam and electric locomotive. Today, especially the perspective of an energy storage which can be recharged from a low temperature heat source seems to be an interesting option against the background of the ongoing discussion about future energy scenarios. This heat accumulator could then easily be used for the supply of either electricity or refrigeration.

Figure 1 shows a Honigmann-Engine with sodium as absorbent and water as working fluid as it was built in the 19th century. The principle of the process is based on the vapour pressure depression of the concentrated solution in comparison to the pure working fluid. Sodium hydroxide solution is filled in a boiler (a) with immersed heat exchanger tubes (c) which are connected to a water tank (b). Heat of the hot solution evaporates water in the exchanger (c), and the water vapour does work in a piston (f). Afterwards the steam is directed into the solution by a pipe (e) where it is absorbed. Heat of absorption is released and subsequently evaporates more water. However, the solution is continuously diluted and the vapour pressure depression decreases, so that less work can be produced. The solution then needs to be regenerated. This can be done by input of heat at a regeneration temperature of 80 to 90°C and condensing the steam by use of cooling water.

This concept can also be utilised to operate a stationary system using a steam turbine instead of the piston. In Figure 2a the process of desorption with heat input is shown in a simplified P-T-diagram with depicted isosteres (Dühring or Van‘t Hoff). The heat which is put into the system during desorption is the heat which can be stored. At another time this energy can be used either according to the Honigmann principle for the production of electricity (Figure 3a) or to produce cold (Figure 3b) in the way of classical batch absorption cooling. The evaporator has to be at a different pressure, but the solution can be on the same level. Thus, there exists the possibility for a twofold usage of the stored heat. Moreover, the regeneration can be accomplished with work also by reversing the Honigmann process (Figure 2b).
Although the process has been used originally with water and sodium hydroxide solution (NaOH) as working pair, it can be realised with all known and unknown absorbents and adsorbents. To name a few, there are water and sulphuric acid (H₂SO₄), lithium chloride (LiCl), lithium bromide (LiBr), calcium chloride (CaCl₂), zeolith, or silicagel, or ammonia (NH₃) with water, ammoniates, activated charcoal, and so on.

Isshiki et al. (2) have built several vehicles with water/LiCl working with the Honigmann process. Apart from this activity no other recent work on the process is known to the authors.

We present the results of basic investigations concerning the energy balances of the process. The results of first approximations for the mechanical energy density and efficiency are values of around $\rho_{\text{mech}} = 1 - 18 \text{ Wh/kg}$ and $\eta_{\text{mech}} = 0.04 - 0.10$ for lithium bromide and water. Compared to batteries this is not at the high end. However, for transformation of low temperature heat into mechanical energy these values are highly promising.

Finally, we will present the outline of an experimental plant which will be erected at TU Berlin.

References

EXPERIMENTAL STUDY OF TRANSPORTATION OF LOW-GRADE HEAT ENERGY OVER LONG DISTANCE BY SINGLE EFFECT AMMONIA-WATER ABSORPTION CYCLE

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Abstract

Energy conservation and environment protection issues have gained more and more attention all over the world, and the research to make more efficient use of renewable energy such as solar energy and waste heat from industry plants and power stations is getting a fast growth. However, the waste heat sites and the demand sites are usually located apart from each other. The absence of efficient ways to overcome the long distance transportation problem leads to great difficulties to use this great deal of waste heat. The ammonia-water absorption technology is a potential solution.

Description of the cycle

Ammonia-water absorption process converts the thermal energy into the concentration difference of liquid solutions which can be transported at ambient temperature, so no insulation is required for pipelines, consequently the distance of transportation is theoretically unlimited. Figure 1 shows the schematic diagram of the ammonia-water absorption long distance heat energy transportation system. The waste heat is injected into the generator and the rich solution is separated into the ammonia vapor and the weak solution, and then the ammonia vapor is condensed in the condenser. The weak solution and the ammonia liquid are transported from the source site to the user site. At the user site the liquid ammonia evaporates in the evaporator to produce cold, or the weak solution absorbs the ammonia vapor in the absorber to produce heat. The rich solution formed in the absorber is then transported back to the generator. Thus there are three liquids which are transported, the ammonia liquid, the weak solution and the rich solution. The heat energy is stored into the concentration difference of the solutions, and transported at ambient temperature.

![Diagram of the ammonia-water absorption system](image)

Figure 1. Principle of transportation of low-grade heat energy over long distance by ammonia-water absorption

Description of the prototype

In this paper, an experimental prototype of 2kW heating capacity is built to investigate the feasibility of the process. Figure 2 shows the outline of the prototype. The prototype is composed of ten main components, i.e. the generator, the rectifier, the partial condenser, the main condenser, the evaporator, the absorber, the solution pump, the solution heat exchangers at the source site and the other at the user site, and the long-distance transportation coils (length of 30 m with the inner diameters of 5mm as a demonstration).
The generator is driven by an electric heater. Heating is transmitted to the inner side of the generator by radiation, and then the solution is heated thereby boiled. The rectifier is a packed column, in which the wire-mesh packing is used. The partial condenser, the main condenser, the evaporator and the absorber are all designed as falling-film heat exchangers. The solution heat exchangers are designed as modified countercurrent coil exchangers. The solution pump is a flow control membrane pump, whose flow rate can be adjusted from 10% to 100% within the operating range. The three pipelines of 30m for long-distance transportation are designed as three coils. Their inner diameters are 5mm, and the velocity of the liquids varies from 1 to 3m/s according to different operating conditions. The transporting pumps of the ammonia liquid and the weak solution is not included, because the pressure difference between the generator and the absorber is great enough to drive the two liquids in this prototype.

**Summary Conclusions**

The experiment results show that the ammonia-water absorption system is suitable to transport heat or cold over long distance. In this way, the power consumption of big cities for air-condition in summer, heating in winter and hot water supply all through the year can be greatly reduced if the waste heat can be transported from the waste heat sites, for example, nuclear power stations and big industry zones, which are located several tens of kilometers away from the user sites.

**Reference**


ABSORBER COATINGS FOR PARABOLIC TROUGH COLLECTORS
A REVIEW

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Abstract

Parabolic trough collectors are a special kind of concentrators that can be used for low to medium temperature applications, like water heating to steam for power generation. The greatest advantage of this and other type of concentrators is that the absorber/receiver area is much smaller than the collector aperture. This creates opportunities to develop and apply novel, high-technology coatings in order to improve the optical characteristics. The most important of such characteristics are the absorptance and the emittance. The former must be as close to unity and the latter as close to zero, as possible. In this way the overall performance of the system can be maximized, thus expanding the areas of applicability. Current materials used for selective absorbers are either very expensive or they loose their properties when stagnation conditions exists. These conditions are obtained when the flow in the collector is interrupted and the collector remains at the focus position. By doing so, very high temperatures on the order of 600-800°C can be developed depending on the magnitude of the available radiation and the collector geometric concentration ratio.

Generally, to maximize the energy collection, the absorber of a collector should have a coating that has high absorptance for solar radiation (short wavelength) and a low emittance for re-radiation (long wavelength). Such a surface is referred as selective surface. The absorptance of the collector surface for shortwave solar radiation depends on the nature and colour of the coating and on the incident angle.

By suitable electrolytic or chemical treatments, surfaces can be produced with high values of solar radiation absorptance ($\alpha$) and low values of longwave emittance ($\varepsilon$). The parameter representing the quality of coating is called selectivity defined as the absorptance/emittance ratio ($\alpha/\varepsilon$). Essentially, typical selective surfaces consist of a thin upper layer, which is highly absorbent to shortwave solar radiation but relatively transparent to longwave thermal radiation, deposited on a surface that has a high reflectance and a low emittance for longwave radiation. Selective surfaces are particularly important when the collector surface temperature is much higher than the ambient air temperature. The cheapest absorber coating is matt black paint however, this is not selective and the performance of a collector produced in this way is low especially for operating temperatures more than 40°C above ambient.

An energy efficient solar collector should absorb incident solar radiation, convert it to thermal energy and deliver the thermal energy to a heat transfer medium with minimum losses at each step. It is possible to use several different design principles and physical mechanisms in order to create a selective solar absorbing surface. Selective absorber surface coatings generally fall into six distinct categories (Kennedy, 2002) as shown schematically in Figure 1:

a) Intrinsic or mass absorbers. These absorbers use a material having intrinsic properties that result in the desired spectral selectivity.

b) Semiconductor-metal tandems. Semiconductor-metal tandems absorb short wavelength radiation because of the semiconductor bandgap and have low thermal emittance as a result of the metal layer.

c) Multilayer absorbers, which use multiple reflections between layers to absorb light.

d) Metal-dielectric composite coatings. Metal-dielectric composites—called cermets—consist of fine metal particles in a dielectric or ceramic host material.

e) Surface texturing. Textured surfaces can produce high solar absorptance by multiple reflections among needle-like, dendritic, or porous microstructure.

f) Selectively solar-transmitting coatings on a blackbody-like absorber. These are typically used in low-temperature applications.
These constructions are explained in some detail in this paper.

Figure 1 - Schematic diagrams of the various types of selective coatings and surface treatments

Today, commercial solar absorbers are made by electroplating, anodization, evaporation, sputtering and by applying solar selective paint coatings. From the many types of selective coatings developed the most widely used is the black chrome. Much of the progress during recent years has been based on the implementation of vacuum techniques for the production of fin type absorbers used in flat plate collectors which are suitable for low temperature applications. The chemical and electrochemical processes used for their commercialization were readily taken over from the metal finishing industry. The requirements of solar absorbers used in high temperature applications however, namely extremely low thermal emittance and high temperature stability, were difficult to fulfill with conventional wet processes. Therefore, large-scale sputter deposition was developed in the late 70’s. The vacuum techniques are nowadays mature, characterized by low cost and have the advantage of being less environmentally polluting than the wet processes (Wackelgard, et al., 2001).

Due to lack of suitable selective coatings parabolic trough collectors usually operate at temperatures of about 400°C. As part of a research project undertaken by the authors, advanced absorber coatings will be developed. These should ideally posses the required properties of high absorptance and low emittance at a temperature of about 500°C. Possible candidate materials to be used for this purpose are Diamond Like Carbon (DLC) materials, which are known for their good optical properties, characterized by strong absorption in the UVA-UVB spectral region (Kassavetis, et al., 2007). An attractive feature of DLC, either in its hydrogenated or in the non-hydrogenated form [known as tetrahedral amorphous carbon (ta-C) with a high fraction of sp³ hybrid bonding] is one that can tailor its optoelectronic properties by varying the sp²/sp³ fraction and the hydrogen content. Therefore, one can vary the optical gap of the material (Mathioudakis et al., 2007) to tune it with the desired photon frequency for optimum absorption. DLC is also known for its high temperature stability (Kelires, 1992; 1994), which in combination with its good absorbing properties, makes it ideal for the proposed type of coating.

References

PERFORMANCE OF A TRIPLE-PRESSURE LEVEL ABSORPTION/COMPRESSION CYCLE

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Abstract

The performance of the triple-pressure level (TPL) single stage absorption cycle operated with various organic refrigerants and absorbents showed many advantages over the common double pressure level (DPL) absorption cycle [1-3]. In order to enhance these advantages (increased COP and decreased generator temperature); the jet ejector was replaced by a mechanical compressor and a mixing device (see Fig. 1). In the modified triple-pressure level absorption cycle, the compressor was inserted in the super heated refrigerant line between the evaporator and the absorber. The influence of the compressor on the performance of the TPL absorption cycle with the working fluid pentafluoroethane (R125) and N,N'-dimethylethylurea (DMEU) was predicted by a computerized simulation program. The performances of the TPL absorption cycle operated with mechanical compressor or jet ejector and the DPL absorption cycle were compared. Based on the analysis the following advantages were achieved: a significant reduction of the generator temperature (i.e., ability to use low grade heat source such as solar energy), increasing of the coefficient of performance (COP), reduction in the circulation ratio ($f$) and the solution heat exchanger (reduction of the actual unit size). The disadvantage of inserting the compressor is increased electricity consumption.

References


Cycle TPL-Comp

Figure 1: Schematic illustration of an enhanced triple-pressure-level (TPL) single-stage absorption cycle.
EFFECT OF FACTORS ON VARIATION OF DESORPTION PRESSURE IN AN ADSORPTION HEAT PUMP

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Abstract

A cycle of adsorption heat pump consists of two isosteric and two isobaric processes. The pressure of adsorbent bed should be constant during the isobaric adsorption and desorption processes to have invariant evaporation and condensation temperatures. Our experimental results show that it might be difficult to maintain the condenser pressure constant and it may change during the desorption process. In the present study, the effects of insufficient condenser capacity and desorption heating rate on the change of desorption pressure are investigated. The experiments were performed on an intermittent adsorption heat pump in laboratory environment. The results of two cycles, one belongs to the adsorption heat pump with a condenser having 0.038 m² heat transfer area and anther is relevant to the same adsorption heat pump with the condenser of 0.226 m² heat transfer area, are presented. In order to show the effects of desorption heating rate on the pressure of adsorbent bed, the results of two cycles with different desorption heating rates as 0.14 °C/min and 1 °C/min are exhibited. The obtained results show that the insufficient condenser capacity not only increases the desorption pressure but also extends the period of cycle. Silica gel-water was used as adsorbent-adsorbate pair.

The tested adsorption heat pump

The components of the experimented adsorption heat pump and the location of thermocouples and pressure transducers are shown schematically in Figure 1. The equivalent diameter of silica gel granules varies between 3 - 5 mm. The water adsorption capacity of silica gel is declared as 25%. The inlet water temperature to the heat exchangers, placed in the condenser and evaporator, is maintained constant by a constant temperature water bath.

Figure 1: Schematic view of AHP system
The adsorbent bed was heated by the electrical resistances which were surrounded around the outer surface of adsorbent bed. The power of electrical resistances was controlled by a PID temperature controller during the isosteric heating and desorption periods. The adsorbent bed was cooled by a fan throughout the isosteric cooling and isobaric adsorption processes.

Result and discussion

The results of the study are presented via diagrams and necessary discussions are performed. A representative diagram is shown in this abstract. Figure 2 shows the variations of the average temperature and pressure in the adsorbent beds of two cycles whose condenser heat transfer areas are different as 0.038 m² and 0.226 m² during the desorption processes. The distribution of outer surface temperature for both cycles is almost identical. For the cycle with 0.038 m² condenser heat transfer area, the adsorbent pressure gradually rises from 7 to 26 kPa and then gradually falls to 13 kPa. The bed pressure of the same cycle with 0.226 m² condenser heat transfer area increases from 7 to 20 kPa and remains constant during the desorption process. For the cycle with the larger condenser heat transfer area, the adsorbate desorbed from the bed is condensed faster and therefore the pressure of the bed does not increase. The period of desorption process for the cycle with 0.226 m² condenser heat transfer area is around 2400 minutes which is shorter than that of the cycle with 0.038 m² (2700 minutes). The use of condenser with insufficient capacity increases the period of cycle, hence the SCP and SHP values become smaller.

Figure 2: The variations of the bed surface temperature and the average pressure and temperature of the adsorbent bed for two cycles having different condenser heat transfer areas.
FUTURE RENEWABLE ENERGY CARRIER: HYDROGEN OR CRYOGEN?
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Abstract

Fossil fuels that meet most of the world’s energy demand today are depleting at an increasingly rapid rate. The use of such fuels has been widely recognised to cause serious environmental issues, including greenhouse effect, ozone layer depletion and acid rains (Veziroglu and Sahin, 2008). It is generally agreed among engineers and scientists that one of the best solutions to the global problems is replace the existing fossil fuels with renewable ones. However, a number of technical barriers have to be overcome before such a solution can be implemented. The first issue is the low energy density of renewable sources, which implies the need for a bulky energy system and hence a high capital cost. Second, renewable energy sources are intermittent, which is creating significant problem in terms of applications. To overcome the above barriers, energy storage is regarded as a solution. As a consequence, various technologies have been developed over the past century; see Table 1 (Chen et al., 2008).

Table 1 Methods of Energy Storage

<table>
<thead>
<tr>
<th>Energy Forms</th>
<th>Mechanical Energy</th>
<th>Thermal Energy</th>
<th>Electrical Energy</th>
<th>Chemical Energy</th>
<th>Nuclear Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Storage</td>
<td>Pumped hydroelectric storage</td>
<td>Flywheels</td>
<td>Compressed Air Energy Storage</td>
<td>Concrete Storage</td>
<td>Phase Change Materials; Cryogenic Energy Storage</td>
</tr>
<tr>
<td>Energy Storage</td>
<td>Battery; Hydrogen based Energy Storage</td>
<td>None</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>

The third issue is related to the mobility of the energy storage system (and the stored energy) because of remote locations of many renewable energy sources e.g. wind and tidal are at remote places and the use of the stored energy by for example cars or other transportation means. As a consequence, the use of an energy carrier, that can not only store the renewable energy, but also transport and distribute the stored energy, becomes part of a key to the solution. This paper aims to assess and compare two most promising renewable energy carriers, hydrogen and air. The former is a form of chemical energy carrier whereas the latter is a physical energy carrier. The assessment and comparison are carried out in terms of the overall efficiency, including production, transportation and application. The environmental impact, waste heat recycling and safety issues will also be considered. It is found that the physical energy carrier may be a better alternate to the use of the chemical energy carrier under some circumstances. This is particularly so when there are waste heat sources, which could boost substantially the efficiency of power generation. As the last part of the paper, prospective applications of the physical carrier will be discussed.

References
AN INNOVATIVE PROTOTYPE OF ADSORPTION CHILLER FOR MOBILE AIR CONDITIONING

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Abstract

Recent developments in adsorption cooling systems demonstrated that such devices can efficiently replace conventional vapour compression chillers and heat pumps. Their applicability is going to become attractive when waste heat is available, such as in trigeneration installation and mobile air conditioning.

An innovative prototype of adsorption chiller for mobile air conditioning has been designed and built at CNR-ITAE. The adsorption machine uses the available heat from the engine cooling circuit and is designed to deliver about 2.5 kW cooling power for the cabin air conditioning. This paper will be mainly focused on the description of the prototype and its experimental performance. The main component of the prototype is a novel double-bed adsorber, which was realized embedding about 1.9 + 1.9 kg of zeolite FAMZ02, produced by Mitsubishi Chemical, into lightweight finned flat-tube heat exchangers. The vacuum chambers were designed to exactly fit the adsorbent beds geometry and realized in aluminium to reduce the weight of the device. Compact evaporator and condenser were specifically realized using special finned-tubes heat exchangers which offer very high exchange surface. The connections between adsorbent beds and condenser/evaporator are regulated by means of electro-pneumatic vacuum valves. A set of electric valves is used to control the circuits for the external heat transfer fluids. A proper data acquisition and management system was realized allowing the automatic operation of the system. Fig. 1 shows the realized prototype, which possesses adequate lightness (60 kg) and compactness (160 l) for air conditioning of the cabin of trucks.

Figure 1: The CNR-ITAE adsorption air conditioner for automotive application

The performance evaluation was carried out by means of a testing station installed at the CNR-ITAE laboratory allowing to test the adsorption prototype under the EU car air conditioning testing conditions. Fig. 2 shows the typical cooling COP and average cooling power (ACP) obtained as function of the outlet temperature of the evaporator. Results obtained demonstrated that the adsorption cooling system is able to deliver adequate cooling power (ACP 1-2.3 kW) with the required efficiency (COP 0.25-0.45). Thus a Specific Cooling Power as high as 300-600 W/kg of adsorbent was obtained
Figure 2: Performance of the CNR-ITAE adsorption air conditioner vs. outlet temperature of evaporation

Acknowledgements
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Abstract

Up to now, technical and economical reasons are in favour of large concentrated solar thermal power plants starting from a rated electrical power of 10 MWel and more. Projects with smaller size power however could have advantages because of flexibility and financing options. Also, the potential benefits of combined generation of heat, cooling and power could be utilized by smaller power stations and if load demands match. Therefore, the German Federal Ministry for the Environment, Nature Conservation and Nuclear Safety (BMU) has financed a study exploring the factors and technologies, which could pave the way to a economical development also for Medium and Small Size Concentrated Solar thermal Power (MSS-CSP) plants from about 10 kWel up to 10 MWel.

Different cycle layouts for co/polygeneration where simulated in TRNSYS. The yearly energy production was calculated on the basis of hourly weather data and load curves for heat, cold and electricity demand. Based on cost assumptions and local feed-in-tariffs, the possible financial savings in comparison with conventional systems were calculated. Cases with high rentability were identified.

Description of the simulated cycle layouts

For solar co/polygeneration different cycle layouts are feasible. Depending on the temperature levels needed, different positions of the heat exchangers in the cycle can be studied with the simulation model. Figure 1 shows a schematic layout of one possible system.
In Figure 1 the Heat Transfer Fluid (HTF) coming from the collector with 300°C is flowing through the heat exchangers of an Organic Rankine Cycle (ORC). Additionally, process heat on a rather high temperature level of 200°C at 12 bar is available. The heat of condensation in the ORC cycle can be used in a thermally driven chiller operating at 85°C.

Other cycle layouts have been investigated and will be reported in the paper.

Procedure

The thermodynamic calculations were run on the basis of hourly values with solar irradiation data and weather conditions for different locations in Europe and Africa. In comparison with a conventional system, the fuel savings for one year where calculated. Based on these energy savings the cash flow respectively the savings for every year of the whole life-span can be calculated under certain assumptions on financing interest rates and fuel price. The possible savings are calculated by calculating the net present value, i.e. summing up the discounted cash flows for each year. Parameters like the solar field size or the storage size are optimized to maximize the savings. If no grid is available, the system is compared with diesel generators which are state of the art in the examined power range.

Findings and conclusions

The simulation results show that solar co/polygeneration of power, heat and/or cooling can be highly profitable if the right cycle design is chosen. The main influencing factors on the technical design are boundary conditions at the given location (i.e. climatic data, solar resource, availability of water, demand and demand profile etc) and financial data. Significant savings compared to fossile fuelled solutions can be achieved by solar thermal co/polygeneration. Even higher savings were found for off-grid applications.
EXPERIMENTAL INVESTIGATION ON THE EFFECT OF THE SWITCHING FREQUENCY ON THE PERFORMANCE OF A THERMAL WAVE ADSORPTION HEAT PUMP

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Abstract

In this work, a two modular zeolite water adsorption heat pump (AHP) based on the thermal wave concept with a direct heat recovery between evaporator and condenser heat exchangers has been introduced and experimentally investigated. Each module composes two heat exchangers contained in a hermetic stainless steel vessel. The first heat exchanger is the adsorber/desorber located on the top of the vessel, while the second is the evaporator/condenser heat exchanger located on the bottom of the vessel.

The effect of the non-dimensional switching frequency defined as the ratio between the heat capacity of the adsorbent bed and the heat capacity rate of the adsorber, or primary cycle, heat transfer fluid multiplied by the half cycle time on the coefficient of performance and the mean heating power of the AHP has been experimentally investigated under a typical working condition of AHPs. The results showed that the switching frequency and the primary cycle fluid flow rate have strong influences on both COP and the mean heating power. It has been also found that there is an optimum switching frequency corresponding to each flow rate, at which the COP attains its maximum value. The obtained optimum switching frequencies increase slightly from 0.26 at a primary cycle fluid flow rate of 0.6 l/min to 0.32 at 1 l/min, which has been attributed to the enhancement of the adsorption bed heat transfer characteristics.

The optimum COP increases with decreasing the primary cycle fluid flow rate, within the studied range of flow rates. A simple control strategy for this kind of adsorption heat pumps is described based on individual correlations for the optimum switching frequency, the optimum COP and the obtainable mean heating power as functions of the primary cycle fluid flow rate.

Keywords: Adsorption heat pump; Zeolite-water; Switching frequency; Thermal wave

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Abstract

Polygeneration describes an integrated process which has three or more outputs that include energy outputs, produced from one or more natural resources. The most popular and wide-spread form of polygeneration is trigeneration, a simultaneous production of heat, cold and power. Nevertheless, other polygeneration systems offer a vast potential as well. Popularising polygeneration would allow applying energy systems fitting precisely the needs of investors. This would mean rising efficiency, reducing costs and obtaining environmental benefits. Polygeneration could therefore be helpful in meeting the requirements of sustainable development.

This article attempts to form a basic classification of polygeneration systems. Along with major technical, economical and environmental features, the advantages and disadvantages of each system are highlighted and some case studies presented. Existing polygeneration systems are various and can be applicable in many sectors of economy or industry. This article aims to demonstrate the technical feasibility of different polygeneration systems and to show that these systems could be beneficial.

The polygeneration systems analysed within this paper can be divided into following major categories:

- trigeneration systems – energy systems generating heat, power and cold in one process,
- gasification systems – production of producer gas (or syngas) in the gasification process to be used for energy or synthesis purposes, with the residual ash to be used in construction or agriculture,
- biogas generation – generation of biogas from organic waste to be combusted for energy purposes (heat and power), with the residual ash to be used in construction or agriculture,
- water desalination and purification – CHP engines can be integrated to deliver power and heat or some part of it to a water desalination or purification station; in result in one system can provide quality water, heat and power,
- CO₂ harvesting – an idea of using a CHP to deliver heat and power (or some part of it) to a greenhouse farm, while cleaned exhaust gases (mainly CO₂) are blown into greenhouse to stimulate the plant growth,
- bioethanol industry – production of ethanol and residues from organic matter; residues are either used to produce heat and power or as feedstock, while ethanol is used as a quality fuel.

The systems basic characteristics (input, output) are given in the table below.

<table>
<thead>
<tr>
<th>Process</th>
<th>Trigeneration</th>
<th>Gasification</th>
<th>Biogas production</th>
<th>Water desalination</th>
<th>CO₂ harvesting</th>
<th>Bioethanol production</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output 1</td>
<td>Heat</td>
<td>Heat</td>
<td>Fertiliser</td>
<td>Heat</td>
<td>Heat</td>
<td>Heat</td>
</tr>
<tr>
<td></td>
<td>Clod</td>
<td>Cold</td>
<td>Biogas</td>
<td>Power</td>
<td>Power</td>
<td>Bioethanol</td>
</tr>
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<td></td>
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<td>• heat</td>
<td></td>
<td></td>
<td>• heat</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>• power</td>
<td></td>
<td></td>
<td>• power</td>
</tr>
<tr>
<td>Output 2</td>
<td>Power</td>
<td>Industrial</td>
<td>Water</td>
<td>Crops</td>
<td></td>
<td>Feedstock</td>
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<tr>
<td></td>
<td></td>
<td>raw material</td>
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</table>

Table 1: Basic classification of polygeneration concepts
Polygeneration offers a very wide range of applications therefore only the most popular polygeneration solutions are covered in this paper. It is worth just mentioning some other interesting polygeneration systems not detailed here e.g.: using waste wood (e.g. lumber mill by-products as bark, shavings, sawdust) for heat (drying) and power generation and processing sawdust to wood pellets or using solar energy to generate electricity with PV system acquiring at the same time the heat from PV module cooling to generate cold in thermally driven chiller.
FIRST RESULTS OF A MICRO-CHCP SYSTEM WITH TWO ADSORPTION CHILLERS

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Abstract

In this study we present preliminary results of a CHCP installation with two adsorption chillers. The system consists of a 19kW therm and 8 kW el CHP unit. The two adsorption chillers with a nominal cooling capacity of 5.5kW each are connected in series at hot water and chilled water loop, while cooling water is in parallel. A description of the concept will be given in this article. Moreover, a detailed analysis of the hydraulic network is presented. It shows the energy demand of the main hydraulic loops such as hot water loop, cooling water loop, chilled water loop, CHP cooling loop and the chilled water distribution network. Also a grouping according to components installed in the hydraulic system such as pipes, fittings, heat exchangers and sensors will be presented. It turns out that the cooling water loop causes very high electricity demand because of its high flow rate and high pressure drop. Moreover, a sensitivity analysis of some parameters regarding the thermally driven chillers (TDC) has been performed. The analysis indicates that good control algorithms have to be found for the cooling water loop in order to minimize electricity demand.

Description of the CHCP-concept

Figure 1 shows a schematic layout of the CHCP system installed in an office building at Fraunhofer ISE in Freiburg, Germany. It consists of the CHP unit, two TDCs a dry heat rejection unit and three buffer storages. The big buffer storage (1000L) between the CHP and the TDC helps to avoid frequent stops for the CHP. The small stratification storage (600L) in the return line protects the CHP from the fast temperature variation of the TDCs, which comes intrinsic with solid sorption chillers. The inlet temperature for the CHP is conditioned by taking water from top (hot side) and bottom (cold side) of the stratified storage. The third storage (1000L) is in the chilled water side and allows to store cooling capacity. The TDCs are connected in series in the hot water and chilled water loop. Series connection in the hot water loop was necessary to match the temperature spread and volume flow of the CHP and TDC (CHP at nominal condition: \(\Delta T = 20K\), \(\dot{V} = 800l/h\), TDC at nominal condition \(\Delta T = 8K\), \(\dot{V} = 1000l/h\)).

On the demand side an open plan office (150 m²) and several single offices (in total 98m²) are cooled. While the large office is equipped with 7 fan-coils distributed strategically in the room, the single offices are equipped with chilled ceilings with embedded micro-encapsulated phase changing material (PCM) to store cooling energy. The combination of CHCP with PCM looks promising since the system can be operated night and day. While the PCM is regenerated during night time the open plan office is cooled during day time. The high heat capacity of the PCM chilled ceilings helps to keep comfortable temperatures in the single offices during a long period of the day without the need to be operated with chilled water. With this concept, a cooling demand almost twice as large as the cooling capacity of the chillers can be satisfied. Additionally, the operation of the chillers during lower night time temperature conditions favours their efficiency. Moreover a heat exchanger between the CHCP system and the already existing heating network of the building installation has been installed. In heating season heat can be transferred to the heating network and successfully demonstrates the integration into conventional system.
Figure 1: Hydraulic scheme of the CHCP-system.
EXPERIMENTAL INVESTIGATION
OF A TWO-PHASE THERMOSYPHON HEAT EXCHANGER

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Introduction

Two-phase thermosyphon (TPhT) operates in such a way, that when heat is supplied to the heating zone the working fluid starts boiling and produced vapour moves to condensing zone where it is condensed giving up the heat of phase transition to the working fluid. And gravity force causes the condensate to return to the evaporator and the processes of evaporation and condensation occur continuously. The characteristic feature of TPhT is that operates as a thermal diode, it means that the heat can be transported only in one direction – from the evaporator to the condenser. TPhT can be divided into two main groups: termosyphon tube with countercurrent flow of the liquid and the vapour and two-phase loop where the evaporator is connected to the condenser by a riser and a downcomer. Two-phase thermosyphon heat exchanger is a recuperator with an intermediate working fluid. This type of heat exchangers is used in a variety of heat transfer applications, but mechanisms governing the heat transfer process in such heat exchangers with shell-side boiling are far from understanding.

Prototype two-phase thermosyphon heat exchanger

Prototype two-phase thermosyphon (TPhTHEx) tested is a shell and tube horizontal heat exchanger made of stainless steel 1.4404 as a welded construction. The shell consists of two cylindrical vessels of 159 mm in diameter and 1 m long connected by a riser and a downcomer. Evaporator is designed as a tube bundle consisted of 19 smooth, corrugated or porous coated tubes (OD 10 mm) with triangular arrangement, with a pitch equal to 2.0d or 1.7d. Condenser is designed as a tube bundle consisted of 31 smooth tubes (OD 10 mm) with triangular arrangement and pitch equal to 1.8. As intermediate working fluids water, methanol and refrigerant R-141b are utilized. Experimental investigations were carried out making use of six prototype heat exchangers specially designed for this purpose.

Experimental investigation

An experimental investigation has been conducted to examine the influence of tube bundle geometry and initial liquid level above the top tube row on overall performance of TPhTHEx for three kinds of tubes used to built evaporator tube bundle, i.e. smooth, corrugated and porous coated, and three working fluids: water, methanol, and R141b.

The test stand consists of three main systems: prototype TPhTHEx, heating water loop and cooling water loop. The test facility is capable of determining of an overall heat transfer coefficient of TPhTHEx.

Experimental procedure

During tests the absolute pressure inside a shell varied from 5 up to 20 kPa for water and methanol and from 90 up to 180 kPa for refrigerant R-141b. The temperature of an intermediate boiling fluid
changed from 48°C to 62°C for water, from 24°C to 32°C for methanol and from 29°C to 50°C for R-141b. Before the tankage of TPhTHEx with an intermediate working fluid a vacuum absolute pressure of 5 kPa was created inside a shell.

Heating as well as cooling water mass flow rates ranged from 0.3 up to 3.5 kg/s. The monitoring of the temperature and pressure reading is facilitated by use of a PC-aided data acquisition system. All data readings have been performed during steady states.

Results

As an example, influence of the initial liquid level above the top tube row on evaporator performance is shown in Fig. 1. The evaporator tube bundle was built in this case from corrugated tubes with pitch equal to 1.7d and methanol served as intermediate working fluid. Dramatic decrease of heat flux density was recorded for the smallest liquid level tested, i.e. 5 mm, because the top tube row of the bundle was not flooded during evaporator operation. No influence of initial liquid level between 15 mm and 25 mm on evaporator performance has been observed. Fig. 2 displays influence of a tube type on evaporator performance with R-141b as working fluid, tube pitch equal to 1.7d and operating pressure ca. 100 kPa. The performance level of the porous coated tube bundle was about two times that of smooth tube bundle. For corrugated tubes heat transfer augmentation was about 30%. The same relationship was recorded for water and methanol.

Conclusions

Stable performance of the prototype two-phase thermosyphon heat exchanger was recorded for wide range of thermal duty.

Application of enhanced tubes, particularly porous coated, results in better heat transfer characteristics of two-phase thermosyphon heat exchanger.

For low wall superheat the best evaporator performance while boiling on porous coated tube bundle was achieved for refrigerant R-141b, but for higher wall superheats better heat transfer coefficients were obtained for water what results from different boiling regimes observed.

For a given wall superheat higher heat flux density transferred from evaporator was observed for smaller tube pitch examined, i.e. 1.7d, independently on kind of tube surface.

No influence of initial liquid level between 15 mm and 25 mm on evaporator performance has been observed.

A modified Péclet equation was proposed for calculation of heat flux transferred in TPhTHEx.

References

PERFORMANCE ADVANTAGES AND OPERATIONAL CHALLENGES IN FREE-PISTON INTERNAL COMBUSTION ENGINES

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Abstract

This paper studies the feasibility and performance of a free-piston engine generator. The free-piston engine is an old concept which has gained attention in recent years due to the development of modern, microprocessor-based control technology. Such engines have the potential advantages of low frictional losses, compactness, and operational flexibility, however require active control of piston motion. Based on extensive previous research and reports from other authors, this paper analyses the potential advantages of and challenges associated with this type of engine.

The operational characteristics of free-piston engines differ significantly from those of conventional engines, and this influences cycle performance. Advanced modelling and simulation tools are used to study the thermodynamic performance of the free-piston engine. Potential advantages over conventional engines in the areas of fuel efficiency and exhaust gas emissions formation are investigated, including detailed studies of engine in-cylinder processes such as heat transfer losses and combustion chamber gas motion.

Due to the absence of the crankshaft, the free-piston engine has the valuable feature of variable compression ratio. However, due to this, active control of piston motion is required in order to secure stable engine operation and avoid engine damage resulting from, for example, excessive in-cylinder gas pressures. Engine control issues have been studied using full-cycle dynamic models, and a number of control approaches have been tested. The influence of rapid load variations on the engine is studied and the capability of different control system designs to maintain stable engine operation is tested.

Description of the engine

The free-piston engine is a linear combustion engine, consisting of two opposed combustion chambers and a linear load device. An illustration of a free-piston engine with a linear electric generator is shown in Figure 1.

![Figure 1: Dual piston free-piston engine.](image)

The engine is in practice restricted to the two stroke operating principle, as a power stroke is required in every cycle. It can use loop-type scavenging, with intake and exhaust ports in the cylinder liner, or uniflow scavenging using exhaust poppet valves. As is well known from the literature, two stroke engines suffer from gas exchange short-circuiting, i.e. the flow of inlet charge into the exhaust system during scavenging. To ensure a high fuel efficiency and low exhaust gas emissions, direct injection operation is therefore most suitable for the free-piston engine.
Free-piston engine advantages

Potential advantages of the free-piston engine discussed include:

- reduced combustion chamber heat transfer losses due to a faster power stroke expansion;
- reduced frictional losses and a high power to weight ratio due to a simple design with few moving parts;
- increased engine operational optimisation possibilities, and suitability for multiple fuel operation;
- enhanced in-cylinder gas motion due to higher piston velocities around top dead centre, leading to improved combustion and reduced emissions formation through enhanced fuel-air mixing and lower peak gas temperatures;
- suitability for homogeneous charge compression ignition (HCCI) operation due to lower ignition timing control requirements.

The control challenge

In order to maintain stable engine operation and realise a high level of engine operational optimisation, accurate control of the piston motion is required. The functions of the crank- and camshafts in conventional engines, namely piston motion control, valve actuation, valve and fuel injection timing, etc., must be replicated by an engine control system. This currently represents the main challenge for free-piston engine developers, however with the use of modern control engineering technology, the control problem can be resolved.

Due to the low inertia of the system, the effect of load changes on engine operation will be different from that in conventional engines. This will be discussed, and simulation results investigating different control approaches will be presented. Other challenges, such as cycle-to-cycle variations, will be presented and potential solutions studied.

Bibliography


A NUMERICAL STUDY ON HEAT AND MASS TRANSFER IN ADSORBENT BED OF AN ADSORPTION HEAT PUMP; EFFECT OF BED THICKNESS

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Abstract

Despite of many advantages of adsorption heat pump, they have some drawbacks which retard their widespread commercialization. One of the main drawbacks of the adsorption heat pump is the poor thermal conductivity of the adsorbents. Moreover in the granular adsorbent bed, the voids between the granules lead a discontinuity between the adsorbent particles and as a result the thermal conductivity of the adsorbent bed is considerably reduced. The low thermal conductivity of the bed slows down the speed of heat and mass transfer in the bed and increases the period of cycle. In order to improve heat and mass transfer in the adsorbent bed, the mechanism of heat and mass transport in the bed should be well understood. The solution of the heat and mass transfer equations for an adsorbent bed provides valuable information for better understanding of heat and mass transport and also for optimization of the adsorbent bed. In the present study, heat and mass transfer in a granular adsorbent bed is studied and effects of bed thickness are investigated. The heat and mass transfer equations for the adsorbent bed and the mass balance equation for the adsorbent granules are solved numerically. The results are obtained for six different thicknesses of the bed from 0.03 to 0.18 m. The adsorption and desorption pressures are considered as 2 and 20 kPa. The study is performed for silica gel-water pair and the porosity is 0.1

The considered adsorbent bed

Figure 1 shows the schematic view of the analyzed annular adsorbent bed filled with the adsorbent granules. The adsorbent granules are silica gel grains and adsorptive is water vapor. The adsorbent bed has a cylindrical annular shape. The adsorptive can easily flow in the center of annulus and enter uniformly to the portion filled with granules. The adsorptive flows from R = Rᵢ surface towards the outer surface, R = Rₒ. The upper and bottom surfaces of the adsorbent bed are insulated and therefore the transfers of heat and mass occur only in radial direction. The thermal resistance of the metal casing is neglected. The inner radius of the annulus is Rᵢ = 0.06 m and the study is performed for six outer radiuses as 0.09, 0.12, 0.15, 0.18, 0.21, 0.24 m. The equivalent radius of the adsorbent granules is rₑ = 0.0016 m.

Figure 1: Schematic view of the adsorbent bed.
Formulation of the problem

Two types of mass transfer occur in an adsorbent bed contains adsorbent grains; mass transfer within the adsorbent granules and mass transfer through the voids between the granules (i.e. intra-particle and inter-particle mass transfers). The mechanisms of heat and mass transfer in a granular adsorbent bed are complicated; hence some assumptions have to be made to pose the governing equations. In this study, the proper assumptions were made and the following governing equations were solved:

\[
\left( \rho C_p \right)_{eff} + \rho_s C_{pw} W \frac{\partial T}{\partial t} = \lambda_{eq} \frac{1}{R} \frac{\partial}{\partial R} \left( R \frac{\partial T}{\partial R} \right) - \frac{1}{R} \frac{\partial}{\partial R} \left( \rho_w C_{pw} R u T \right) + \rho_s \Delta H_{st} \frac{\partial W}{\partial t} \tag{1}
\]

\[
\frac{\partial \rho_w}{\partial t} + \left( \frac{1}{\varepsilon} \right) \rho_s \frac{\partial W}{\partial t} + \frac{1}{\varepsilon R} \frac{\partial}{\partial R} \left( R \rho_w u \right) = 0 \tag{2}
\]

\[
\frac{\partial W}{\partial t} = \frac{15 D_{eff}}{r_p^2} \left( W_{\infty} - W \right) \tag{3}
\]

The equations (1) and (2) are the heat and mass transfer equations for the adsorbent bed. The Eq. (3) is the mass balance equation for the adsorbent granule. The distribution of adsorptive pressure in the bed can be found from the distribution of adsorptive density by using the ideal gas relation. The velocity field can be obtained from the adsorptive density by using Darcy relation.

Result and discussion

Based on the obtained results, the distributions of temperature, pressure, adsorptive density and adsorbate concentration in the adsorbent bed through a cycle are plotted and their variations are discussed. A representative diagram is presented in this abstract. Figure 2 shows the variation of average adsorbate concentration in the bed through the cycle. As is seen, the average adsorbate concentration increases from 0.115 to 0.30 kgW/kgS during the isobaric adsorption process and it remains constant during the isosteric heating and cooling processes. In the isobaric desorption process, the water is removed from adsorbent particles by heating the adsorbent bed and the average adsorbate concentration falls to 0.115 kgW/kgS.

![Figure 2: Variation of adsorbate concentration in the adsorbent bed through the cycle.](image.png)
HEAT2COOL – COOLING OF CHARGED INLET AIR WITH EXHAUST HEAT FOR INTERNAL COMBUSTION ENGINES

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Abstract

A feasibility study comparing different heat-driven cooling cycles and their integration an automotive engine is conducted at the TU Berlin.

Engine-side aspects

About fifty percent of all gasoline engines nowadays produced are equipped with a supercharging device. A supercharger supplies the combustion engine with air at a pressure level higher than ambient increasing the engine’s power and efficiency and lowering emissions. The supercharger can either be driven by a mechanical connection with the engine’s crankshaft reducing engine power or be powered by an exhaust gas turbine (turbocharger). Increasing efficiency demands predict a broad application of superchargers in gasoline engines in the near future. Nowadays common diesel engines are almost exclusively turbocharged, direct injecting engines.

Elevated pressure of the inlet air, however, results in elevated temperature which reduces the effect of the pressure rise; the increase of the power density is proportional to the suction density and not the suction pressure. Especially in gasoline engines, the higher temperature may lead to uncontrolled combustion, i.e. engine knocking. Therefore, the compressed inlet air is being cooled down before the intake manifold in an intercooler which is an air-to-air or air-to-liquid heat exchanger. At most the inlet air can be cooled down to ambient air or cooling water temperature, respectively.

Figure 1: System integration of the cooling device
The application of a cooling device as seen in Figure 1, however, allows the cooling of the inlet air to a temperature even lower than ambient. If the cooling device is driven by waste heat, there is even no shaft power taken from the engine and a considerable increase in efficiency can be expected. The chemical energy of the fuel is more or less in equal measure converted to mechanical energy, thermal energy in the cooling water and thermal energy in the exhaust gases. Therefore, there is a high potential of thermal energy in the exhaust gases to drive the chiller without any negative energetic feedback to the engine.

**Investigated chilling procedures**

Exhaust-gas-driven cooling devices and their potential for inlet air cooling are the focus of the present study. Sorption chillers with liquid sorption media are an established technology in stationary but not yet mobile applications. Depending on the working media, crystallization can be an issue with these well-proven machines. Sorption chillers with solid sorption media are an alternative option, eliminating the risk of crystallization but adding additional bulk and weight. The Vuilleumier process is a gas process not limited by vapor pressure curves or crystallization. In this process, two pistons displace gas between cold and hot heat storages invoking pressure oscillations which lead to heat exchange at different temperature levels. Thermoelectric devices use the Seebeck or Peltier effect, respectively, to provide an electrical current or a temperature difference. By coupling of thermoelectric generators with thermoelectric cooling devices, the exhaust energy can be converted to an electrical current powering the cooling element. This system has the advantage of low weight, high flexibility and reliability while providing only moderate efficiency. In thermoacoustics, temperature differences are induced by pressure waves and vice versa. The coupling of a thermoacoustic generator with a thermoacoustic cooling element can be employed to cool the charged inlet air as in thermoelectrics. Steam jet cooling machines employ a steam jet compressor in order to suck in and compress low-pressure steam by a high-pressure steam jet and thus providing evaporative cooling on the low pressure side.

**Scope of study**

The scope of the study is the evaluation of the identified chilling procedures with regard to their respective cooling potential or efficiency given certain operating conditions from the engine side and their technical feasibility in a mobile application. Therefore, basic thermodynamical models will be elaborated to represent the respective cooling procedures. These models will in a first step include the decisive parameters of the different cooling processes and evaluate their efficiency in a direct coupling with the engine models.

Hypothetically, an engine of 60 kW would provide a thermal power of about 20 kW in the exhaust gases. Using only ten percent of that energy depending on the efficiency of the cooling device would yield 2 kW of refrigerating capacity which would be sufficient to significantly (more than 10 K) cool down the inlet air.

**Acknowledgments**

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EVALUATING NEW LINES OF APPLICATION OF PASSIVE THERMAL CONTROL DEVICES

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Abstract

With the increasing demand on energy all over the world, new investments are needed to explore old and new sources of energy, being ultra-deep water petroleum and natural gas reservoirs, solar and wind energy, fuel cells and gasification technologies. For all those energy sources, equipments need to be designed and built for power generation, which are directly related to internal combustion engines, steam and gas turbines. Even though most of this technology is available, constant development and research is required to improve their performances and such improvements are directly related to a proper thermal control. Considering the current needs for power generation and the seek for more clean and environmentally correct processes, this paper presents some areas where the passive thermal control could contribute to the improvement of power generation machineries. A brief presentation and discussion regarding the potentials on applying passive thermal control devices on several areas is presented. The main objective of this paper is to show that currently known technologies should be used to improve the power generation matrix in many countries, improving their thermal performances by using heat pipes and loop heat pipes technologies, in order to make them less aggressive to the environment without loosing performance.

Introduction

There have been several investigations related to the improvement of power generation machineries such as internal combustion engines and turbines (gas and steam). Since those machineries require the use of a fuel, some options are directly related to natural gas (fossil fuel and thus there is a need for replacement by renewable sources) and Syngas from gasification process. For this last process, many countries such as China, USA, Japan, Brazil, India, among others are investing on the improvement and development of gasification process, basically focusing on mineral coal. In some countries where biomass is abundant, the gasification process has been develop towards the use of agricultural residues for power generation using gasification/pyrolysis processes to produce gas to be burned in turbines and internal combustion engines in a so-called IGCC (Integrated Gasification and Combined Cycle). As an advance on the gasification technology, upon obtaining the Syngas, which is reach in \(H_2+CO\), it is possible to obtain synthetic fuels when applying Fischer-Tropsch technology, where as a result the products are methanol, gasoline, diesel, etc (Higman and van der Burgt, 2003; Basu, 2006). On the side of power generation machinery, i.e. internal combustion engines, gas and steam turbines, the losses caused during the combustion and rotatory movement on the bearings cause a loss on the efficiency towards the transformation of chemical/thermal energy in mechanical energy, which is then transformed in electric energy. Great amount of such losses are responsible for heat generation and special heat exchangers must be designed to overcome the rise on temperatures. This is especially important when closed-circuit oil lubrication is used in gas turbines bearings. However, in some case, temperature gradients are encountered in turbines rotors and casts, as well as in internal combustion engines where the heat management must be properly addressed to avoid failures and even greater losses on the thermal and mechanical efficiencies.

Passive thermal control technology

The passive thermal control technology is largely applied in several areas, but has gained a lot of attention from the aerospace area. In this case, satellites and spacecrafts widely apply this passive thermal control technology, where a device is designed to manage the changes on the amount of
heat generated by an electronic component or propulsion system and dissipate the heat to the cold space. Sometimes, the high temperature heat source is located far from the low temperature heat sink and a proper device must be designed to transport this heat. Such technology applies heat pipes and loop heat pipes for temperature equalization and heat transport over long distances.

**Thermal management requirements**

In the case of turbines (gas and steam), the thermal management is vital for the proper operation of this power generation machine. Points of concentrated heat fluxes may represent a potential risk for future failures and thus must have an adequate thermal control. Usually, the application of materials with high thermal conductivity helps the thermal dissipation but sometimes a more elaborated management is necessary. The application of passive thermal control devices as HPs and LHPs can be used to promote such thermal control, especially on surfaces where high heat fluxes are present and there is a potential risk for temperature overshooting that could lead to structural damages or loss of performance. Figure 1 shows the thermal simulation of a turbine and the spots where there are high temperatures. In such places, the positioning of high temperature HPs would be enough to homogenized the temperatures and minimize the thermal stresses caused by the influence of concentrated heat fluxes. Specific designs should also allow the use of LHPs where more precise temperature control is required.

![Figure 1 - Turbines thermal simulation and concentrated heat flux.](image)

Addition of heat pipes in gasifiers allow more compact designs due to the high heat transfer rates between the heat pipe and fluidised bed (in case of using this specific technology for gasification purposes). Some investigation has been performed in late 70’s related to this subject, where it was already pointed the benefits upon using HP technology for gasification process (Basilius and Ewell, 1977). Latest investigations have demonstrated the important gain on the overall gasification efficiency when using heat pipe reformers for biomass gasification.

![Figure 2 - Temperature distribution in a gasifier reactor.](image)

**References**


ETHANOL SORBENTS “SALT CONFINED TO POROUS MATRIX” FOR ADSORPTIVE COOLING

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Abstract

Water and methanol are principal refrigerants for environmentally benign adsorptive cooling (AC) driven by low temperature heat. Methanol has higher vapour pressure and lower freezing point that make it promising for both the air conditioning (AirC) and ice making (IM). However, it is toxic and harmful for environment and its usage is forbidden in some countries. For this reason an increasing attention has recently been paid to ethanol as a refrigerant, which possess similar physical chemical properties, but in contrast to methanol is not toxic. The main drawback of ethanol as refrigerant is its low latent heat (840 J/g) that is a tierce of water. Common ethanol adsorbents are activated carbons which possess the adsorption capacity insufficient for AirC and IM cycles (Li, 2004). In order to compensate this intrinsic drawback new adsorbents which exchange large amount of ethanol within the boundary conditions of particular AC cycles are welcome.

Recently a new family of composites “a Salt inside a Porous Matrix” (CSPMs) has been proposed and tested for water, methanol and ammonia sorption. These materials are shown to combine advantages of salt hydrates, solid porous adsorbents and liquid adsorbents that results in high sorption capacity and low regeneration temperature. The sorption properties of CSPMs can be modified intently or even nanotailored to fit the demands of particular AC cycle (Aristov, 2008).

This study is aimed to the extending of the CSPM approach to preparation of composite ethanol sorbents specialized for AirC and IM cycles. It is based on the fact that a number of inorganic salts MeA can reversibly absorb ethanol by forming crystalline solvates MeA + nC₂H₅OH = MeA∙nC₂H₅OH. Firstly, a review of literature data on the system “salt-ethanol” was performed in order to choose proper salts. Then a number of composites, based on the chosen salts and various matrices were synthesised and their sorption ability was measured under operating conditions of typical AC cycle. Finally the most challenging material was selected and its isosteric chart was measured. This chart was used to plot the basic AirC and IM cycles and estimate its performance.

Findings and summary conclusions

Express-testing of an adsorbent is a time saving tool to evaluate the amount of working fluid exchanged within AC cycle (Gordeeva, 2007). It was used to test the prepared composites and some common porous adsorbents under typical operating conditions of the AirC (T_{ev} = 283 K) and IM (T_{ev} = 273 K) cycles (T_{con} = T_{ads} = 303 K, T_{des} = 363 K) (Fig. 1). The most promising appears to be LiBr/SiO₂ composite which sorbs w_{ads} = 0.68 g/g at T_{ads} = 303 K and P_{ev} = 30 mbar. The majority of ethanol sorbed was removed during desorption stage (T_{des} = 363 K and P_{con} = 100 mbar), so that the uptake variation between the rich and weak isosters was Δw = 0.56 g/g. LiBr/SiO₂ composite represents quite large uptake variation Δw = 0.34 g/g for IM cycle as well. It was larger than that for the conventional adsorbents among which an activated carbon fibers ACF20 ensured the largest variation Δw = 0.47 g/g for AirC cycle. The advantage of the new composite becomes more pronounced in term of ethanol sorption per unit volume of the adsorbent: LiBr/SiO₂ composite exchanges Δw = 390 kg/m³ along the mentioned cycle, instead of Δw = 25 and 70 kg/m³ for ACF20 with the packing density ρ = 50 and 150 kg/m³ (El-Sharkawy, 2006). For LiBr/SiO₂ composite we directly measured the isosteric adsorption chart and plotted the AirC and IM cycles (Fig. 2). The uptake variation checked from the plot (0.56 and 0.40 g/g for the AirC and IM cycles) agreed well with the express-evaluation. The isosteric heat of sorption H_{is} = -(45 – 51) ± 2 kJ/mol was calculated from the lnP – 1/T diagram.
The data obtained allows an estimation of the Coefficient Of Performance (COP) of AC cycles based on “LiBr/SiO₂ composite – ethanol” working pair considering the following approximations: a) the specific heat of the composite is a sum of those for SiO₂, LiBr and liquid ethanol with appropriate weight coefficients; b) the desorption heat is an average value of those for boundary uptakes. The COP of the cycle was estimated as \( \text{COP} = \frac{Q_{ev}}{Q_{ish} + Q_{des}} \) = 0.66 and 0.61 for the AirC and IM cycles, respectively, that is comparable with COP of the best adsorbents of water and methanol. Thus the design of CSPM with the properties adapted to particular AC cycles allows extra-large variation of the uptake within the cycle borders which almost compensates the low vaporization heat of ethanol.

**Nomenclature**

- COP – Coefficient Of Performance
- \( H \) – isosteric enthalpy of adsorption, J/mol
- \( Q \) – heat, J/g
- \( P \) – ethanol pressure, mbar
- \( T \) – temperature, K
- \( w \) – ethanol uptake, g/g

**Subscripts:**

- ev – evaporation
- con – condensation
- ads – adsorption
- des – desorption
- ish – isosteric heating

**References**


EXPERIMENTAL RESULTS OF A SMALL-SCALE TRIGENERATION SYSTEM
WITH A 5.5 KW\textsubscript{E} INTERNAL COMBUSTION ENGINE AND A 4.5KW\textsubscript{C}
ABSORPTION CHILLER

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Abstract

Within the project of the 6\textsuperscript{th} FP called PolySMART different small-scale micro-trigeneration systems are being developed and tested. These systems consist of a micro-CHP (Combined Heat and Power production) unit, whose heat can be also used to drive a thermally driven chiller, and produce cooling for different purposes. This paper describes the installation and the first experimental results of one of these demonstration systems, a combination of a Senertec ICE (Internal Combustion Engine) and a Rotartica absorption chiller, built in the facilities of Ikerlan-IK4.

Description of the installation

The small-scale CHCP (Combined Heat, Cooling and Power production) installation provides heating, cooling and DHW (Domestic Hot Water) to some office rooms and laboratories of Ikerlan-IK4 facilities (see layout in Figure 1). Basically, the installation consists of the integration of a Senertec HKA G 5.5 cogeneration unit (5.5 kWe, 12.5 kWth) and a Rotartica 4.5 unit (4.5kW \( @ T_{DCh}=90^\circ C, T_{RCh}=35^\circ C, T_{CCOut}=12^\circ C \)), which is a thermally driven chiller with rotary absorption technology. The re-cooling heat is rejected through a dry cooling tower, provided by Rotartica.

Figure 1: Layout of the small-scale trigeneration system

Highlights of the installation are, first, the use of a special version of the Senertec micro-cogeneration unit that is able to work at higher temperatures (up to 90°C) in the heat recovery circuit in order to optimize the coupling with the TDC (Thermally Driven Chiller). Secondly, the development of control strategies in the management of the storage tanks to improve the overall performance of the micro-CHCP installation.
The installation was completed by June 2007, and has been running since then. Its performance has been exhaustively monitored, making a special effort to ensure high accuracy of the measurements. Linear error propagation analysis was also carried out in order to identify the accuracy of all the Performance Figures derived from the monitored data.

First Results

The installation and its components have proven to be robust. The Performance of the CHP and TDC units is in accordance with the performance data provided by the manufacturers. During the summer of 2008, the installation ran with a specific control strategy that promoted an early start of the cooling production. This strategy avoided the running of the TDC in the most unfavourable hours, and improved the overall electricity consumption ratio of the whole installation. The higher chiller capacity and efficiency achieved by this means helped also in the coupling of the CHP and the TDC. In this coupling process, it turned out also crucial the achievement of higher output temperatures in the recovery circuit of the CHP.

As an example of overall performance data in the summer of 2008, the following data is given:

<table>
<thead>
<tr>
<th>Operation time</th>
<th>Eprod</th>
<th>QChpGN</th>
<th>Qchp</th>
<th>QTdcDe</th>
<th>QTdcCc</th>
<th>QDHW</th>
<th>ECons</th>
<th>ECOP</th>
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</thead>
<tbody>
<tr>
<td>Units</td>
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<td>kWh</td>
<td>kWh</td>
<td>kWh</td>
<td>kWh</td>
<td>kWh</td>
<td>kWh</td>
<td>-</td>
</tr>
<tr>
<td>Uncertainty</td>
<td>-</td>
<td>±1.7%</td>
<td>±2.5%</td>
<td>±2.2%</td>
<td>±2.8%</td>
<td>±4.6%</td>
<td>±3.5%</td>
<td>±0.2%</td>
</tr>
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<td>737</td>
<td>28</td>
<td>183</td>
</tr>
<tr>
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<td>677</td>
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<tr>
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<td>4,844</td>
<td>2,825</td>
<td>172</td>
<td>750</td>
</tr>
</tbody>
</table>

Table 1: Performance data during the summer of 2008. \( E_{prod} = \) electricity produced in the CHP, \( Q_{Chp\_GN} = \) natural gas consumed, \( Q_{chp} = \) heat recovered in the CHP, \( Q_{Tdc\_De} = \) heat used in the driving circuit of the TDC, \( Q_{Tdc\_Cc} = \) cooling produced in the cooling circuit of the TDC, \( Q_{DHW} = \) heat used for Domestic Hot Water, \( E_{Cons} = \) electricity consumed by the BOP (Balance of Plant) and the TDC, and \( ECOP = \) cooling produced per electricity consumed in the CHCP installation (\( = \frac{Q_{Tdc\_Cc}}{E_{Cons}} \)).

Findings and summary conclusions

A small-scale trigeneration system has been successfully built and has been monitored since June 2007. Highlights of the installation were: the CHP adjustment to achieve higher temperatures in the recovery circuit and the design of control strategies that reduce the ratio of overall electricity consumption of the whole system. Due to typically small temperature differences, it is highly recommended to make always detailed error propagation analysis to asses the quality of the monitored data in small-scale CHCP installations.
WATER AS REFRIGERANT – EVAPORATOR DEVELOPMENT FOR COOLING APPLICATIONS

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Abstract

Within the last years intensive research work towards the topic of developing efficient adsorption storage and heat pump systems was carried out at Fraunhofer ISE. In doing so it has been found out that not only the optimisation of the adsorber but also of the evaporator is required. For most of the materials used in adsorptive cooling machines with solar or waste energy, water is used as refrigerant. To run the process water is to evaporate in a temperature range between 3 an 20 °C, which corresponds to a pressure of 7.6 to 23.4 mbar. Even though there is extensive literature on evaporation of water, data on the characteristics of water evaporation under vacuum conditions are still scarce.

The evaporator development at Fraunhofer ISE using water as refrigerant concentrates on two different levels. On the one hand basic research - orientated. Here, a test rig is used to determine the (pool) boiling curves for different surface structures under pure water vapour atmosphere conditions. This test rig has been put into operation in 2007 and first results were presented at the International Sorption Heat Pump Conference 2008 in Seoul (Schnabel et al, 2008).

On the other hand application – orientated. Therefore a second test rig serves to characterize different heat exchanger structures at the same pure water vapour atmosphere conditions, but taking additionally the influences of heat exchanger design and internal fluid flow into account. Its design, the measuring principle and first results of both a finned and a tube bundle heat exchanger, i.e. a comparison of different evaporation rates and temperature conditions, have already been presented at the International Conference on Solar Air Conditioning 2007 in Tarragona (Schnabel et al, 2007).

The paper will present results of both test rigs. At the evaporator test rig the presented measurements focus on investigating different market available micro- (cf. Figure 1) but also different combinations of micro- and macro-structured copper tubes (cf. Figure 2).

Figure 1: Micro-structured copper tubes
Figure 2: Combination of micro- and macro-structured surfaces (Ip-tubes)
In market available heat exchanger designs, the boiling regime is limited to convective boiling. For this reason thin-film evaporation is the most efficient way to evaporate at low driving temperatures as it will be demonstrated by the measurements for different microstructures.

On the more basic research orientated level the general boiling characteristic of different surface structures is investigated. Here, one of the central questions is whether the nucleate boiling regime can generally be reached using water as refrigerant within the pressure range mentioned above or not. First results on exemplary samples were quite promising as they have been presented on the International Sorption Heat Pump Conference 2008 in Seoul (Schnabel et al, 2008). In this paper the results for metallic short fibre structures will be presented. Figure 3 and Figure 4 illustrate both a picture and the dimensions of this fibre structure (copper).

![Figure 3: Metallic short fiber structure with a porosity of 83%](image1.png)

![Figure 4: Dimensions of metallic short fiber structure (porosity of 83%)](image2.png)

For this structure nucleate boiling is already reached at wall superheats of approx. 4 K for 2 kPa and 7.5 K for 1 kPa.

The paper will report on the different measurement results and will discuss their influence on further development of water evaporators for low pressure applications.

**Key Words**

Evaporation, water, low pressure, vacuum

**References**

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SOLAR COOLING AND AIR-CONDITIONING IN THE GERMAN SOLARThERMIE 2000PLUS PROGRAMME – INSTALLED PLANTS AND MONITORING RESULTS

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Abstract

In the national funding scheme Solarthermie 2000plus, the German Ministry for the Environment, Nature and Nuclear Safety supports large solar thermal installations in different application fields, of which one is solar cooling and air-conditioning of buildings. The funding programme is executed by the project management organisation PtJ Jülich.

Realised systems are equipped with a monitoring system, allowing for a comprehensive analysis of the plant operation as well as of the overall energy balance, including the use of backup systems and parasitic electric energy use. Monitoring partners in this programme are ZfS Hilden, University for the applied Sciences Offenburg and the Technical Universities Chemnitz and Ilmenau.

Fraunhofer ISE is responsible for accompanying research in solar cooling demonstration projects, providing decision support to the PtJ in the selection of applications for funding, and for the analysis of monitored data.

Three realised solar cooling projects in Solarthermie 2000plus and first monitoring results are presented:

1. Solar air-conditioning of 26,000 m² office area at the Technology Center of FESTO AG, Berkheim/Esslingen. With the completion of the building in 2001, cooling of the office area is performed with a central chilled water network, operated by three adsorption chillers with a chilling capacity of total 1050 kW. Driving heat for the chillers was provided by gas boilers and by waste heat from the production facility. Now, a third heat production system is added, consisting of 1218 m² vacuum-tube collectors. The collector fluid is pure water. The produced cold is distributed by means of supply air cooling and slab cooling. The collector is in operation since the end of 2007; the monitoring system was installed by the University for the applied sciences Offenburg.

2. Solar cooling at a radiological practice in Berlin. In this application, cooling demand appears throughout the day and throughout the year, mainly caused by tomographic equipment. The load is basically covered by a centralised chilled water network (operated by an electrically driven compression chiller), serving also other consumers in the building. In the radiological practice, an additional absorption chiller with 10 kW chilling capacity lowers the use of network cooling during daytime, since the chiller is driven with solar heat from a 40 m² evacuated tube collector array. The monitoring system was installed by the Technical University Chemnitz.

3. Solar air-conditioning at the low energy building of the Technical colleges at Butzbach. In this building, demand for cooling and air conditioning arises from high internal loads through occupation and computer training courses. The building is used throughout the summer. At present, air-conditioning of the 335 m² seminar area is performed with two supply/return air units with heat recovery. The additional solar cooling system, which went into operation end of 2008, consists of 60 m² vacuum collector tubes, providing driving heat for two absorption chillers of 10 kW chilling
capacity each. In periods of high demand for air-conditioning and cooling, the concept allows for a separate operation of the chillers: while one operates at low chilled water temperatures for supply air dehumidification, the second one operates the chilled ceilings in the seminar rooms at higher chilled water temperatures. The system will be operated primarily in a solar autonomous cooling mode. The monitoring system is planned and installed by ZfS Hilden.

![Diagram](image)

**Figure 1:** Simplified scheme of the solar assisted air-conditioning plant at FESTO AG in Berkheim/Esslingen.

![Diagram](image)

**Figure 2:** Simplified scheme of the solar air-conditioning system in the low energy building of the Technical colleges at Butzbach.
THERMALLY DRIVEN AMMONIA-SALT TYPE II HEAT PUMP:
DEVELOPMENT AND TEST OF A PROTOTYPE

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Abstract

Upgrading industrial waste heat over the pinch temperature has great potential for reducing industrial energy use. For small temperature lifts, this can be achieved using (re)compression systems. For high temperature lifts (over 50 ºC), the options are more limited. At ECN, a heat pump has been developed to increase waste heat in the temperature range of 90 to 160 ºC with more than 50 ºC up to temperatures of 220 ºC. This type II thermally driven heat pump is based on absorption and desorption of ammonia onto LiCl, the low-temperature salt and MgCl₂, the high temperature salt. Currently, a prototype system is build and the performance of this system will be presented in this paper.

Heat pump principle

The heat pump cycle is based on the following reactions for charging and discharging the system:

**Charge:**
- LiCl1NH₃ + 2 NH₃ → LiCl3NH₃ + heat (T_{ambient})
- MgCl₂6NH₃ + heat (T_{wasteheat}) → MgCl₂2NH₃ + 4 NH₃

**Discharge:**
- LiCl3NH₃ + heat (T_{wasteheat}) → LiCl1NH₃ + 2 NH₃
- MgCl₂2NH₃ + 4 NH₃ → MgCl₂6NH₃ + heat (T_{abovepinch})

The pressure at charging is approximately 1 bar while discharging occurs at pressures up to 20 bars.

Design

Earlier systems using these salts showed limited performance due to degradation of the salts, salts escaping from its matrix and poor heat and mass transfer to and from the salts. To avoid these problems, a method was developed to deposit the salts into metal foam, which increases the heat transfer while limiting the degradation and mobility of the salts, and thus maintains proper mass transfer. The final design of the reactors is based on a shell and tube type heat exchanger in which each fin supports a sheet of metal foam containing the salt. Fig.1 shows the final design of the reactor with on the left the shell and tube heat exchanger containing 80 fins with coated metal foam while on the right the fin layout is shown. The overall system design is shown in Fig. 2.

Fig 1. Left: Reactor design with shell and tube heat exchanger, right: fin layout
Performance

Each reactor is designed to contain about 1 kg of salt and generates approximately 2500 kJ of heat upon absorption of ammonia. With a cycle time, consisting of one charge and one discharge phase, is estimated to be 1 hour with a resulting power output of 250W of useful heat. The total input of waste heat is estimated to be 1kW resulting in an overall COP of 0.25. These values are based on model calculations that assume heat transfer as the rate limiting factor for the cycle time.

Measurement program

To confirm the model calculations, measurements will be conducted at various temperatures, cycle times and flow rates. A custom made unit provides the system with Cal-Flo thermal oil at the desired temperatures and flow rates. At set times, the unit will switch between the charge and discharge phase of the heat pump cycle, changing the provided temperature from low&medium temperature to medium&high temperature. The heating and cooling capacity of the unit is approximately 5 kW.

The thermal power production and dissipation of the heat pump system is determined from the flow rate of the thermal oil, its heat capacity and the difference in temperature at the inlet and the outlet of the reactors. To obtain more insight in the reactors processes, additional parameters are monitored. These parameters include the ammonia pressure in each reactor and the pressure difference between them and the temperatures of the salt in the metal foam matrix. Currently the system is assembled and mechanically tested and the first performance measurements are due January 2009.
THERMOHYDRAULIC MODELING AND DYNAMIC SIMULATION OF POLYGENERATION SYSTEMS

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Abstract

The combined generation of heat, cold and electric power (CHCP or polygeneration) represents one attractive alternative to improve the primary energy efficiency of the energy supply. In the frame of the European PolySMART project (PolySMART, 2008) 12 demonstration plants in 7 different European countries have been running using different technical solutions to polygeneration, in order to investigate and improve the performance of these types of systems. One of these demonstration plants has been installed at an office building of the Institute of Energy Engineering of the TU Berlin. The plant combines combined heat and power production (CHP) at the power plant with decentralized cooling via a thermally driven chiller (TDC). A 10 kW thermally driven absorption chiller runs with hot water from the municipal district heating network and provides cooling to 2 server rooms, 2 seminary rooms and 3 offices rooms.

To achieve a better understanding of the dynamic behavior of the hydraulic system and to improve the performance of the control strategies, dynamic thermohydraulic models of the systems have been developed at the TU Berlin. A new package (a set of cogeneration components and systems models) has been designed using the object oriented simulation language Modelica (Modelica Association, 2008). With this package, different cogeneration systems can be modeled and simulated.

Modeling approach

In a first approach, a Modelica package was developed based on the work made by D. Klein in his Masterthesis at the TU Berlin (Klein, 2008). Different models were developed for absorption chillers, water storage tanks, heat rejection devices, pumps, valves and control blocks.

In this first package, the pressure levels at the different circuits are not considered and the pressure drops taking place in the different components are not implemented. For the electric consumption of the single devices simplified equations like those used in TRNSYS are used.

This model approach finds difficulties when dealing with complex systems with large amounts of valves and pumps involved, where the mass flow rates at the cooling circuits are coupled with the pressure drops at the valves, and vary with valve position and pump rotational speed.

In order to deal with these difficulties, a new package is developed based on the Modelica Fluid Library (Casella et al., 2006). The new package uses and/or adapts standard components from the Modelica Fluid library for valves, pumps and tubes, where different expressions for the calculation of pressure drops are included. The models developed in the first approach are translated and adapted to the Fluid specification and include pressure drops. For pumps and valves, the standard components of the fluid library are adapted to account for non ideal behaviour of these devices.

The new package based on the Modelica Fluid Library implements also reusability, as the new components can be now be used in combination with other models based on this Library, like the model developed by Schicktanz and Nuñez (Schicktanz, 2008) for adsorption chillers or thermodynamic detailed models for the absorption chiller under development at the TU Berlin.
Contents of presentation

In this paper, the models developed for the plant installed in Berlin are presented and its results are compared with those measured at the demonstration plant.

The simulation results show the advantages the developed models offer for the simulation of large and complex thermohydraulic systems, with a large amount of controlled valves and pumps involved, determining by its interaction the pressure drops and chilled water mass flow rates at the different cooling circuits, and the parasitic power consumption of the polygeneration plant.

References


ENERGY AND CO₂ EMISSION PERFORMANCE OF BUILDING INTEGRATED POLYGENERATION SYSTEMS

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Abstract

Building integrated co- and polygeneration systems are related to a potential for reductions in primary energy demand and associated greenhouse gas emissions. The distributed generation nature of the technology also has the potential to reduce electrical transmission and distribution inefficiencies and reduce utility peak demands. However, the technology competes with other supply options like innovative central electricity generation and heat pump technologies and - for renewable energies - solar thermal and photovoltaic systems and biomass fuelled systems.

In the frame of the European PolySMART project (PolySMART), the performances of natural gas driven polygeneration (combined generation of heat, cold and electric power) systems were assessed in terms of primary energy and CO₂ emissions. The polygeneration systems considered basically include a cogeneration device (CGD) and a thermally driven chiller (TDC), supplemented with a boiler and a mechanical chiller. Buildings considered are residential multi-family buildings and small office buildings.

Modeling approach

Two levels of modelling are applied: (a) detailed and transient modelling of system and building using TRNSYS (2006), and (b) a simplified approach, using a spread-sheet program based tool.

In the TRNSYS simulations, cogeneration models specified within Annex 42 of the International Energy Agency's Energy Conservation in Buildings and Community Systems Programme (IEA-ECBCS) (Kelly and Beausoleil-Morrison 2007) and the models for the absorption TDC and the cooling tower as described by Corrales Ciganda (2007) are used. For the combustion based cogeneration device, data from laboratory measurements (Beausoleil-Morrison 2007) were used for the calibration of the model. For the SOFC devices, performance data for different capacities were derived using a detailed SOFC system model, comprising individual models for the SOFC cell, stack and balance of plant components. This SOFC model was specifically developed for this purpose.

With the simplified tool, implemented in MS Excel, the calculations are based upon hourly load profiles for space heating/cooling (determined e.g. using a single zone or a multi-zone building model like TRNSYS), domestic hot water and electric demand (Kräuchi and Dorer 2008). CGD and TDC are assumed to operate also at part-load, but with constant efficiencies, however. The capacity of the TDC was assumed to fully comply with heat power supplied from the CGU. For the CGU and TDC analyzed, average efficiencies based on measured performance data or on manufacturer’s data were used. Several operation modes for heat and cold demand driven operation and a modulation capacity within predefined power range were considered. The locally generated electricity was either directly used or exported into the grid.

Buildings considered are residential multi-family buildings of different energy levels, and office buildings of variable sizes, heated and cooled using thermo-active building systems. Building and occupants related inputs are defined for heat (total of space heating and hot water), cold and electric power.
Performance evaluation

The performance evaluations are based on the methodology developed in the frame of IEA Annex 42 (Dorer and Weber 2007). The energetic system performance was determined in terms of non-renewable primary energy (NRPE) demand and carbon dioxide equivalents emissions, and compared to the reference system. The reference system for all cases comprises the gas boiler and the mechanical chiller, and grid electricity according to the selected mix. Three different grid electricity mixes were considered, using primary energy and emission factors from the Ecoinvent database: European mix according to UCTE, combined cycle power plant and Swiss mix.

This paper compares TRSNYS simulations with the simplified tool, and highlights the applicability of such simplified tools for sensitivity studies. Then results are given in terms of increase of operation time of the CGD when coupled to the TDC and in terms of NRPE and CO2 equivalents reductions, for a selected number of cases (building, CGD and TDC capacities and efficiencies, grid electricity generation mix).

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INFRARED IMAGING AS A MEANS OF CHARACTERIZING TEMPERATURE DISTRIBUTIONS OF THERMOACOUSTIC REGENERATORS

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Abstract

A thermoacoustic engine (TE), which has no moving parts, is a new type of long-life, non-polluting engine that can utilize thermal energy, even low grade thermal energy. It could be widely used in the fields of electronic device cooling, natural gas liquefaction, and pulse tube refrigeration. Onset is one of the most important processes of TEs, and it indicates the transition of thermoacoustic engines from the stationary to periodic oscillating state when the temperature gradient in the regenerator is larger than the critical value (so-called critical temperature gradient). As the most fundamental thermoacoustic phenomenon, onset mechanism attracts more and more academic attentions. However, to date there is not conclusive understanding of it and plenty of work should be done. At present, most researches are qualitative, lacking of comprehensive, accurate, and quantitative analysis of the onset process, detailed and comprehensive experimental data are urgently needed.

The regenerator is a key component of a TE, its temperature distribution has a dominant influence on the onset process. Visual images of the radiated heat of the objects can be obtained using infrared thermal imaging equipment in the absence of light, thus the visual temperature distribution can be measured. The temperature evolutions of the thermoacoustic regenerator are first visually measured by us in the onset and damping processes in the cases below: comparison of the onset and damping processes with and without the pressure disturbance method (Qiu L.M, 2006), as it can significantly decrease the onset temperature of the engine; temperature evolution comparison of the regenerator with and without the Helmholtz resonator, as a TE connecting a Helmholtz resonator (Sun D.M., 2008) at the compliance delays the onset process; temperature evolution comparison of the regenerator with and without the streaming suppression method in the loop of a TE, as the temperature distribution of the regenerator changes with this method.

The study supplies a visual and comprehensive understanding of the regenerator performance during the onset and damping processes and makes quantitative discussion on the factors that influence the two processes, which enhance the understanding of the onset mechanism and provide an important reference on CFD study of the TE.

Description of the system

Figure 1 shows the schematic diagram of the traveling-wave TE with an infrared imaging camera. The travelling-wave TE is detailed elsewhere (Sun D.M., 2005), which consists of a loop and a resonator tube. The loop is the main part converting thermal energy to the acoustic power, which consists of heater, cooler, regenerator, and phase adjusters such as the feedback tube and compliance. A Helmholtz resonator is connected to the TE with a ball valve, which can be opened when it is necessary. An infrared imaging camera made by FLIR SYSTEM is used to measure the temperature distribution of the regenerator. The pressure data acquisition system is used in the experiments, which consists of pressure sensors (linear silicon piezoelectric type, KPY 46R), a data acquisition clip (NIPC-1200, 12 bit precision and a sampling frequency of 10 kHz), a computer, and a self-developed program for LabView 7.1 by National Instruments, Inc. Heating power is adjusted by changing the charging voltage to the heaters, and is displayed by a digital dynamometer.
Findings and summary conclusions

Clear images of the temperature distribution were obtained by using infrared imaging camera as a new method to measure the temperature evolutions in the regenerator, which enhances our understanding of the onset mechanism and regenerator function of a TE.

References

EXPERIMENTAL INVESTIGATION OF ORGANIC RANKINE CYCLES FOR DOMESTIC MICRO CHP

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Abstract

Many research centres are involved in development of a micro CHP cycle for operation in single households or small enterprises. The major problem in such developments is an expansion machine. Most of the time the scroll compressor has been appropriately modified for that purpose. The major problem with its operation is that the internal efficiencies are not too impressive, namely do not exceed 50%. Bearing in mind that a micro CHP is theoretically capable to produce about 3kW of electricity (at a condenser capacity of 10kW), and then such a device is unsuitable for commercial implementation. The Institute of Fluid Flow Machinery PAS in cooperation with the Gdansk University of Technology has started significant efforts on assembling the prototype of such cycle based on the expansion devices available on the market. The present paper reports the major findings from the research into two prototypes of such cycles based on a scroll compressor and a pneumatic device.

Experimental

The experimental activities started some time ago aimed at development of the prototype realizing the C-R cycle, Fig. 1. The working fluid has been selected as R123, however perspective the Solvay SES36 is envisaged. R123, similarly as SES36, is the fluid having relatively good heat transfer characteristics, is inexpensive and obeys a majority of requirements facing the perspective working fluids. Evaporator and condenser are the plate heat exchangers manufactured by Secespol with heat transfer surfaces of 1.8m² (LB47-40 PCE) and 0.9m² (LB47-20 PCC) respectively, corresponding to 15kW and 11 kW capacities. Thus far only preliminary experiments have been accomplished with the inverted scroll compressor LG ELECTRONICS model HQ028P featuring the capacity of 6.974 kW. The optimal parameters determined by the producer for operation with R407C are $T_{\text{evap}}/T_{\text{cond}} = 7.2/54.4$ °C. The dimensions of the scroll were: width/depth/height equal to 235/235/374 mm.

Figure 1: Schematic of experimental stand
Preliminary results of experimental research performed using the scroll expander LG HQ028P are presented in Table 1. Physical properties of R123 have been taken from Refprop8. Notation for nodal points of ORC cycle has been presented in Fig. 2. In the course of calculations determined have been values of cycle thermal efficiency ($\eta_t$), which was evaluated without account of work necessary for pump operation, the maximum Carnot efficiency ($\eta_C$), internal efficiency of expansion machine ($\eta_i$) and finally the exergetic efficiency ($\eta_b$). The internal efficiency of expansion machine was determined from equation (1), where state 1 denoted inlet to expander and 2 – the outlet (2s is the state after adiabatic expansion). In all cases state 1 was found in superheated steam region.

\[ \eta_i = \frac{h_2 - h_1}{h_1 - h_{2s}} \quad [1] \]

\[ h_1, h_2, h_{2s} \text{– enthalpy}[kJ/kg] \]

The results of in-house experiments have been compared with experimental findings of Quoilin (2007). In these experiments R123 was also a working fluid and scroll expander was applied for expansion in the pressure range from $\sim 10$ to $\sim 3$ bar. The comparisons are presented in Table 1. Subsequently the scroll expander has been substituted with a pneumatic device featuring the "turbine mechanism" together with the planetary gear. Adapted device was a BoSCH drill (series 180W, model 0607 153 513), appropriately modified to the needs of experiments. The modification relates to separation of mechanisms of change of rotational velocity of the spindle and a set of valves and followers. The basic parameters of the drill were: rotational velocity 1800/3000 rev/min, power 180 W, feeding pressure 6.3 bar, weight 0.96 kg. The body of the drill was made of steel. In experiments the machine was coupled with the hydraulic brake which enabled determination of effective power. Somee results of measurements are presented in Table 1.

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Conclusions

Experiments showed the superiority of pneumatic device over the scroll expander, indicating the possible internal efficiencies in the range of $61 \div 82\%$. The volume of such device is much smaller than the scroll expander which makes it more suitable for a domestic micro CHP. Small rotational velocities enable to conclude that conversion to electricity will also be simpler in case of a pneumatic device. The pneumatic device under scrutiny here could be an alternative to the typical vapour turbine in the ORC cycle.
A NEW CRITERION FOR SELECTION OF WORKING FLUID FOR SUBCRITICAL AND SUPERCRITICAL MICRO CHP

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Abstract

In recent years there is observed a clear tendency, both worldwide and in the countries of European Union (EU), to increase the importance of so called dispersed generation, based on local energy sources and technologies utilizing both fossil fuels and renewable energy resources. Micro CHPs based on organic Rankine cycle fit very well to that strategy.

Analysis of literature regarding the relevant organic fluid suitable for the ORC cycle is not complete and does not show how to appropriately select good fluid to specified conditions. There are selection criteria provided in literature but they are of no general character. Up to date research was focused mainly on subcritical cycles. There is, however, an unexplored area of supercritical parameters which offer also great opportunities for exploitation. The analysis of supercritical fluid parameters may lead to higher efficiencies making the micro CHP even more attractive. The critical point of organic fluids is reached at lower pressures and temperatures compared with steam. Therefore supercritical fluid parameters are much easier to be realized in practical applications in these cases than in case of water. However, the objective for a domestic micro CHP is also designing of small sized heat exchangers and for that reason not all supercritical fluids could be suitable, which will also be discussed in the paper. A simple analysis has been carried out which resulted in development of a thermodynamical criterion for selection of an appropriate working fluid. Such criterion should be used with other criteria related to environmental impact, etc.

The thermodynamical criterion

The algorithm of calculations is based on well known relations describing the Rankine cycle. It has been assumed that the maximum pressure in installation should not exceed 60bar nor temperature of heating oil 300°C, respectively. Notation of state points for subcritical and supercritical cycles is presented in Fig. 1.

Figure 1: Schematics of subcritical (a) and supercritical (b) cycles

The analysis commences with the expression for the cycle efficiency:

$$\eta = \frac{t_{\text{net}}}{q_{\text{in}}} = \frac{h_3 - h_2}{h_1 - h_2}$$

[1]

In relation (1) enthalpy change due to the presence of pump has been neglected. Enthalpies present in (1) can be written in terms of temperatures and fluid properties. In the case of a subcritical cycle the terms of a corresponding liquid saturation state yields:
On the other hand, in case of a supercritical cycle that is:

\[ h_1 = h_3 + c_p (T_1 - T_2) \]  

In both cases the enthalpy at the end of expansion yields:

\[ h_2 = h_3 + x_2 h_{lv} + \Delta h_{superheat} \]  

Relation (4) is a general formula describing the state after the expansion in turbine. In case of expansion of dry fluids, the quality after the expansion in turbine is equal \( x_2 = 1 \) and some vapour superheat, \( \Delta h_{superheat} \), is present, whereas in case of expansion of wet fluids \( \Delta h_{superheat} \) = 0. In general two latter terms in relation (4) can be combined to yield a formulae:

\[ h_2 = h_3 + \Delta H(T_2) \]  

In (5) \( \Delta H = x_2 h_1 \) or \( \Delta H = h_{lv1} + \Delta h_{superheat} \). Substituting relations (3), (4) and (5) into (1) we obtain the cycle efficiency. In case of subcritical cycle it yields:

\[ \eta = \frac{h_1 + c_p (T_1 - T_2) + h_{lv} - h_3 - \Delta H(T_2)}{c_p (T_1 - T_2) + h_{lv}} \]  

In case of supercritical case it is:

\[ \eta = \frac{h_1 + c_p (T_1 - T_2) - h_3 - \Delta H(T_2)}{c_p (T_1 - T_2) - h_3} \]  

Temperature difference between condensation and evaporation levels can be expressed in terms of the Carnot cycle efficiency and then for the subcritical cycle:

\[ \eta = 1 - \frac{\Delta H(T_1)}{h_{lv}(T_1)} = 1 - \frac{\Delta H(T_2)}{h_{lv}(T_2)} \]  

In case of the supercritical cycle:

\[ \eta = 1 - \frac{1}{c_p T_1 \Delta H(T_2)} = 1 - \frac{1}{Ja(T_1, T_2) \eta_c} \]  

Analysis of (8), valid for a subcritical cycle, enables to conclude that the overall cycle efficiency is a function of a ratio \( \Delta H(T_1)/h_{lv}(T_1) \) and the Jakob number for the same Carnot efficiency. It stems directly from (8) that we should consider the ratios of \( \Delta H(T_2)/h_{lv1} \) and \( c_p/h_{lv1} \), when we want to consider a substance as a working fluid in the subcritical cycle. In case of a supercritical cycle the analysis of relation (9) enables to conclude that the overall cycle efficiency is a function merely of the ratio \( c_p T_1/\Delta H(T_2) \) for the same Carnot efficiency. It stems directly from (9) that we should consider the \( c_p T_1/\Delta H(T_2) \) ratio when we want to consider a substance as a working fluid. Values of \( Ja(T_1) \), \( \Delta H(T_2)/h_{lv1} \) \( c_p T_1/\Delta H(T_2) \) will be calculated for a selection of fluids and presented in a full body of the text. Apart from such thermodynamical criterion also other criteria should be considered such as influence on a human, environment, explosive character. Presented criterion points that the best one from those considered thus far is R141b. In case of ethanol and R134a, the quality at the end of expansion process in turbine indicate only a small content of liquid present \( (x>0.95) \), which increases the turbine life span. In case of other fluids the expansion process ends in the superheated steam region, requiring a regenerator to increase the efficiency of the cycle.
SIMULATION OF A SMALL SIZE SOLAR ASSISTED ADSORPTION AIR CONDITIONING SYSTEM FOR RESIDENTIAL APPLICATIONS.

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Abstract

The energy consumption for air conditioning increased very much in the last years especially in Mediterranean countries. Main reasons are the increased level of quality of life and the consequent need of an improved indoor thermal comfort. As direct consequence of this phenomenon the demand of energy from electricity grid increased. In fact there is a clear correlation between the increase in the peak load on the electricity grid in summer and the increasing sales of electric small air conditioner, mainly in southern Europe. In such a background the solar air-conditioning systems can be a viable alternative to conventional air conditioners and in particular adsorption chillers have high potential for solar air conditioning applications. Indeed, such devices can be thermally driven by low temperature heat source (70-90°C), utilize safe and not polluting materials (e.g. water vapour and zeolite or silica gel), do not present moving parts. Moreover, adsorption machines have Coefficient Of Performance (cooling COP) of about 0.5-0.6, which is comparable with the typical COP values for LiBr single-effect absorption chillers operating under the same conditions [1]. Silica gel adsorptive machines with cooling capacity of about 75 kW (or more) are already available on the market since several years. Recently, few units of small-size adsorption machines have been introduced in the market.

This work is focused on the development of a dynamic model simulating the operation of a small size (5 kW) solar assisted adsorption air conditioning system for residential or commercial buildings. The system lay-out is presented in Fig. 1.

Fig. 1 Lay-out of the solar assisted air conditioning system. 1) Solar collectors; 2) Thermal storage; 3) Back-up boiler; 4) Adsorption chiller; 5) Dry cooler; 6) Fan coil
The double-bed adsorptive chiller is driven by the thermal energy coming from the vacuum tubes type solar collectors or from the auxiliary boiler. The system also includes a thermal storage system properly dimensioned. The adsorption chiller is cooled by an air/water cooler and produces chilled water to be used by a common fan coil or a cooling ceiling.

The model was developed in TRNSYS, a software environment with a graphic user interface, able to dynamically simulate the behaviour of each component of the system. The main diagram of the software realized is shown in Fig. 2.

The climatic data used as input parameters refer to the city of Messina. Values of solar radiation, relative humidity, wind speed and direction and ambient temperature were used to perform the simulations. The thermal load profile used for simulations has been calculated considering the thermal needs of a small residential building.

The 5 kW adsorption chiller has been modelled on the basis of experimental data collected on a laboratory prototype realized by CNR-ITAE that uses an innovative heat exchanger and an advanced zeolite type properly synthesized for low temperature heat utilization. Tests on the prototype have been carried at the following conditions: condensation temperature 28-48 °C, evaporation temperature 5-12 °C, regeneration temperature 75-90 °C. At the above mentioned conditions the experimental device produced a cooling power ranging from 3 to 5 kW with a COP of 0.3 – 0.6.

The results of a base-case simulation show that the system is able to produce up to 5 kW of chilled water at 14 °C with an external temperature of 32 °C. Afterwards, a parametric analysis was carried out in order to identify the most influencing parameters and to find out the most appropriate size for the various components of the system with the objective of maximizing the ratio between performance and total installation costs.

References
INFLUENCE OF INTERNAL IRREVERSIBILITIES
ON THE CHARACTERISTIC EQUATION OF ABSORPTION CHILLERS

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ABSTRACT

It is well known that the part load behaviour of sorption chillers is mainly influenced by the external temperatures of hot, chilled and cooling water. In order to better understand the impact of these temperatures on the load characteristics and to simplify the description thereof a fundamental method has been developed several years ago which combines the external temperatures into one characteristic temperature function, which is a double difference, $\Delta \Delta t$.

With the method of characteristic equations the cooling capacity can be expressed as

$$\dot{Q}_E = s_E \cdot (\Delta \Delta t - \Delta \Delta t_{\min}) = s_E \cdot \Delta \Delta t + r_E,$$

where slope $s_E$ and intersect $r_E$ (or $\Delta \Delta t_{\min}$, respectively) are characteristic parameters of the chiller, and $\Delta \Delta t$ is the characteristic temperature difference. For single stage cycles it is the difference of external temperature thrust ($t_D - t_A$) and lift ($t_C - t_E$) weighted by $B$, the Dühring parameter.

$$\Delta \Delta t = \Delta t_E - B \cdot \Delta t_L = (t_D - t_A) - B \cdot (t_C - t_E)$$

Obviously, $\dot{Q}_E$ in Eq. [1] is a linear function of $\Delta \Delta t$ if the characteristic parameters are constant (e.g. load dependent variation of heat transmission coefficients is negligible, constant flow rates etc.). Often the correctness of these assumptions is supported by measurements (or simulations) since $s_E$ and $r_E$ can easily be derived from a linear data fit (see Fig 1 left hand side). Here, values of simulated cooling capacity of a 10 kW single stage absorption chiller under part load conditions are plotted against $\Delta \Delta t$. Closed symbols show the simulation results for external inlet temperature variations between $55^\circ C < t_{Di} < 95^\circ C$, $21^\circ C < t_{Ai} < 33^\circ C$, and $9^\circ C < t_{Ei} < 18^\circ C$, respectively. These 'exact' results are independent of the characteristic equation method.

$$\dot{Q}_E^{\text{global}} = 0.46 \cdot \Delta \Delta t - 0.13$$

$$\dot{Q}_E = f(\Delta \Delta t)$$

$$\Delta \Delta t_{\min} = f(\Sigma \Delta t)$$

$\Delta \Delta t_{\min, \text{fit}} = a \cdot \Sigma \Delta t + b$

$a = 0.052 \text{ K/K}$

$b = -1.56 \text{ K}$

Fig. 1: Simulation results of $\dot{Q}_E$ and $\Delta \Delta t_{\min}$ (symbols) and data fits (solid lines).
From the simulation results, as well as from measurements, a mean characteristic straight line can be derived from a linear data fit which uses all the data points together (thin straight line $\dot{Q}_E^{\text{global}}$ in Fig. 1). In contrast to this 'global' fit procedure of simulated or measured data, it is (in the simulation) also possible to calculate the characteristic parameters directly (i.e. without fitting) for each load condition. These values are called the 'differential' characteristic parameters.

In the differential slope parameter the internal irreversibilities related to the refrigerant mass flow rate (like throttling, superheating and desuperheating of refrigerant vapour) are accounted for inherently. Therefore $s_E$ is relatively constant. But the variation of $\Delta \Delta t_{\text{min}}$ was found to be remarkable, since it holds not only for the irreversibility of solution heat exchanger loss but also for the irreversibilities of changing concentration and necessary temperature glide. Its variation is also plotted as dashed lines in Fig. 1 for three discrete variations, where one inlet temperature ($t_{D_i}$) is varied and the other two are kept constant.

From Fig. 1 it is seen that $\Delta \Delta t_{\text{min}}$ is increasing with increasing $t_{D_i}$ (i.e. increasing thrust) and decreasing for increasing $t_{E_i}$ (i.e. decreasing lift). In contrast, the $\Delta \Delta t_{\text{min}}$ value is more or less constant with decreasing $t_{D_i}$. Consequently, a more uniform behaviour can be achieved, if all the simulated $\Delta \Delta t_{\text{min}}$–values (triangles in Fig. 1) are plotted against $\Sigma \Delta t$, the sum of external temperature thrust ($\Delta t_T$) and lift ($\Delta t_L$). Allthough a considerable scatter is left, a linear fit could be performed using all simulated values (bold line with given values for $a$ and $b$ in Fig. 1). By inserting this linear approximation for the variation of $\Delta \Delta t_{\text{min}}$ into Eq. [1], the characteristic equation is refined as a linear combination of external thrust and lift

$$\dot{Q}_E = s_E \cdot (1 - a) \cdot \left( \Delta t_T - \left( \frac{B + a}{1 - a} \right) \cdot \Delta t_e - \left( \frac{b}{1 - a} \right) \cdot \Delta t_{\text{e}} \right) = s_E \cdot \left( \Delta \Delta t^* - \Delta \Delta t_{\text{min}}^* \right)$$  \[3\]

Where, $\Delta \Delta t^* = \Delta t_T - B^* \cdot \Delta t_e$, $B^* = \frac{B + a}{1 - a}$, $\Delta \Delta t_{\text{min}}^* = \frac{b}{1 - a}$, $s_E^* = s_E \cdot (1 - a)$.  \[4\]

Equation [3] is again a linear function when plotted against the modified $\Delta \Delta t^*$ with characteristic parameters $s_E^*$, $\Delta \Delta t_{\text{min}}^*$ and $B^*$, which all together account for the mean linear change of the irreversibilities in the solution cycle as function of $\Sigma \Delta t$. Open symbols in Fig. 1 show the results when the modified characteristic equation is used (i.e. $\dot{Q}_E$ plotted against $\Delta \Delta t^*$). Hereby the reproduction as a single straight line is improved.

**CONCLUSION**

In this work it is shown that the intersect of a characteristic straight line is constant only apparently. This is a consequence of a nearly linear dependency of intersect as function of the sum of temperature thrust and lift. Thus, the deviation of measured or simulated cooling capacity plotted against the characteristic temperature difference to a global characteristic straight line, is not only scatter, but is due to real partload behaviour as compared to the assumptions in the method. Partly these deviations are inherent in the method if the variability of internal irreversibilities is not considered.

But, as long as the variation of differential slope can be neglected and the intersect can be approximated by a linear function of external temperature thrust and lift, the characteristic equation can be interpreted as a linear combination of linear equations with constant slope and variable intersect. The variation of intersect is mainly due to variable internal irreversibilities in the solution cycle. Alternatively, a modified characteristic temperature difference can be used, where a modified Dühring parameter accounts for their influence.

**NOMENCLATURE**

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<td>$\Sigma \Delta t$</td>
<td>Sum of temperature thrust and lift</td>
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Indices:
- X: placeholder for component
- A: Absorber
- C: Condenser
- E: Evaporator
- D: Desorber
- T: Temperature thrust
- L: Temperature lift
- i, o: Inlet, outlet
INVESTIGATION OF THE PERFORMANCE CHARACTERISTICS OF AN AMMONIA-WATER ABSORPTION CHILLER IN A TRIGENERATION SYSTEM ARRANGEMENT

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Abstract

The energy and carbon savings from CHP installations can be as high as 30% compared to separate heat and power generation but this depends on many factors such as the size of the scheme and the nature of the heat load. For maximum savings there needs to be simultaneous demand of heat and electricity and a fairly constant heat demand throughout the year. In many applications where the demand for heat does not remain constant throughout the year, the utilisation efficiency of CHP plant can be increased if the excess heat is used to drive thermally driven (sorption) refrigeration systems. The integration of CHP with sorption refrigeration technologies is known as Combined Heating Refrigeration and Power (CHRP) or trigeneration.

Trigeneration systems have been used in a number of applications including commercial buildings and industrial facilities. Most of these have been for space cooling applications with a smaller number for refrigeration applications in the food processing industry which requires temperatures below 0°C. This paper is concerned with the investigation and development of modular trigeneration systems for food industry applications based on the integration of microturbines and ammonia water absorption refrigeration systems. Specifically, the paper presents results of experimental investigations on the performance of a 12 kW capacity ammonia-water absorption refrigeration system driven by thermal energy recovered from the exhaust gases of a microturbine in a trigeneration arrangement. The heat transfer between the microturbine exhaust heat exchanger and the generator of the absorption refrigeration system is performed by a heat transfer fluid in a closed heat transfer loop. Tests were performed at different brine temperatures and heat transfer fluid temperatures. The performance of the unit was evaluated and found to compare favourably with the performance of a directly gas fired absorption chiller. In addition, the overall efficiency of the trigeneration system was also evaluated.

Test facility

The trigeneration test facility incorporates three main modules; CHP module, absorption refrigeration system module, and a refrigeration load module. These modules were comprehensively instrumented to measure temperatures, pressures and flowrates to enable the performance of the unit to be evaluated. A schematic diagram of the test facility using oil as the heat transfer medium between the microturbine and absorption refrigeration system is as shown in Tassou et al, 2008.

The CHP module is based on a Bowman 80 kW recuperated microturbine generation package MTG80RC-G with in-built boiler heat exchanger (exhaust heat recovery heat exchanger). The microturbine consists of a single stage radial compressor, single radial turbine within an annular combustor and a permanent magnet rotor (alternator) all on the same rotor shaft. Other systems in the engine bay include the fuel management system and the lubrication/cooling system. Heat recovery from the exhaust gases is performed in a stainless steel flue-gas/liquid heat exchanger. The heat recovery fluid is Diphyl-THT that can operate at temperatures up to 340°C.
The absorption refrigeration system currently employed is a packaged ROBUR (ACF-60LB) which was modified to operate with heat recovered from the exhaust gases of the microturbine. Heat is supplied to the generator by the heat transfer fluid that flows in a loop between the microturbine exhaust heat recovery heat exchanger and the absorption refrigeration system generator. The modification included the replacement of the gas burner with a heat transfer jacket around the generator. The jacket design and heat transfer fluid flow rate were optimised using Computational Fluid Dynamics (CFD).

**Results**

The performance of the absorption unit, in its gas fired format, was established from tests in the laboratory. For brine flow temperatures between –11 °C and +3 °C, the refrigeration capacity varied from 8.5 kW to 15 kW and the COP from 0.32 to 0.57.

A summary of the performance of the modified unit is shown in Figure 1. It can be seen that the modified unit performs as well if not better than the gas fired unit. At brine flow temperature of –8.0 °C both units have a refrigeration capacity of 12 kW. If the heat transfer fluid pump power is taken into consideration, both units will have a COP of around 0.53. The refrigeration capacity of the modified absorption refrigeration system is not significantly influenced by the brine delivery temperature. The brine temperature, however, has a small influence on the system COP which increases as the brine delivery temperature rises until it reaches a plateau at a brine delivery temperature of -4.0 °C.

![Figure 1: Performance of exhaust heat driven absorption chiller](image)

**Reference**

Abstract

In recent years, there has been significant research progress on micro- and mesoporous adsorbent materials for closed and open system heat transformation applications like thermally driven adsorption chillers, adsorption heat pumps or desiccant cooling systems (Henninger 2007). Novel adsorbents with specifically tailored properties are becoming available and new composites out of these materials on novel high surface area support structures are under development. Some of these materials or composites have so far only been synthesized or prepared on a laboratory scale. But with regard to new adsorbers based on these high performance materials or composites and a possible introduction to the market an intense life cycle analysis is needed (Schmidt 2005). This paper presents first results on the stability investigations of different materials like aluminophosphates, silica-aluminophosphates and the new class of metal organic framework (MOF) materials under hydrothermal conditions and evaluates their suitability for the use in sorption heat pumps and cooling machines.

Description of the analysis methods – Short term cycle tests

As the variety of materials within this work is quite large, different analysis methods have been performed. Pure powders in the case of the most novel class of MOF materials, granules or extrudates and even composites (metal support with coated adsorbent material) have been analyzed. This represents the evolution steps each material has to pass through. In an early stage of the development, the samples have been characterized with combined thermogravimetry/differential scanning calorimetry methods in order to evaluate their suitability for the use in sorption process in terms of water uptake.

Figure 1: Loss of water uptake capacity in percent of the initial loading value of a commercial Silica Gel, a SAPO-34 and a MOF sample. Errors are in the range of symbol size.

Following, the samples were exposed to a first stability test. Figure 1 shows the loss of water loading capacity through hydrothermal treatment as percentage of the initial value of a commercial available Silica gel, a SAPO-34 and a sample out of the new class of metal organic frameworks. The performed stability test consists of a repeated adsorption at 20°C followed by a desorption step.
at 140°C. This is realized through stepwise heating up and cooling down of the sample under a continuous humidified carrier gas flow with a partial water vapor pressure of 12 hPa.

As can be seen, the SAPO-34 sample out of the morpholine synthesis shows a dramatic degradation with a loss of capacity of almost 60% compared to the initial value within a few cycles. Degradation can also be observed for the Cu-BTC sample. Although the MOF shows a very high initial water loading, the capacity is reduced by 40% of the initial value within 15 cycles. The widely used Silica Gel 127 B shows a small degradation with a loss of capacity of about 5% within 24 cycles.

Whereas the SAPO-34 and Cu-BTC sample show a dramatic degradation within few cycles in the short term test and will not investigated further, the Silica Gel sample has to pass long term cycle tests which are described in the following section.

Description of the analysis methods – Long term cycle tests

With regard to commercial applications, composite materials consisting of an adsorbent coated or glued to a metallic support with a base area size of 5x5 cm² have been investigated in a cycle test apparatus. Previous to this cycle test and after every several hundred cycles the samples are characterized by thermogravimetric analyses with a fast fingerprint method. The cycle apparatus consists of three independently operated vacuum chambers. By the use of metal coldplates with a fixation carrier (see Figure 2), the samples are heated up to 110°C and cooled down to 20°C within 3 minutes (see Figure 3). Furthermore a pure water vapor atmosphere with constant pressure is applied to the measurement cell. This procedure is repeated several hundred times in order to simulate a life-time stress.

Figure 2: Cycling test facility, one of the chambers opened during preparation. In the center is the sample holder with fixation unit and a UOP-DDZ70 composite.

Figure 3: Temperatures and system pressure during long-term cycling in the cycling test facility.

REFERENCES

SMALL CAPACITY TRI-GENERATION SYSTEMS IN THE EUROPEAN PROJECT POLYSMART: OVERVIEW, DESIGN ASSESSMENT AND EVALUATION METHODOLOGY

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Maider Usabiaga

Alternative generation systems area
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Abstract

The main goal of the geographically dispersed project PolySMART (acronym for polygeneration with advanced small and medium scale thermally driven air-conditioning and refrigeration technology) is to support the market entry of small and medium capacity tri-generation systems. These small-size systems are generally composed of a micro-Combined Heat and Power unit (mCHP) and of a Thermally Driven Chiller (TDC). The operating time of a mCHP system can be significantly increased by using the heat released during power production to drive the cooling machine. The alternative concept of connecting a TDC to a district heating network with central CHP is another aspect of the investigation in the project.

PolySMART aims at demonstrating the cost-effectiveness and reliability of small-scale combined heat, cooling and power (mCHCP) units for a wide range of buildings and industrial applications. Most of the major TDC manufacturers in Europe and manufacturers of mCHP systems are partners in the project along with others participants from industry and universities, making a total of 32 participants from 7 Countries.

Eight demonstration sub-projects using different TDC technologies are currently carried out on twelve sites in seven European countries dealing with a wide spectrum of applications. Different combinations of TDC and mCHP systems have been developed at a nearly pre-commercial stage. The operation of the demonstrators will be investigated under real operation conditions (varying loads) and assessed with regard to cost-savings and energy performance. For that, a common comprehensive metering and monitoring methodology was developed by the partners for an in-depth performance analysis and for determining which kind of application each combination performs best with. Special attention is paid to the collection of experience and lessons learned made during the system design and assembly and field trials in order to provide future customers with a set of best-practise-rules for an optimal sizing, construction and running of their mCHCP plant.

Project status

The project PolySMART started in summer 2006. Beside the prototype building and demonstration activities, the project handle other aspects of market entry issues such as a comprehensive market potential analysis of mCHCP systems, the development of component and system models for engineering, the development of design guidelines and tools for professionals, and the production of documents for training and dissemination. Workshops and seminars for technology transfer and training are scheduled in different countries.

The prototype building, especially the coupling of mCHP with TDC required some tuning and adaptation of the main components in order to have the heat flux released by the one matching well the requirements of the other (essentially in terms of temperature level and flow rate). The usage of thermal storages (cold and hot) was considered with respect to dynamic issues and for energy performance purposes (i.e. running the TDC at night when outside temperature is more appropriate for heat rejection). Another issue was the selection of a high-performance small-size heat rejection system that operates with low electric consumption even at part load.
The installations are equipped with sensors in accordance with the project standards. Thus, major energy fluxes are constantly recorded for performance assessment and investigation of system behaviour. Big attention is paid to energy efficiency. Performance considerations are made also in terms of “primary energy” thus to take into account the parasitic electric consumption of the balance of plant (i.e. circulation pumps, ventilators for heat rejection, ...). Automatic data evaluation procedures were designed and developed to compute commonly defined performance figures and make an harmonised evaluation of all the demonstrators in an homogeneous way.

Content of paper

First, the paper will give a brief account about the project aims and goals and will introduce shortly the project participants.

Second, the paper will present briefly the 12 demonstrators that were designed and are currently operated in the PolySMART project. Technical data of the main components and of the applications will be given in a tabular form to enable comparison between the 12 different demonstration projects. An updated project progress status will be given. The system concept and the interconnection of the main components of a couple of selected demonstrators will be explained in a systematic way using the especially for the project developed “functional block diagram”.

Third, the paper will deal with the evaluation of mCHCP concepts in the design phase, using a theoretical approach in a first step. mCHCP systems directly compete with other processes for electricity and heat and cold production such as central power plants, condensing boilers, compression chillers and their like. Therefore it is important to make sure that the newly developed mCHCP concepts can compete in terms of energy-efficiency, CO₂-reduction and cost-savings. A new approach based on a primary energy analysis was worked out for that purpose. A tool was developed especially for the design phase in PolySMART in order to determine thresholds (i.e. in terms of electrical efficiency for the mCHP or in terms of COP for the TDC) to make sure that the demonstrators will have market attractive features in comparison to a baseline system. The presentation of the method will be followed by a sensibility analysis where the most influencing parameters will be assessed and ranked. This will assist future planners to choose the right components for the design of a CHCP system.

Fourth, the paper will deal with the evaluation program of PolySMART for the investigated demonstrators. The common monitoring method applied to the 12 subprojects will be summarily presented, showing the different level of details opted for, for an comprehensive assessment on a system or on a component level. The used performance figures will be detailed and the tools for automatic data evaluation will be briefly explained.
SYSTEM DESCRIPTION AND FIRST MONITORING RESULTS OF A TRIGENERATION INSTALLATION FOR COMBINED HEATING, COOLING AND POWER

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Abstract

Micro-scale trigeneration systems offer a wide range of technology applications for small industries and other small-scale premises like residential houses. Lately developed systems arouse constantly more interest for their potential of saving primary energy, reducing electricity peaks on the grids and improving cost-benefit ratio of micro-CHP units when combined into trigeneration systems. The operating time of a CHP system can be significantly increased by using the heat produced for thermally-driven cooling, which creates a remarkable improvement in cost efficiency.

Accordingly, in technology supporting projects like the FP6 integrated project PolySMART (POLYgeneration with advanced Small and Medium scale thermally driven Air-conditioning and Refrigeration Technology) [PolySMART] various test installations are being erected and investigated that already provide measuring results of first testing periods. Several types of combined heat and power units as the driving heat source for thermally driven chillers are being investigated in the project’s scope. Basically two types of system topology are studied: micro-scale combined heat power and cooling using small CHP units together with small thermally driven chillers and decentralised cold production with thermally driven chillers within a district heating network fed by centralised CHP units.

One of the first running demonstration installations has been erected at the premises of Technische Werke Ludwigshafen in Germany and is being monitored by the Fraunhofer Institute for Solar Energy Systems. This system corresponds to the second system type presented above. Monitoring data for the first cooling period was obtained where test runs with different driving and cooling temperatures have been performed in order to examine the system’s operation.

**Figure 1: Functional block diagram of the CHCP system in Ludwigshafen, Germany**

**Description of the system**

*Figure 1* shows a functional block diagram of the installation. Depending on the end-user’s demand the system can be operated in cooling or in heat pump mode.
The trigeneration system consists of a thermally driven 5.5 kW adsorption chiller (provided by the company SorTech AG of Germany) which operates with the silica gel / H2O pair. The driving energy is provided by a district heating grid generated by a waste incineration plant. Therefore the thermal operating power of the system is constantly available. The heat rejection system is a 20 kW dry heat rejection system with an optional water spray function for peak loads which is controlled by the adsorption chiller. The system is integrated in an existing air-handling unit which supplies a canteen. The decentralised system is too small for covering the actual loads and affects the outlet temperature for about 1 – 1.5 K. Although installed in a real application, the system is merely planned as a test plant. Investigation of this system is focussed on providing the greatest number of operating hours in order to analyse different operating combinations for identifying the system’s performance within the district heating configuration. The decentralised TDC unit is examined under base load conditions as peak demands are covered by the existing system.

Monitoring results of one cooling period are obtained and show constant operating conditions due to the coupling of the thermally driven chiller to the district heating grid. 16 test runs were performed with four different set driving temperatures (65 – 95 °C) and four different set cooling temperatures (10 – 16 °C). Due to the working fluid water/glycol in three circuits and a lower volume flow in the heat rejection circuit a lower power and efficiency of the TDC is evaluated. Further examination is focused on finding optimised operating conditions and drawing conclusions on the beneficial combination of district heat, cooling and electricity generation.

References

COMPARISON OF THE HEAT TRANSFER CHARACTERISTIC OF TWO ADSORPTION HEAT EXCHANGER CONCEPTS

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Abstract

The design of the adsorption heat exchanger is a key issue in the development of adsorption heat pumps and chillers. The aim is to increase power densities without decreasing the COP of the machine.

This paper will discuss some aspects to evaluate different adsorption heat exchangers. As an example, a tube-fin and a plate type heat exchanger coated with adsorbent will be investigated in simulation studies using the finite element method (COMSOL Multiphysics).

Figure 1 shows the geometry of the two concepts. Regarding symmetry, a 2d element is investigated. The details implemented are highlighted grey in the respective geometry.

Figure 1: Elements of a tube-fin (a) and a plate type (b) adsorption heat exchanger

In the tube-fin heat exchanger, the adsorbent is placed on the fins whereas the plate type concept uses a porous metal structure sintered onto the heat exchanger plate as a carrier for the active material. Dimensions are chosen in a range that can be realized in the production of full size heat exchangers.

Adsorption equilibrium data from measurements on a zeolite type with suitable adsorption properties for the given application is implemented using Dubinin’s adsorption model [Dubinin, 1975]. Simulation and experimental results on small composite samples with ideal heat removal ([Van Heyden, 2008], [Schnabel, 2008], [Füldner, 2008]) will be transferred to realistic geometries. The work presented here looks at the heat transfer within the geometry of the two heat exchanger concepts, applies criteria like mass and volumetric ratio of adsorbent, and discusses them together with the outcome of the
simulations. The fluid flow and pressure drop of the heat transfer medium inside the channels will not yet be investigated within this framework.

The results show that within the geometries assumed, the plate type adsorption heat exchanger reaches a better adsorbent to metal mass ratio while the mean cooling power per liter of adsorption heat exchanger is considerably higher during the isobaric adsorption stage (see Figure 2).

![Figure 2: Cooling power per volume of adsorption heat exchanger](image)

**References**


EVALUATION OF SORPTION MATERIALS FOR THE APPLICATION IN AN EVAPORATIVELY COOLED SORPTIVE HEAT EXCHANGER

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Abstract

A concept of a novel desiccant evaporative cooling cycle for the application in residential buildings and small offices was developed [1]. Currently a ventilation unit for air dehumidification and cooling is developed which realizes this novel concept. The scheme of the system is given in Figure 1.

The system consists of two sorptive coated heat exchangers (ECOS – Evaporatively Cooled SOrptive heat exchanger) which are alternating between adsorption and desorption mode, providing a nearly continuous air flow to the building. The core component of this novel cycle is a plate heat exchanger of which the primary side is covered with a sorptive coating. Ambient air is dehumidified by adsorption of water vapour onto this desiccant coating. The walls of the primary channels are in thermal contact with the secondary cooling channels which are passed by building return air and into which water is introduced. Due to the evaporation of water in the cooling channels the adsorption heat is transferred to the cooling air side, leading to a temperature decrease of the sorption material and the process air. The lower mean temperature of the sorption material which can be reached due to simultaneous cooling and adsorption shifts the sorption equilibrium towards higher loads. Thereby, this concept enables an enhanced dehumidification of ambient air. The relevant operating points differ from a standard adiabatic DEC-system and the choice of sorption material as a performance dominating parameter must be evaluated for these conditions.

Figure 1: Schematic of the ECOS system: Two sorptive heat exchangers are operated in parallel. The upper heat exchanger is regenerated with heated ambient air while fresh ambient air is dehumidified and cooled in the primary channels of the lower heat exchanger. Building return air passes the cooling channels of the lower sorptive heat exchanger in which water evaporation takes place.
The paper evaluates the influence of two different sorption materials - one silicagel and one zeolite - on the performance of a cooled sorptive cross-flow heat exchanger. A first evaluation of these sorptive materials for their applicability in a cooled sorptive heat exchanger is shown with respect to equilibrium sorption data described according to Dubinin’s theory of volume filling [2]. The dependence of the maximum loading spread on the curve shape in the operating window of Dubinin’s characteristic curve typical for open sorption systems is discussed. A dynamic model of a sorptive heat exchanger was implemented in the modelling language Modelica [3]. With the help of this model the dynamic behaviour of the different sorption materials in this respective implementation and their influence on the key performance parameters COP and removed moisture and in particular the operating parameter cycle time is studied.

References


Abstract

In a building project the decision about the selection of the energy supply system has to be taken in an early phase. At the same time of the decision making process often only little information is available about the detailed design and use of the building. For this purpose the planner needs simple, but sufficiently accurate tools which help to provide some first indications about the viability of a specific technical solution. In particular systems using combined heat and power (CHP) and even more systems using a thermally driven chiller which is coupled to the CHP unit, i.e. combined heat, power and cooling (CHCP) systems, are complex and normally linked to higher investment costs but lower operation cost. Also systems using solar thermal energy in combination with thermally driven chillers (TDC) for building air-conditioning are complex but require early decision making, particularly when they shall be installed in new buildings. For that reason it is important to develop tools that allow a solid decision on a certain technical solution in a building design phase based on minimal input information.

In the course of the project PolySMART “Polygeneration with advanced small and medium scale thermally driven air-conditioning and refrigeration technology” /1/ which is carried out in the 6th Framework Programme of the European Union we have developed a fast pre-design method for CHCP systems. In Task 38 “Solar Air-Conditioning and Refrigeration” /2/, an international collaborative project carried out within the IEA Solar Heating and Cooling Programme, we have contributed to this pre-design method systems using solar thermal energy in combination with thermally driven chillers.

The fast pre-design method involves the following major steps:

Calculation of an annual load file (hourly values) based on meteorological data of the site of the building project. For this purpose a special software, the so called “Load Generator” has been developed which produces an annual load file (heating, cooling, electricity, hot water) based on minimal required input information about the building.

Computation of an annual energy balance of the selected overall system. Systems which can be modeled are:

CHCP systems which may include a backup boiler and a backup vapour compression chiller

Thermally driven chillers in combination with a district heating network

Solar cooling systems composed of a solar thermal collector and a thermally driven chiller

Conventional reference system composed of boiler (e.g. natural gas) for heating and hot water and vapour compression chiller for cooling.

Sizing of the key components (e.g. CHP, TDC, solar collector)

Economic comparison of selected solutions over the whole life cycle.

As a result of the method the user will have a better basis for decision making. However, the fast pre-design method does not replace a subsequent detailed design of the system and sizing of each single component.
A general scheme about the full approach is shown in Figure 1. In our paper we will describe the method and present a virtual case study of a hotel at two different sites (central Europe, southern Europe). The example application of the method provides a comparison of three different designs of the energy supply system of this hotel building, namely a conventional solution is compared to a solar assisted air-conditioning system and to a CHCP system in terms of primary energy consumption, CO₂ emission and total annual cost.

Figure 1: Schematic flow chart of the pre-design method

References

/1/ for more information see www.polysmart.org

/2/ for more information see www.iea-shc.org/task38/index.html
Abstract

The use of water adsorption in porous adsorbents is a promising technology for heat transformation in thermally driven heat pumps or chillers. The goal in optimizing such machines should be the realization of high power densities with acceptable COP values.

This paper will present samples, consisting of an adsorbent layer fixed on a metal surface like it has to be done for coated adsorbers. The material used is a compact foil consisting of zeolite crystallites embedded in a polymer (UOP) which has been identified as a promising new adsorbent for use in heat transformation applications driven by low temperature heat [Dawoud, 2007].

The samples schematically shown in Figure 1 have been characterized by kinetic measurements with a constant volume test rig. The results of the test rig are time related uptake measurements, showing a combined heat and mass transfer rate. Some results have already been presented at ISHPC 2008 [Schnabel, 2008].

Additionally, adsorption equilibrium measurements with a thermogravimetric balance on these composite layer samples with a size of 50x50 mm² will be presented. To find the limiting factors for the adsorption kinetics and to optimize such samples with respect to e.g. layer thickness, a one dimensional finite difference model of the coupled heat and mass transfer has been set up with COMSOL Multiphysics using Dubinin’s adsorption model [Dubinin, 1975] to include the layer’s adsorption equilibrium properties [Füldner, 2008].

Simulations are validated by transient measurement data of one sample. Then, the model is used to show the cyclic behaviour of the sample under realistic operating conditions. By maximising the product of COP and volume specific cooling power (VSCP), layers are optimised with respect to layer thickness for realistic geometric assumptions of an actual adsorber, leading to a high volume specific power density with COP values greater 0.6.
References


THE MARKET POTENTIAL OF MICRO-CHCP IN EUROPE: OVERVIEW AND SELECTED CASE STUDIES

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Abstract

Micro-CHCP can become a viable alternative to separate production of heat, cool and electricity under both economical and environmental perspectives. Within the PolySMART project, a market study in different European countries has been carried out following a two-step approach. The first step consisted in defining the conditions necessary for the energy and economic viability and analyzing how they are met in different countries. The second step concerned an in-depth analysis of selected case studies, for which detailed energy and financial simulations have been performed. In the simulations, several key aspects have been taken into account: the time varying heat, cool and power load; feed-in and purchase electricity tariffs; escalation of energy prices; efficiency of plant at conditions different than nominal (ambient conditions and part load); thermal losses from hot and chilled water storages. Two financial case studies are presented: (i) Hotels in Italy and (ii) Offices in Germany.

Figure 1: Payback time (years) in dependence of "hotel category/size" and "macro region/business type" for the three scenarios "business as usual", "moderate support" and "strong support".

Hotels in Italy have been grouped into clusters defined as a combination of category-size classes (3 stars 24 rooms, 3 stars 48 rooms, 4 stars 56 rooms and 4 stars 112 rooms) and four macro region - business type classes (North-Business, Centre-Business, Centre-Tourism and South-Tourism). Three different scenarios have been tested: business as usual, moderate support and strong support. In the moderate support scenario, the electricity feed-in tariff is set to 80% of the purchase tariff and escalation of gas price is set to 5% (electricity price is assumed to vary as a function of natural gas price according to a correlation equation). The strong support scenario differs from the
moderate support in that the electricity feed-in tariff is set to 100% of the purchase tariff and a capital subsidy of 19% of the total investment is granted. The best financial performance is achieved when only a certain fraction of the yearly heating and cooling demand is met in CHCP mode, the remaining part being covered by conventional auxiliaries. Payback time is more attractive in large hotels (see Figure 1), although the share of heating and cooling demand covered by the CHCP plant is higher in medium sized (about 50 rooms) 3 and 4 stars hotels.

Figure 2: Benefits / surplus investment ratio in dependence on annual gas and electricity price escalation for representative German office buildings

Results are shown for offices of 2.000, 4.000, 8.000 und 16.000 m² useful areas (see Figure 2), and for the best performing µ-CHCP system over a live time of 15 years. The results for an annual electricity augmentation rate of 2% and an annual gas price augmentation rate of 5% are marked. The pie-charts show the CHCP system’s coverage of heat (including heat for TDC operation) and cool demand. The bar graphs show the amount of electricity produced by the CHP, self used inside the office building and sold to the grid respectively. Values on the y-axis above 1 indicate an economically viable solution and values below 1 a non-viable solution.

In the selected cases, the economical potential is strongly affected by the coverage of heat and cool demand and by the share of generated electricity sold to the grid. The higher the heating and cooling coverage (i.e. small hotels and small offices), the higher the share of electricity exported to the grid and the lower the economical potential. On the contrary, a better potential is likely to be found when the share of heat and cool coverage is approximately 50%: this is due mainly to a higher number of operation hours of the CHCP systems and partly to a lower share of electricity exported to the grid. The potential is also influenced by the political and economic scenario: capital subsidy seems to have the higher positive effect on profitability in the Italian case and the sensitivity to different escalation rates for gas and electricity prices is huge for Germany. Escalation of gas price less than 5% p.a. and electricity higher than 2% p.a. is the minimum condition for positive profitability in large offices. Either larger gas price escalation rates or lower electricity price escalation rates would determine poor economical results.
Abstract

The goal of this work is to suggest, design and develop new heat pipe heat exchangers to increase fuel cells efficiency. Besides this main category of heat pipe application in fuel cells thermal control there are possibilities to apply heat pipes in ancillary systems as fuel cartridges thermal control and systems for fuel cells heat recovery (co-generation and tri-generation). Heat pipes for fuel cell thermal management ought to have high effective thermal conductivity and be insensitive to the gravity forces. The vacant porous media for such micro/mini heat pipes is a metal sintered powder wick or a silicon/carbon porous wafer with biporous (micro/macro pores) composition, saturated with working fluid. Actually it is almost proved that the ultimate solution for the global energy shortage and greenhouse problems is realization of “Hydrogen economy” in which the energy sources are renewable and energy distribution networks are electricity network and hydrogen network, which run in cooperative way, Winter Carl-Janchen, 2005. The hydrogen network with fuel cells will replace the current oil network for fuelling the transport vehicles. Both stationary power generations, heating and cooling, as well as fuelling transportation systems will count for about 40%, 30% and 20% of energy respectively. The problem of fuel cells thermal control is the key element to increase fuels cells efficiency. In this paper we would like to show, that fuel cells thermal management can be efficiently performed using heat pipes of different types. For micro/mini fuel cells thermal control so called “Micro heat pipe” phenomena is efficient to consider, Vasiliev L. et al, 2007. “Micro heat pipe” (MHP) phenomena is often available in nature. For example there is an analogy between MHP operation and functioning of a sweat gland, Dunn P.D. and Reay, 1976. Open – type MHPs were considered by Reutskii V.G. and Vasiliev L.L 1981, Vasiliev L.L.,1993 as a system of thermal control of biological objects and drying technology. The MHP concept is interesting to realize for micro/mini fuel cells, that includes a fuel cell stack and ancillary systems. Heat pipes fuel cell thermal management can be performed as [Faghri and Guo, 2005; Vasiliev L.L. et al, 2008]:

1) Micro/mini heat pipes for micro/mini fuel cells (< 10 W)
2) Heat pipes for medium fuel cells (10 – 100 W)
3) Heat pipes for portable fuel cells (> 100 W)
4) Heat pipes systems for stationary fuel cells (stationary electricity generation)

Besides this category of heat pipe application in fuel cells thermal control there are possibilities to apply heat pipes in ancillary systems as:

1) Fuel cartridges thermal control
2) Systems for fuel cells heat recovery (co-generation and tri-generation)

Potential applications of heat pipes for fuel cells thermal management includes also systems of wasted heat recovery and fuel cartridges. Thermal link between fuel cell stack and fuel cartridge, or between fuel cell stack and energy recovery system (heat pump) can be also efficiently performed by heat pipe heat exchangers. Sorption heat pumping and cold production can be also performed efficiently by heat pipe heat exchangers. Sorption heat pumps efficiency depends on the heat transfer enhancement in the sorbent bed, the time of the cycle and the sorption capacity of the sorbent material. The cycle based on sorption heat pipes is new (sorption heat pipes - the important component of new heat pumps and coolers), environmentally friendly and economically competitive alternative to conventional heat pumps. They have short cycles and can be powered by low grade waste heat. New power sources efficiency (cogeneneration and tri-generation systems, fuel cells, photovoltaic systems) have a good perspective to be increased, if sorption heat pipes are applied.

Here three directions of sorption heat pumping are considering: 1. the use of waste gases energy (120-300 °C) for air conditioning; 2. tri-generation of energy (solid sorption heat pump + fuel cells, or internal/external combustion engines); 3. solar energy application for air conditioning and ice production.

The sorption heat pipe (SHP) combines the enhanced heat and mass transfer in conventional heat pipes with sorption phenomena of a sorbtion material. The original design of such a sorption heat pipe – cooler was patented in (Vasiliev, Bogdanov,1992), Figure 1.
Figure 1. Sorption heat pipe 1 – vapor channel; 2 – sorption structure; 3 – finned surface of heat pipe evaporator/condenser; 4 – porous wick; 5 – porous valve; 6 – low temperature evaporator with porous wick; 7 – working fluid; 8 – cold box with thermal insulation.

Two modifications of sorption heat pipes we used in experiments: 1. sorption heat pipes in the systems of gas/gas; 2. loop sorption heat pipes in the systems of gas/liquid and liquid/liquid. The adsorbents like activated carbon, zeolite, silica gel, and alumina are physical adsorbents. Usually they have highly porous structures with high surface-volume ratios in the order of several hundreds that can selectively catch and hold working fluid. When saturated, they can be regenerated simply by being heated. Heat pipe heat exchangers are the good tool to perform the procedure of intense heating/cooling of heat pump containers, the latter can also be smaller; the devices contribute to a higher COP. Some attempts to apply heat pipes to stimulate the heat and mass transfer in sorbent materials were performed in the past (Vasiliev, Bogdanov, 1992, Rockenfeller, 1994, Wagner et al., 1996). In sorption heat pipes the adsorbent and working fluid are contained in the same vessel, the latter can also be smaller; the devices contribute to a higher COP. Some attempts to apply heat pipes to stimulate the heat and mass transfer in sorbent materials were performed in the past (Vasiliev, Bogdanov, 1992, Rockenfeller, 1994, Wagner et al., 1996). In sorption heat pipes the adsorbent and working fluid are contained in the same vessel, the adsorbent would maintain the pressure by adsorbing the evaporating fluid. The process is intermittent because the adsorbent must be regenerated when it is saturated. For this reason, multiple adsorbent beds (multiple sorbent heat pipes) are required for continuous operation. The multi sorbent heat pipes system schematic is shown on Figure 2. With the implementation of heat pipes and thermosyphons cycles can be managed easily and can be very rapid.

Figure 2. Sorption heat pump with heat pipes. 1, 2 – adsorber; 3, 4 – sorption heat pipes; 5 – gas flame; 6 – boiler; 7 – fan; 8, 9 – gas distributors; 10-11 – reversing valves; 12 – expansion valve; 13 – condenser; 14 – evaporator; 15-18 – flow valves; 19-22 – water flow inlet and outlet.

Loop sorption heat pipe as electronic components cooler was published in (L. Vasiliev, L. Vasiliev Jr., 2005). Loop sorption heat pipes are insensitive to some “g” acceleration and can be suggested for space and ground application (for example fuel cells thermal management).

References
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STUDY OF THERMAL CONDUCTIVITY, PERMEABILITY AND ADSORPTION PERFORMANCE OF SOLIDIFIED COMPOSITE ACTIVATED CARBON ADSORBENT FOR REFRIGERATION

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Abstract

As a type of physical adsorbent for refrigeration, activated carbon (AC) has the advantages of high mass transfer performance, stable adsorption performance, and no corrosion to the metal material if compared with the chemical adsorbents such as chlorides. The disadvantage of AC is low adsorption quantity, which is only 1/4 of that of calcium chloride. The adsorption refrigeration performance of adsorbent mainly related with the cycle adsorption rate, which is the value of cycle time divided by cycle adsorption quantity, thus for AC the high efficient adsorption performance only can be obtained for the condition of short cycle time, which is mainly related with the heat and mass transfer performance. The solidified adsorbents of AC with high heat transfer performances are developed by the mix of AC with chemical binding materials (Tamainot-Telto and Critoph, 2001, Wang et al, 2003). For such a method the heat conductivity could be improved a lot, but the mass transfer performances will be influenced. In order to develop a type of composited solidified adsorbent with better heat transfer performance and mass transfer performance, the simple composite solidified adsorbent of AC and expanded nature graphite is developed.

Developments of solidified blocks

Before develop the solidified adsorbent, the density of granular AC is tested, and the value is 396kg/m³. In order to keep the density of activated carbon over than 350 kg/m³ in solidified adsorbent while the mass ratio of composite adsorbent between AC and ENG is 1:1, the density of solidified adsorbent in the experiments is between 700–720kg/m³. For the production of the solidified composite adsorbent, firstly the AC and ENG are simply mixed, and then the adsorbent is pressed, and solidified. The ratio between AC and ENG is less, the easier for the adsorbent to be solidified. The adsorbent with different ratios are shown in Figure 1. The surface of the adsorbent is very smooth and there are no cracks on the adsorbent when the ratio of AC is less (Figure 1a). When the ratio of AC increases the numbers of cracks increases (Figure 1b and 1c). The largest ratio between AC and ENG is 2:1, otherwise the adsorbent cannot be solidified (Figure 1d).

Figure 1 The solidified composite adsorbent with different ratios between AC and ENG, (a) 1:2, (b) 1:1, (c) 2:1, (d): 2.5:1

Results of thermal conductivities research

The thermal conductivity is tested by the guard-hot plate method (Tamainot-Telto and Critoph, 2001). Results show that the heat conductivity increases when the ratio of ENG in the adsorbent increases. The density of AC decreases when the ratio of ENG in the adsorbent increases. The increase of heat conductivity is helpful for the adsorption process, and the decrease of the density of AC in the compound adsorbent will influence the volume adsorption quantity. The highest heat conductivity of carbon composite adsorbent is about 2.5W/(mK), which improved the value of
granular carbon by about ten times. The solidified composite adsorbent has similar density with granular AC is analyzed, which is sample 3 with the density of AC about 350 W/(mK), and it has the heat conductivity of 1.8 W/(mK), which also has been much improved a lot if compared with the value of granular AC.

Results of permeability experiments

For the adsorbents with cracks on the surface, the permeability test results will be influenced by the gas transfer through cracks. Thus only two types of adsorbents that don’t have cracks on the surface are tested, and the results are shown in table 1. Table 1 shows that the permeability increases while the ratio of AC inside the composite adsorbent decreases, and the value is improved about 100 times if compared with the solidified carbon adsorbent that solidified by chemical bonding materials(Tamainot-Telto and Critoph, 2001).

<table>
<thead>
<tr>
<th>Sample</th>
<th>Ratio between AC and ENG</th>
<th>Density of AC (kg/m³)</th>
<th>Permeability (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sample 4</td>
<td>1:2</td>
<td>232.6</td>
<td>4.378 × 10⁻¹²</td>
</tr>
<tr>
<td>Sample 5</td>
<td>1:1.5</td>
<td>286.1</td>
<td>5.017 × 10⁻¹²</td>
</tr>
</tbody>
</table>

Results of adsorption performances tests

The adsorption performances results are shown in Fig.2. Fig.2a shows that when the evaporating temperature is lower than -10°C, the granular AC has the best adsorption performance. It is mainly related with the better mass transfer performances of granular AC and the critical mass transfer performance of ammonia for the low saturated pressure of refrigerant. The performances of solidified adsorbent improve fast when the evaporating temperature increases. For the evaporating temperature of -7°C (Figure 2b), the adsorption performances of two types of solidified adsorbents are similar with that of granular adsorbent. For the evaporating temperature of 8°C (Figure 2c), The value of solidified sample 3 is improved by 29% compared with that of granular AC. It is mainly because of the better heat transfer performance of sample 3.

Fig.2 The adsorption rate vs. time, (a) -14°C; (b) -7°C; (c) 8°C

References


Abstract

The flow in a straight rotating heated pipe is an important feature in engineering. Such rotating passages are used in cooling systems of gas turbine blades or in cooling devices for electric generators and rotor drums.

Barua, 1954 [1] studied the flow in a straight rotating pipe with circular cross-section and showed that the rotation generates a secondary flow and the fluid particles move in spirals relative to the pipe. Barua also concluded that the flow rate, compared to the flow rate in a stationary pipe, decreases. Benton, 1956 [2] considered small rotational velocities and constructed a small perturbation expansion about the Hagen-Poiseuille flow. Later, Benton and Boyer, 1966 [3] analyzed the laminar flow in a rotating straight pipe considering high rotational speeds and law Reynolds numbers. The first experiments concerning a rotating pipe were conducted by Trefethen, 1957 [4]. He observed that rotation transfers the onset of turbulence to higher Reynolds numbers. He also found that the laminar regime can be correlated by the parameter $R_e R_\alpha$, where $R_e = \frac{W_m \alpha^2}{\nu^2}$ and $R_\alpha = \frac{\Omega \alpha^2}{\nu}$ are the Reynolds numbers of the axial flow and the rotating flow respectively and $W_m$ is the mean velocity, $\alpha$ is the radius of the pipe, $\nu$ is the kinematic viscosity and $\Omega$ is the angular velocity of rotation.

In the present work we study the flow of a viscous incompressible in a straight rotating heated pipe. We show that the flow depends on two parameters, the Reynolds number due to rotation $R_e$ and the Rayleigh number $R_\alpha$ based on temperature gradient along the pipe wall. Analytic solutions are derived in power series of $R_e$ and $R_\alpha$ respectively.

Figure 1: Secondary flow pattern and isovelocity contours
In Figure 1 it is shown that rotation and heat modify the flow properties and the rate of heat transfer. It is shown that each force generates its own secondary flow and for each flow the fluid particles rotate in the same sense.

Finally, we calculate certain parameters that depend on the simultaneous action of rotation and heat on the fluid, such as the rate of discharge, the friction factor and the Nusselt number.

References

NOVEL SORPTION GENERATOR FOR HEAT PUMP AND REFRIGERATION APPLICATIONS

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Abstract

There is considerable interest in the potential of heat driven refrigeration or heat pump systems to reduce the CO₂ emissions associated with heating or cooling and adsorption systems offer one way of achieving these aims. A prototype of a compact adsorption generator using the activated carbon-ammonia pair and based on a plate heat exchanger concept has been designed and built. The novel generator has low thermal mass and good heat transfer. The heat exchanger uses Nickel brazed shims and spacers to create adsorbent layers (4 mm to 12 mm thick) between pairs of liquid flow channels of very low thermal mass. The prototype sorption generator manufactured (Figure 1) has an overall generator UA value of 380 W/K while driven by water as the thermal fluid and the cycling period is 3 minutes. This paper presents experimental results of three prototype machines:

- Car air conditioning system driven with waste heat from the engine coolant water (at 90°C): The early lab demonstration unit has produced an average cooling power of 1.6 kW and a COP of about 0.23 under the EU typical normal conditions (33°C and 20°C for ambient and cabin temperatures, respectively).
- Gas fired heat pump: This demonstration unit is designed for domestic hot water production and space heating. The machine could deliver hot water at a flow rate of 10 litre/min with a temperature rise of 30°C and a nominal heating power of 7 kW (conditions suitable for the majority of homes in UK). Preliminary test are ongoing.
- Solar driven mobile container for food conservation: This demonstration solar cooling unit is designed to produced 2 kW cooling with 10 m² of solar thermal collectors operating in a desert environment (40°C and -10°C for condensing and evaporating temperatures, respectively). Preliminary testing will be carried out at Warwick but field tests will follow at a desert site.

The construction of this prototype is still ongoing.

Figure 1: Prototype generator
INVESTIGATION OF ACTIVATED CARBON-R723 PAIR FOR SORPTION GENERATOR

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Abstract

The refrigerant R723 is an azeotropic mixture of 40% Dimethyl Ether (DME) and 60% Ammonia. Copper and copper alloys are prohibited for use with pure ammonia, however R723 appears to be compatible with refrigeration copper tube and some copper alloys such as CuNi10. Although R723 is more flammable than ammonia due to the high flammability of the DME component of the mixture, it has similar GWP (zero) and ODP (zero). The prospect of manufacturing a sorption generator with copper instead of stainless steel has prompted an investigation into the activated carbon-R723 pair. This paper presents the preliminary experimental tests aimed to demonstrate the potential sorption properties of the carbon-R723 pair. The photograph of the experimental set-up used for the purpose is presented in Figure 1. The experimental procedure consists of weighing the mass of adsorbate gas $M_a$ within the sample (from weigh difference between unloaded and loaded generator with refrigerant) when at ambient temperature (typical value: 20°C) and with various refrigerant saturation temperatures ranging from -40°C and 20°C. The R723 concentration within the sample $x$ is then calculated from the sample mass itself ($M_c$):

$$x = \frac{M_a}{M_c}$$

where: $x$ is the R723 refrigerant concentration (kg refrigerant/kg Carbon); $M_a$ is mass of adsorbate gas within the sample (kg) and $M_c$ is the sample mass (kg).

The experimental results show that this refrigerant remains stable and balanced throughout the sorption process. R723 concentration is 15% to 30% higher than the pure ammonia concentration at the same operating conditions and this is further evidence that the Ammonia-DME mixture is adsorbed by the activated carbon.

Figure 1: Photograph of experimental set-up.
A R744 TRANSCRITICAL SYSTEM WITH HEAT RECOVERY FOR A
SUPERMARKET – A CASE STUDY

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ABSTRACT

CO₂ is a suitable natural refrigerant for food refrigeration in supermarkets. In the UK, using R744 as a refrigerant is a relatively new concept accounting a small number of plants in the supermarket sector. However, the UK government has stated that “continued technological developments will mean that HFCs may eventually be able to be replaced in applications where they are presently used; the scope is to improve energy efficiency of the system and use alternative refrigerants such as natural refrigerants where applicable” [DEFRA, 2006].

In this paper, the energy consumption and CO₂e emissions of an existing monitored supermarket are described. A transcritical R744 system has been designed to meet the cooling capacities of the store and innovatively uses its recovered heat for other applications. A model has been developed using the store’s data, the characteristics of the R744 system and the proposed utilisation of the heat recovered. The environmental and economic impacts of the store with this innovative refrigeration system are compared to the current installation.

Energy consumption in a traditional supermarket

A supermarket located in South Wales with 5,600 m² of sales area was chosen for the case study. This store was fully monitored for twelve months from February 2007 using Automatic Monitoring and Targeting energy software (AM&T). The AM&T is able to perform automatic meter readings and data collection of mains and sub-mains electricity and gas consumption. Figure 1 summarises the electricity usage recorded by the AM&T system for the store, in kWh and as a percentage of the total. The food refrigeration represents 33% of the total electricity use for the store. Using the CO₂e conversion factor of 0.422 and 0.196 kgCO₂/kWh respectively for electricity and gas, the CO₂e emission of the gas consumption represents 11% of the store indirect CO₂e emission against 89% from the electricity consumption. The total annual CO₂e emission of the store is 2,732,098 kg of CO₂e including the declared refrigerant leakage of 252 kg/year [Space Engineering Services, 2008] of R404a for this store. The leakage rate is the actual annual amount of R404a used to refill the refrigeration system which has a global warming potential (GWP) of 3860. 64% of the store’s total global warming impact is from energy use and 36% is due to refrigerant leakage.

LT/MT R744 transcritical system with heat reclaim

R744 transcritical systems do not have high COPs because of the high pressure ratio but it can be improved by using the high discharge temperature for other applications. For instance, the heat reclaimed can be used for top-up or to provide hot water for heating, HWS or to drive Ab/Adsorption chillers. To demonstrate the potential of the heat reclaimed within the supermarket and its impact on the store’s CO₂e emissions, a feasibility study has been performed to examine a R744 system in place of the actual R404a systems. The R744 systems will cover the cooling capacities of the refrigeration systems. The alternative system shown in Figure 2 is an enhanced booster R744 refrigeration system that provides LT cooling for cold room/freezer food cabinets and MT cooling for chilled food cabinets. The system is running transcritically but instead of cooling the gas in a gas cooler, the heat is recovered by two heat exchangers for applications described in this paper. There are three large suction heat exchangers in the system to ensure that no liquid returns to the compressors, to maximise the evaporation effect and to increase the discharge temperature of the transcritical compressors.

The overall COP of the system including the heat reclaim is:

\[
COP = \frac{Q_{MT} + Q_{LT}}{W_{MT} + W_{LT}} \quad [1], \quad COP_{overall} = \frac{Q_{MT} + Q_{LT} + HE_1 + HE_2}{W_{MT} + W_{LT}} \quad [2]
\]
Findings and summary conclusions

The design of an LT/MT enhanced R744 transcritical system with heat reclaim has been proved to be more efficient than the same system without heat reclaim, with a COP comparison of 4.7 against 1.8. The energy saved of the store by using the heat reclaimed from the R744 system. There are potential savings of 20% in electricity consumption used by the refrigeration systems to provide air-conditioning, 60% in gas consumption for the heating system by providing low temperature underfloor heating and 90% of hot water services. The R744 system is also slightly more efficient than R404a system and has saved 9% of the refrigeration electricity consumption. 2% of the total store electricity has been saved because of the saving from the HVAC and the food refrigeration. The most significant saving is the total gas consumption that has been reduced by 65%. In CO₂e terms, the emission has fallen from 2,732,098 kgCO₂ to 1,575,699 kgCO₂e, the total store CO₂e saving being 42%.

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THERMO-ECONOMIC ANALYSIS OF MICRO ENERGY SYSTEMS FOR APPLICATIONS IN INDIAN RURAL DEVELOPING VILLAGES

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Abstract

This conference paper provides a comparison of two energy systems for rural developing villages under different loading profiles, allowing for analysis of the two systems under different conditions. The loading profiles are based on notional values for 40 family rural villages in Bihar, India. An evening household electricity demand is estimated, with the rest of the energy demand depending on what is being produced or caught in the village. This includes:

Case 1: tomatoes and brinjal, wheat and rice.
Case 2: potatoes and onions, wheat and rice.
Case 3: fish.

System one includes an engine and generator powering a vapour compression chiller, and system two includes an engine and generator supplying heat to a carbon ammonia chiller. Biomass is burnt in a

Description of the model

The wheat and rice fields require varying amounts of irrigation over the course of the year. Tomato and brinjal fields required food processing electricity and heat during harvest (to turn them into pulp), and cooling power during and for two weeks after harvest. Potato and onion fields require cooling power (at a lower temperature than the tomatoes and brinjal) during and for up to three months after harvest. The fishing village requires a constant amount of ice, available each morning. These demands can be summarized as follows:

Case 1: steady evening household electricity, varying irrigation electricity, food processing heat and electricity for a short period of time, low power cooling for a short period of time.
Case 2: steady household electricity, varying irrigation electricity, medium power cooling for a long period of time.
Case 2: steady evening household electricity, steady high power cooling.

System one (Figure 1) takes oil from jatropha plants and uses this to run an engine that meets the household and food processing (where present) electricity demands. The electricity also runs a vapour compression chiller that meets the cooling demand. Food processing heat demand is first met by utilizing any heat from the engine jacket and exhaust, or by firing a biomass boiler when more heat is required. A low capacity battery bank is used for electricity storage when necessary. System two (Figure 2) is similar to system one except that the cooling demand is met via a carbon ammonia chiller that is driven from the engine jacket and exhaust heat and a biomass boiler when required.

These systems are modeled using a combination of first principles, well established models, and empirical data. Hourly weather data for a year in Ranchi, India and the demand profiles for a particular case allow the model to simulate the performance of the energy system for a year. Costs are then used along with the energy balance obtained from the model to perform a thermo-economic analysis of the systems.
Findings and summary conclusions

Based on notional costs, Table 2 shows values of cost per unit exergy for the two systems meeting the demands of the different loading regimes. It should be noted that a field trip to Bihar, India, is planned for early 2009 to obtain more accurate model input data. The major trends are apparent from the study are as follows; first, the longer the cooling power is required, the cheaper the overall cost per kWh cooling power becomes, due to the spreading out of the initial capital cost. Second, as the ratio of ‘free’ engine heat to fuelwood heat increases, the relative cost of cooling power decreases. Third, when the heat is used at a time when the chiller is most efficiency (i.e. at lower condensing temperatures), the relative cost of cooling power decreases.

<table>
<thead>
<tr>
<th>chiller type</th>
<th>VAPOUR COMPRESSION</th>
<th>CARBON AMMONIA</th>
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<td>demand</td>
<td>tomato/brinjal</td>
<td>potato/onion</td>
</tr>
<tr>
<td>fish (Rs/kg)</td>
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<td>25.8</td>
</tr>
<tr>
<td>wood (Rs/kg)</td>
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<tr>
<td>elec (Rs/kWh)</td>
<td>12.4</td>
<td>12.2</td>
</tr>
<tr>
<td>coolth (Rs/kWh)</td>
<td>370.0</td>
<td>114.0</td>
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</table>

Table 2: Notional model analysis values
Abstract

Adsorption chillers/heat pumps showed great potential for utilization of low grade waste heat or solar energy. Such applications require particular operating conditions that can be satisfied only through the development of new sorbent materials which possess large sorption capacity and low regeneration temperature [2]. Advanced heat exchanger concepts must also be considered in order to improve the dynamic performance of the adsorption machine [3].

The aim of this paper is to evaluate the real performance of a composite sorbent of water “SWS-8L” which was specifically designed for utilization in adsorption machines driven by low temperature heat [4]. The SWS-8L sorbent belongs to the family of Selective Water Sorbents, which is a new class of composite sorbents based on a chemical active salt confined to pores of a host matrix. SWSs have been synthesised at BIC-RAS (Novosibirsk, Russia) and investigated together with CNR-ITAE [5]. The sorbent was synthesised by impregnating a mesoporous silica gel with an aqueous solution of Calcium Nitrate. The salt content was 45 wt.%, which is a proper concentration to ensure good water sorption capacity without reducing the mass transfer efficiency.

The adsorption properties of the realized sorbent were measured by a CAHN microbalance with a constant pressure unit, which includes an evaporator filled with liquid water thermostatically controlled at fixed temperature. Results of characterization demonstrated that the water sorbent SWS-8L presents the regeneration temperature lower than 80°C and the maximum sorption capacity of 52 wt.%.

Real performance of SWS-8L were measured by testing the sorbent in a single bed adsorption chiller installed at CNR ITAE laboratory (1kW max cooling capacity). With this aim, 437g of sorbent grains 0.25-0.425 mm in size were embedded inside a compact heat exchanger of a finned flat-tube type. This configuration presents the following peculiarities: a) compactness and low weight, as the heat exchanger is made of aluminium; b) good heat transfer properties due to the high heat transfer area; c) high vapour permeability provided by a granular packaging mode. Fig. 1 shows the test facility which consists of a testing station for providing external heating and cooling energy at the required temperature and the 1 kW adsorption chiller itself. The main component of the chiller is the vacuum chamber where the adsorber is placed. The chamber is connected to an evaporator and a condenser by means of pneumatic vacuum valves.

Figure 1 The CNR-ITAE test facility with 1kW single bed adsorption chiller
A data acquisition and control system was specifically realized and interfaced with PC by Labview environment. The system operation is fully-automatic allowing continuous cycling at fixed working conditions.

Fig. 2 reports the experimental cooling COP and Specific Cooling Power vs. cycle time measured for the above described adsorber “SWS-8L embedded into finned flat-tube type heat exchanger”. All experiments were carried out by fixing the same inlet temperatures of the external heating/cooling fluids ($T_{\text{des}} = 95^\circ\text{C}$, $T_{ev} = 15^\circ\text{C}$, $T_{con} = 30^\circ\text{C}$).

![Figure 2 Experimental cooling COP (dot line) and Specific Cooling Power (solid line) measured for the adsorber “SWS-8L embedded into finned flat-tube type heat exchanger”](image)

Very high Specific Cooling Power (390 W/kg of dry sorbent) was measured for the ad/desorption cycle time of 10 min. The correspondent cooling COP was 0.3. As expected, longer cycle times allow an increase of the COP up to 0.4 at the expense of the Specific Cooling Power, that decreases to 190 W/kg. Performance measured are surely attractive, demonstrating that the high water adsorption capacity of the sorbent SWS-8L, joined with the good heat transfer efficiency of the flat-tube type heat exchanger, allow to realize highly performing adsorber to be used in adsorption machines driven by low temperature heat.

Acknowledgments
This work has been partially supported by the CNR - RAS bilateral agreement and RFBR (grant 08-08-00808 and 07-08-13620).

References
SILICAGEL-WATER ADSORPTION COOLING PROTOTYPE SYSTEM FOR MOBILE AIR CONDITIONING

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Abstract

The use of air-conditioning (AC) systems in cars and trucks contributes significantly to the greenhouse gas emissions. Both the EU directive [1] on the use of fluorinated refrigerants for mobile air conditioning (MAC) systems as well as lowering the CO₂ emission targets for the automotive sector force the car manufacturers to look for more sustainable ways for climate control of the vehicle interior. Sorption cooling technology can potentially provide a more sustainable alternative than conventional compression cooling technology. Thermally driven sorption cooling systems can reduce both the direct impact (avoid the use of high GWP refrigerant) and the indirect impact (reduction of the additional fuel consumption) of MAC’s on global warming.

Within the first phase of the EU funded TOPMACS project a lab-scale prototype of a silicagel-water adsorption cooling system was developed and tested in the laboratory [Boer 2008]. The results of these tests indicated that the silicagel water adsorption cooling system can provide sufficient cold output at the required efficiency. It should be able to cool down the passenger compartment by using the available waste heat in the engine coolant to drive the adsorption AC system. This was the starting point for the next development phase, being an on-board silicagel-water adsorption cooling prototype. The goal of this phase is to test and demonstrate the performance in the real application.

System development

A passenger car of the type Fiat Grande Punto was selected as the demonstration vehicle. Due to big size constraints of the components of the adsorption cooling system it was chosen to use the trunk space for locating the prototype. The thermal compressor section consists of two reactors, each filled with 3 kg of silicagel, and a central part that has the refrigerant check valves. The silicagel is put on the fin side of an aluminium tube-fin automotive heat exchanger.

Because of the trunk location in the car an air cooled condenser could not be applied. A water cooled condenser is used. The cold produced in the evaporator is transferred to a chilled water loop. The chilled water loop is connected to an air-cooler for transferring the cold to the air entering the passenger compartment. A design drawing of the prototype system is shown in Figure 1.

![Design drawing of the adsorption cooling system](image-url)

Figure 1: Design drawing of the adsorption cooling system
Laboratory measurements and results

Laboratory measurements were done at ECN to verify the proper operation and thermal performance of the system before to install it on the car. The adsorption cooling system was connected to a heating water circuit, a cooling water circuit and a chilled water circuit. The flow rates and temperatures of these circuits were set according to expected values for the on-board situation, during normalized European driving cycles.

Figure 2 shows the measured chilling power and COP at varying evaporator temperature in the laboratory test under typical conditions of heating at 90°C and cooling at 35°C. More detailed results of the laboratory tests will be described and discussed in the full paper.

The laboratory tests conditions are static, with constant temperatures and flow rates. The on-board tests will in the following phase indicate how the system performs in more dynamic operating conditions with more fluctuating flow rates and temperatures.

At present the prototype system is being installed in the trunk of the car at CRF in Turin, Italy. (Figure 3) The water circuits for heating, cooling and chilling are connected to the prototype. All circuits have flow and temperature sensors to measure the thermal performance of the system. Performance tests in a climatic chamber are due in February 2009.

References


THE EFFECT OF CYCLE BOUNDARY CONDITIONS AND ADSORBENT GRAIN SIZE ON THE WATER SORPTION DYNAMICS IN ADSORPTION HEAT PUMPS

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Abstract

Intelligent choice of adsorbent optimal for transformation of low temperature heat should be based on comprehensive analysis of both thermodynamic and dynamic aspects. Attractive approach to the dynamic characterization of an adsorptive heat transformer (AHT) is based on the idea that it is obligatory to take into account the adsorbent and heat exchanger as an integrated unit and measure dynamic properties of this unit under real operating conditions of AHT. The uptake evolution q(t) uniquely characterizes the dynamics of isobaric stages of AHT. The aim of this study was to measure this function for the simplest configuration of an adsorbent – heat exchanger (A-HE ) unit at various boundary conditions of the cycle. For a basic 3T cycle, the boundary set (T_e, T_c, T_{HS}) is given by the temperatures of the evaporation, condensation and heating, respectively. In this communication we fixed the regeneration temperature T_{HS} at 80°C and consider the four boundary sets (5, 30, 80), (10, 30, 80), (5, 35, 80), (10, 35, 80) that correspond to typical air-conditioning applications driven by low temperature heat. We selected a Fuji silica type RD as adsorbent because exactly this material was used in commercialized adsorption chillers which utilize low-temperature heat (Matsushita et al, 1987).

Experimental procedure

In order to measure the uptake evolution q(t) we used a Large Temperature Jump (LTJ) method (Aristov et al, 2008). It allows measuring the uptake curves for a monolayer of loose adsorbent grains located on a metal holder subjected to a fast temperature jump (from the minimal desorption temperature T_2 to T_{HS}) or drop (from the maximal adsorption temperature T_4 to T_c). The temperatures T_2 and T_4 were determined directly from the cycle diagram plotted for each boundary set (Aristov et al, 2006) (Table 1). The grains of Fuji silica type RD were used as fractions 0.2-0.25, 0.4-0.5, 0.8-0.9, and 1.6-1.8 mm.

Findings and summary conclusions

The most important finding of this work is the fact that for each boundary set and grain size the experimental uptake curves q(t) can be described by an exponential function q(t) = q(0) ± Δq exp(-t/τ) up to 80-90% of the final uptake (Fig. 1A). Hence, for the studied configuration the dynamics of water adsorption during AHT isobaric stages is very simple and can be described by a single characteristic time τ (Table 1). The evolution of the residual 10-20% is slower (Fig. 1B) and can be described by the characteristic times τ_{0.8} and τ_{0.9}. The typical ratio τ_{0.9}/τ_{0.8} is 1.5, hence, the duration of AHT isobaric stages probably should be restricted by the time τ_{0.8}.

For these boundary sets, the desorption is much faster than adsorption (τ_{ads}/τ_{des} ≈ 2.2 - 3.5). This is due to the higher average temperature and pressure during the desorption as well as to the convex shape of the water sorption isobar for Fuji silica RD, that is profitable for desorption (Glaznev et al, 2008).

Knowledge the function q(t) allows the calculation of the instant specific cooling/heating power W consumed/released in the evaporator/condenser during the ad/desorption stages at (Table 1). The maximal (initial) specific power W_{max} can be estimated as W_{max} = h_{fg} Δq/τ, where h_{fg} is the latent heat of water vaporization (2478 kJ/kg). It reaches 15-30 kW/kg that looks very attractively for practice.

The tendency is revealed that the boundary conditions of the AHT cycle have much smaller effect on the adsorption and, especially, desorption dynamics with respect to the grain size (Fig. 1). The ratio τ_{0.8}(1.6-1.8 mm)/τ_{0.8}(0.8-0.9 mm) is close to 4, that indicates the dominant contribution of intragrain diffusion resistance. For smaller grains this ratio is gradually reduced, and very close dy-
namics is observed for grains of 0.2-0.25, 0.4-0.5 mm in size; the process is controlled by the rate of metal cooling/heating.

**Figure 1** A) The uptake curves for runs 58=>80 (solid), 69.1=>80 (bold), 51.8=>80 (dashed) and 62.8=>80°C (dotted). B) Comparision of experimental kinetics data for adsorption run 55.9=>30°C (−) and desorption 51.8=>80°C (◦). Solid lines - exponential approximation with characteristic times 21.8 and 8.4 s.

<table>
<thead>
<tr>
<th>parameter</th>
<th>run</th>
<th>$\Delta w$, g/g</th>
<th>$\tau$, s</th>
<th>$\tau_{0.5}$, s</th>
<th>$\tau_{0.8}$, s</th>
<th>$\tau_{0.9}$, s</th>
<th>$W_{\text{max}}$, kW/kg</th>
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**Table 1** The characteristic sorption time $\tau$, specific uptake change $\Delta q$ and specific powers $W_{\text{max}}$, $W_{0.5}$ and $W_{0.9}$ for various boundary conditions. Fuji silica RD, grain size 0.4-0.5 mm.

**Acknowledgments.** The authors thank the Russian Foundation for Basic Researches (grants 08-08-00808, 08-08-90016) for financial support and Dr. M. Ito for providing us with a Fuji silica type RD.

**References**
EXPERIMENTAL INVESTIGATION OF VAPOR ABSORPTION SYSTEM USING PROPANE / ALKYLATED BENZENE AB300 OIL

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Abstract

Increase in electricity cost and environmental challenges have made heat-powered vapour absorption cooling systems more attractive for both residential and industrial applications. Absorption chillers are now widely utilised in the air-conditioning industry because they can be activated by heat sources such as hot water (solar collectors), steam, and direct fired natural gas, instead of electricity. Absorption cooling systems can also be integrated with combined heat and power (CHP) systems to provide power, heating and cooling for applications such as the food and process industries [Tassou et al, 2008 and Maidment and Posser, 2000]. This integration is known as Trigeneration and as Combined Heating Refrigeration and Power (CHRP). Trigeneration systems can reduce a typical retail store’s Carbon Footprint by up to 25%.

Current absorption cooling systems utilise two refrigerant / absorbent pairs; water / lithium bromide and ammonia / water. While the water / lithium bromide absorption system is commonly used for air conditioning applications, the ammonia / water system can be used for applications where temperatures lower than zero are needed. Water / lithium bromide has the drawbacks of crystallization at high concentration, corrosion to metals and is expensive. The ammonia / water absorption system has the disadvantage of the great miscibility of ammonia and water, which means large rectification columns are required to generate the ammonia, which adds to the complexity of the system and the high temperature demand of the generation process (125-170 °C). Furthermore, ammonia is a toxic refrigerant and has the additional drawbacks of corrosiveness and explosiveness. Research into alternative working pairs included using Trifluoroethane TFE (refrigerant) / N-methyle-2-pyrrolidone (NMP) absorbent [Sawada et al, 1993), Propane (refrigerant) / Mineral oil (absorbent) [Fukuta et al, 2002) and Dimethylethylurea (DMEU, absorbent)/ R32, R125, R134a and R152a (refrigerants) (Jelinek et al, 2008). The refrigerants used by Sawada et al, 1993 and Jelinek et al, 2008 are HFC refrigerants that have a significant global warming potential. On the other hand, Propane is a natural refrigerant that has zero ozone depletion potential and less than 3 global warming potential.

This paper describes the performance of a vapour absorption refrigeration system using Propane (refrigerant) and alkylated benzene (AB300 – refrigeration lubrication oil, absorbent). Preliminary experiments to assess the miscibility of propane in various lubricating oils were conducted using a Parr high pressure reactor vessel. The lubricating oils tested are Shell Clavus oils 32 and 64 and alkylated benzene oils AB150 and AB300. The results indicated that Propane is most miscible in alkylated benzene AB300. Figure 1 shows a schematic diagram of the experimental facility constructed to test the performance of Propane / AB300 vapour absorption system. It consists of a condenser, expansion valve and evaporator (all extracted from a split unit air conditioning system) and absorber, generator, Propane / oil mixture circulating pump and a pressure regulating valve. The system is equipped with heat pipes installed between the evaporator and the generator to transfer the heat of absorption from the absorber to the generator. Experiments at various evaporator, absorber and generator temperatures were conducted to assess the performance of the system. It was found that the coefficient of performance of the system increased with increasing the generator temperature and with decreasing the evaporator temperature (see figure 2). The coefficient of performance obtained is higher than those of water / lithium bromide absorption system as obtained by Florides et al, 2002 and Kececiler et al, 2000.
Figure 1 Schematic diagram of the Vapour Absorption System

Figure 2 The Coefficient of Performance versus generator temperature

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DOUBLE DIFFUSIVE NATURAL CONVECTION IN ENERGY STORAGE UNITS WITH MULTIPLE PORTS

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Introduction

Natural convection in porous medium has been studied extensively in past decades, due to it is frequently encountered in electronic cooling, crystal growth, contaminant transport, and energy storage, and etc (Ingham and Pop 2005). In the present work, double diffusive natural convection in a cavity with three free openings, simulating the multi heat exchanging processes in one cryogenic energy unit (Ding 2005), will be investigated numerically. To the authors’ knowledge, this problem has never been studied. The porous medium considered here is modeled according to the Darcy-Brinkman formulation (Zhao et al. 2008). Additionally, visualization of the heat and solute transports, using streamlines, heatlines and masslines (Zhao et al. 2007, 2008), would be conducted in the present work.

Physical model and problem statements

The physical domain under investigation is a two-dimensional fluid-saturated Darcy-Brinkman porous enclosure (see Fig. 1). The porous matrix is assumed to be uniform and in local thermal and compositional equilibrium with the saturating fluid.

Results

Finite volume method is employed on non-uniform grids for the solution of the present problem. The obtained heatlines and masslines are shown to be a very effective way to visualize the paths followed by heat and solute through this porous partial enclosure.

Broad range of buoyancy ratios, covering solute-dominated opposing flow, thermal-dominated flow and solute-dominated aiding flow, is examined for several different Darcy numbers (Fig. 2). As the permeability the porous medium is decreased, fluid flow, heat and mass transfer tend to be diffusion dominated. The main contributions of decreasing the Darcy number are predicted to be a
flow retardation effect and a suppression of the overall heat and mass transfer in the enclosure.

As Darcy number is low (limiting to Darcy flow), heat and solute are transported to ambient medium through the bottom vent when thermal-dominated flow and solute-dominated aiding flow subsist, otherwise, they are mainly transported toward top and side vents. Whereas, as Darcy number is high (limiting to fluid flow), heat and solute are transported to ambient medium through the top vent when thermal-dominated flow and solute-dominated aiding flow subsist, otherwise, they are mainly transported toward bottom vent. These trends could be intensified as buoyancy ratio increases positively or decreases negatively.

Promoting thermal Rayleigh number could enhance fluid flow and corresponding heat and mass transfer (Fig. 3). As the porous thermal Rayleigh number exceeds 10, natural convective fluid flow is enhanced greatly, simultaneously increasing the heat and mass transfer rates on top and side vents and decreasing those on bottom vent. Heat and mass transfer rates on the bottom vent vary nonlinearly with the thermal Rayleigh numbers.

References

REACTION STUDY OF CHEMICAL HEAT PUMP FOR MEDIUM TEMPERATURE HEAT STORAGE

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Abstract

A chemical heat storage material which can store waste heat at medium-temperature around 200–300°C was newly developed. The performance of developed material was demonstrated in a thermo-balance and packed bed reactor. Waste heats around 200–300°C are emitted as exhaust gases from internal combustion engine, cogeneration, high-temperature process and solar systems. Amount of exhaust gas emission is quite large, and needed to utilize well for energy efficiency improvement. A mixed hydroxide, Mg$_α$Co$_{1-α}$(OH)$_2$, which was mixed with magnesium hydroxide, Mg(OH)$_2$, and cobalt hydroxide, Co(OH)$_2$, was one of candidates of the heat storage materials. The heat output performance of material was examined in a packed bed reactor. It was demonstrated that the chemical material was capable to store heat around 200–300°C, and to output it over 200°C. It was shown that the material was useful for a chemical heat pump for the medium-temperature heat utilization.

Chemical heat pump for medium temperature heat storage

Waste heat recoveries from high-temperature processes are well developed for heat at over 400°C by steam and gas turbines, and also at less 100°C by sensible and latent heat storage technologies. On the other hand, medium-temperature heat at 200–300°C has not been utilized well. Efficient utilization of the medium-temperature heat would be one of important way for an improvement of energy efficiency of high-temperature processes. Vehicle is one of key high-temperature systems. Number of vehicle production is still increasing in the world. Impact of vehicle usage on carbon dioxide emission is an important subject for global warming phenomena. Energy conversion efficiency from fuel to driving work in vehicle is around 20% in enthalpy-base, and rest 80% of energy is emitted as exhaust heats. Although huge efforts have already made for the improvement of engine efficiency for axial output, the efficiency has already attained at almost mechanical upper-limitation. On the other hand, vehicle still has room for energy efficiency improvement by utilization of excess heats emitted as an exhaust gas from a muffler and a heat exchanged air from a radiator. Heat storage of the waste heat as a heat at lower than 100°C is possible by using conventional sensible and latent heat storages, however, from the stand point of heat quality (exergy) recovery, medium-temperature heat storage would be much more effective. On the other hand, heat storage technologies for the temperature range are still insufficient.

Figure 1: 1 Thermal hybrid vehicle system combined with a heat storage function for heat recovery from exhaust gas of an engine.
Medium-temperature heat recovery is important for not only vehicle, but also solar power, fuel cells and high-temperature processes. For heat process in practical use, an influence of instable thermal operations on a reduction of total energy efficiency is not negligible. For a cogeneration engine in practical use, a mismatch between heat output from engine and heat demand generates plenty amount of waste heat to atmosphere. Then, waste heat storage function for medium-temperature heat becomes important for an efficient operation of high-temperature processes. Authors proposed a thermal hybrid vehicle system depicted in Fig. 1, which had heat storage function for heat recovery from exhaust gas of an engine or other high-temperature processes. Chemical heat pump has possibility to establish the thermal hybrid system by chemical heat storage of medium-temperature heats. However, there were small candidates which could be operated at medium-temperature range except magnesium oxide/water (MgO/H2O) chemical heat pump (Kato et al., 1996).

\[
\text{MgO} + \text{H}_2\text{O} \leftrightarrow \text{Mg(OH)}_2 \quad [1]
\]

On the other hand, although MgO/H2O system has high reactivity, the dehydration of the reaction system is needed relatively higher temperature of around 350°C. The authors tried to survey chemical reaction systems for the chemical heat pump thermodynamically. Finally, the authors proposed mixed hydroxide materials contained with magnesium hydroxide and other hydroxides for medium-temperature chemical heat pump (Ryu et al., 2007).

**Packed bed experiment of chemical heat pump**

A mixed hydroxide, Mg_{\alpha}Co_{1-\alpha}(OH)_2, which was mixed with magnesium hydroxide, Mg(OH)_2, and cobalt hydroxide, Co(OH)_2, was one of candidates of the heat storage materials. Reactivity of the developed mixed hydroxide was studied by using a packed bed reactor. An experimental apparatus consisting of the packed bed reactor and a water reservoir was prepared. The reactor made from SUS304 steel has an inner diameter of 11.6 mm and a length of 100 mm. Mg_{0.5}Co_{0.5}(OH)_2 of 10.0 g as reactant was charged into the reactor. The bed temperature was monitored by a thermocouple set in the bed. The hydroxide material, Mg_{0.5}Co_{0.5}(OH)_2, was dehydrated by joule heating using a surrounded ribbon heater. The bed was cooled down to initial bed temperature for hydration after dehydration process. When bed attains stable state at the temperature, the joule heating was turned off and water vapor at targeted pressure was supplied into the reactor, then the change of bed temperature was monitored during hydration period. The mixed hydroxide performance was demonstrated in the packed bed reactor.

**Findings, summary and conclusions**

Mg_{\alpha}Co_{1-\alpha}(OH)_2, which was mixed hydroxide of magnesium hydroxide, Mg(OH)_2, and cobalt hydroxide, Co(OH)_2, was one of candidates of the mixed hydroxide materials for medium-temperature heat storage. The material was dehydrated, that is, capable to store heat at around 200–300°C. Hydration of Mg_{\alpha}Co_{1-\alpha}O which was corresponding to heat output operation was observed over 200°C. It was shown that the material methodology had possibility to widen medium temperature utilization by chemical heat pump, and to realize thermal hybrid system combined with a medium-temperature chemical heat pump and a high-temperature process for efficient energy use.

**References**


A COMPUTATIONALLY FAST MODEL OF ADSORPTION CHILLERS

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Abstract

I propose a highly idealised model for the generator component of adsorption chillers, predicting coefficient of performance and cooling power. Notwithstanding the cyclic, time dependent operation of the generator, the model is intended to run with sufficient speed to be part of a whole plant simulation. (For example, solar driven cooling, where the plant includes a chiller plus solar collectors, thermal storage, and food stores). The model (Fig. 1) implicitly ignores resistances to mass transfer by treating adsorptive capacity (X) as being reached instantaneously - particularly appropriate for high-pressure refrigerants like ammonia.

Key

1, 2) Check valves connecting to condenser and evaporator
3) Isothermal slab representing non-sorbing components (metallic)
4) Interface, thermal conductance $h_i$
5) Non-isothermal slab representing sorbent
6) Specified heating/cooling
7) Heat flux at sorbent boundary, according to $q = h_i (T_3 - T(x=0))$
8) Adiabatic boundary, $dT/dx = 0$

Conduction in region 5 follows,

$$\rho_b \frac{\partial h_s}{\partial t} = -\nabla q + \rho_b h_s' \frac{\partial X}{\partial t}$$  (1)

$h_s$, enthalpy per unit mass sorbent
$h_s'$, specific enthalpy of adsorbing or desorbing refrigerant
$\rho_b$, bulk density of sorbent

Figure 1: Essential aspects of the model

Heat transfer within the bed of sorbent (5) is idealised as principally one-dimensional, time-dependent conduction through an isotropic medium. The non-sorbing parts of the generator (3, walls, fins and tubes) are represented by a single, isothermal lumped mass to which heat is applied, and which is connected to the sorbent via an interface (4) of conductance $h_i$. In equation 1, at each time step pressure was either set constant (isobaric) or chosen to get the expected average $X$ (isostereic, or specified mass recovery).
Predictions required ~ 6 s cpu time per cycle on a 1.6 GHz, single-processor notepad computer. For a basic cycle (Fig. 2) I found (1) cyclic steady-state well before the sixth cycle (2) steep temperature gradients within the sorbent slab. Longer cycle times yielded better coefficient of performance (e.g. COP = 0 at a cycle time of 16 s increasing to COP = 43% at a cycle time of 100 s). There was an optimum cycle time of 56 s with regard to cooling power (0.95 kW per kg carbon). The model was adapted for a two-stage generator, useful for dealing with big temperature lifts. Although heating liquid was “switched off” when the lower stage reached its target pressure ($T_{\text{sat}} = 295$ K), continued heat transfer from the non-sorbing parts forced the generator pressure beyond the target (Fig. 3).

Notes

Sorbent temperatures at $x = 0$ mm, 1.5 mm. Transfer liquid at 308 K, 383 K. The fluid to vessel heat transfer coefficient is 1000 W m$^{-2}$ K$^{-1}$, and the vessel-to-carbon conductance is 200 W m$^{-2}$ K$^{-1}$.

Fig 2: Carbon temperatures during basic cycle

Fig. 3 Pseudo-Clapeyron diagrams for two stage generator.
HYDROTHERMAL AND MICROWAVE SYNTHESIS ON METAL SUBSTRATES OF SAPO ZEOLITES FOR HEAT PUMPING APPLICATIONS.

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Abstract

SAPO zeolites belong to the family of the phosphor based zeolites, meaning that their framework is constituted of alternating Al₂O₃ and P₂O₅ tetrahedra tridimensionally arranged in space, in which some Si substitution in the main network is produced (Lok, 1984). The particular adsorption properties of such zeolites at low equilibrium temperatures and pressures have recently raised particular interest as new, high performing, adsorbing materials for heat pumping applications working by low regeneration temperatures (Jänchen, 2005). In fact, the most diffuse aluminosilicate zeolites, like zeolite 4A, X and Y, despite their elevated water adsorption capacity require working conditions (high temperatures) which could be difficult to obtain, for examples in mobile or in residential applications. The main difference shown by the SAPO zeolite families is the reduced acidity of the active sites in the spatial lattice compared to the more “active” aluminosilicate zeolites, which causes the water molecules to desorb from the lattice sites at lower temperatures (Sastre, 1997). In this work, a study on synthesis and properties of SAPO zeolites having a chabasite framework has been conducted preparing different zeolite formulations by varying main components ratios Al/Si and Si/P and using different organic templates, in an autoclave at low temperatures (< 200 °C). The SAPO syntheses were then replicated in a microwave oven and the zeolite obtained compared with the respective “non-microwave” prepared. Zeolites characterizations and comparison were based on X-ray, SEM, EDX and adsorption curves evaluations (figure 1)

Figure 1: SAPO obtained by usual hydrothermal synthesis (left) and by microwaves (right)

Microwave synthesis has demonstrated a significant time reduction compared to the usual methodology maintaining all the adsorption characteristics of the produced zeolites. This is an important result considering the long synthesis time normally needed to crystallize SAPO zeolites. The adsorption capacity of the most promising zeolites synthesized was measured by termogravimetric technique based on the utilization of a Chan 2000 microbalance. Figure 2 shows the water adsorption isobar (10.6 mbar) measured at equilibrium for a typical sample synthesized.
Result obtained confirmed that the SAPO zeolite presents large adsorption capacity (up to 24 wt.%) and low regeneration temperature (T<100°C).

![Figure 2: SAPO zeolite/water adsorption isobar measured at 10.6 mbar](image)

Finally, a preliminary study on the direct synthesis and growth of SAPO zeolites on metal supports was conducted in both the two synthesis environment, usual hydrothermal and microwave (figure 3). Results demonstrated that properly adapting the synthesis environment, both the two methodologies are feasible, opening new interesting perspectives to the deposition of performing adsorbent coatings on high surface heat exchanger for high performing heat adsorption pumps.

![Figure 3: Microwave deposition of SAPO zeolite on aluminium support](image)

**References**


Abstract

The current German legislation on building energy consumption (EnEV) and the share of renewable energies on the heating market (EEWärmeG at national level, EWärmeG in the federal state of Baden-Württemberg) will contribute to the more widespread use of efficient technologies (e.g. heat pumps) as domestic heat sources. Rising energy costs, increasing requirements in terms of home energy efficiency and greater environmental awareness have pushed the demand for heat pumps. This trend provides an opportunity for gas heat pumps to secure the future of natural gas as a heat energy source. The natural gas-based highly efficient renewable heating system is a technology with a potential to be a trend-setter. The gas industry has teamed up with appliance manufacturers in a joint ‘Gas Heat Pump Initiative’ to further develop this technology to market maturity through practical laboratory tests and field trials.

Market development

Initiatives and innovations were the prerequisites for the market entry of natural gas. The spreading and consolidation of natural gas on the market was based, among other factors, on developments of low-pollution burners, energy-efficient low-temperature boilers, and later condensing boilers.

The role of natural gas as a modern and ecologically compatible energy source (“Blue skies over the Ruhr”) was further extended by the emergence of condensing boiler technologies in the 1990s (Fig. 1).

Figure 1 - Phases in the development of heating technologies

Natural gas was able to displace light fuel oil from its leading position on the heating market. The technical efforts were flanked by corresponding measures addressing the market partners (manufacturers, installation trade, etc.), as well as politics and public relations.

The heating market is the largest sector for the consumption of natural gas. The inherent benefits of the product, in combination with modern, efficient and convenient technologies, have until today raised natural gas to the status of no. 1 energy choice among customers.

The expansion of the gas supply networks is in the meantime reaching the limits of economic feasibility. The daily public coverage of energy and climate policy discussions was in the end a reason for potential customers from the new building segment, and even operators of natural gas systems, to withdraw their favour from this environment-friendly, CO₂-reducing heating technology.

The gas industry must respond to the customers' new environmental awareness and increasing demands for autonomous solutions with innovative and modern heating technologies and with renewable energies. Alongside the established condensing boilers with solar solutions, gas heat pumps, in particular, enable suppliers to meet the market's political demands for highly efficient heating systems in conjunction with a renewable energy source.
Technological development in the gas sector

The challenge now facing the German gas industry, together with its appliance manufacturers, is to develop and offer the customer appropriate technical alternatives under the difficult conditions encountered in competition with other energy fuels (to a large extent renewable energies). In addition to the technologies already available today, such as the combination of condensing boilers with solar water heating and space heating support, the gas heat pump technology can be expected to assume an important role in building heating in the near future.

Arguments in favour of the gas heat pump are:
1. Gas heat pumps display a high energy efficiency and – properly configured – reduce natural gas consumption by up to 30% compared to a condensing boiler (lower operating costs).
2. The increased efficiency comes hand-in-hand with CO₂ reductions and thus also meets climate protection demands.
3. The heat pump technology is able to raise the ambient heat energy to a temperature level suitable for the heating of buildings.
4. Combinations with further renewable energy sources, e.g. solar thermal energy or bio natural gas, promise additional efficiency improvements in conjunction with the gas heat pump technology.

Fig. 2 places the development of appliances and the use of bio natural gas on a timeline. From the point of view of E.ON Ruhrgas, the broad market entry of the gas heat pump (from 2011) could be followed very closely by that of micro-cogeneration. The very promising fuel cell is still farthest away from series production. Bio natural gas, on the other hand, is already now being fed into the gas supply network and is thus already contributing to CO₂ reductions.

Conclusion

Rising energy costs, increasing requirements in terms of home energy efficiency and greater environmental awareness have pushed the demand for heat pumps. Recognition of the necessity for concerted cooperation and a targeted application of resources gave birth to the idea of a “Gas Heat Pump Initiative” (founded in February 2008). The bundled know-how of all these member companies is to be exploited to promote the market maturity of the future-oriented technology “gas heat pump” and to establish a new, innovative gas product which gives due consideration to all public demands. The comprehensive laboratory tests of gas heat pumps are currently being performed at E.ON Ruhrgas in coordination with project partners. Some 250 field trial systems will be ready for installation in the next two years and are expected to permit statements on the efficiency and practical suitability of the newly developed gas heat pumps. The first results appear to confirm the great potential of this technology. The marketing of gas absorption heat pumps with outputs up to 40 kW for larger buildings and multi-family housing (new and existing buildings) is already to begin this year. The wider market availability of gas heat pumps for (new) single-family homes should follow from 2011. Given successful future further development of the corresponding products, the gas heat pump technology will be able to complement present gas appliances, in particular condensing boilers, for the heating market in the medium term.
PERFORMANCE OF A SINGLE VESSEL CHEMICAL HEAT PUMP USING HYSRATION REACTION OF CALCIUM CHLORIDE

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Abstract

The hydration reaction of calcium chloride is suitable to drive a chemical heat pump for refrigeration or air conditioning using low-level thermal energy. Using calcium chloride as a reactant we have been developing a compact chemical heat pump for household cogeneration systems. In this study, the performance and characteristics of our chemical heat pump have been examined based on the experimental results with two prototypes. The main features of this chemical heat pump system are as follows; 1) integrated reactor beds, which combine a plate-fin heat exchanger with the composite reactant of calcium chloride and expanded graphite (Fujioka et al., 2008), are used in order to enhance heat transfer, 2) reactor beds, evaporator and condenser are placed in a single vessel to make the reactor simple and compact, 3) continuous operation is carried out with one reactor vessel.

Reactors

Prototype 1: The structure of the first prototype is shown in figure 1. Four plate-fin heat exchanges (Drawn Cup type, 25 columns) connected to the piping for heat transfer water are set vertically in a square vessel (400W x 300 x 700H). Two of the four heat exchangers are packed with the composite reactant. The other two are left empty to work as condensers. Another same type heat exchanger (10 columns) is placed at the bottom of the vessel to work as an evaporator. The experimental setup consists of the reactor vessel, a hot water tank, boiler, piping and other devices, whose diagram is omitted since it is similar to that for the second prototype shown in figure 2.

Prototype 2: Figure 2 shows the experimental setup with the second prototype. It is designed to improve the problems of the prototype 1 revealed from experiments. The reactor is a cylindrical vessel with an outside diameter of 457mm in which five reactor beds with same structure and dimension as the first prototype are hung from the upper flange and a coiled tube of 16 mm outer diameter and 4.14m length lines up to work as a evaporator/condenser. A smaller inner pod is put on the coiled tube so that the water for hydration is held in the gap between the vessel and the inner pods. This configuration makes it possible that the evaporator/condenser has a large surface area with small amount of water and that the coiled tube positioned at the vessel wall reduces the heat loss.

Results

An example of the test results with the prototype 2 is shown in figure 3. It shows a series of three cycles of operation stating with a dehydration process. Dehydration takes 15 to 20 minutes followed by 10 minutes of hydration process. Each cycle repeated almost the same variation of temperature, pressure and heat duty, showing the stability of the heat pump system. The average cooling COP of the first prototype was 0.34 and that of the second one was 0.60. The experimental results show that further improvement can be obtained by enhancement of the heat transfer outside
the coiled tube. Advantages and disadvantages of the prototypes 1 and 2 are also discussed.

Figure 2: Experimental setup with the prototype 2 reactor and photographs of (a) reactor beds and (b) coiled tube evaporator/condenser and inner pod

Figure 3: An example of the test result ($P_v$: saturated vapor pressure)

References

Abstract

The inlet air cooling helps in increasing the performance of gas turbines. In this paper, a new approach to enhance the performance of gas turbines in hot climates is investigated. One form of the combined cycle is the combined gas turbine and steam turbine with absorption chiller unit, where the waste heat is used to power an absorption chiller to produce cooling for inlet air entering the compressor of a gas turbine on high ambient temperature days. The waste heat from the exhaust gases may be utilized to produce hot water via double-pipe heat exchanger. The hot water produced could then be used to drive a single-effect lithium-bromide absorption chiller machine which in turn could cool the incoming air.

An analysis carried out by taking the weather data of Tripoli (Libya) indicates that reducing the intake ambient temperature below the ISO standard conditions could help to increase the power output.

The performance improvement is calculated for ambient temperature of 47 °C. The results indicated that the intake temperature could be lowered below the ISO standard with power increase up to 32%.

The simulation program IPSEpro has been used to do the modelling and simulation of the combined cycle in this study.

Description of the system

The proposed system, shown schematically in Figure 1, consists of a combination of two separate cycles, one at high temperatures (topping cycle) and the other at relatively low temperatures. The most common combined cycle is the gas-steam combined cycle where a gas-turbine cycle operates at the high-temperature range and a steam-turbine cycle at the low-temperature range. The upper and bottom cycles are shown in Figure 1.

Findings and summary conclusions

A new model for improving the performance of combined gas turbine cycle and eliminates the warm weather power degradation has been developed using IPSEpro program.

Heat powered absorption refrigeration cycles are attracting increasing interest because an absorption refrigeration cycle can be driven by low-temperature heat sources, and may therefore provide a means of converting waste heat into useful refrigeration. Absorption cooling is a technology that allows cooling to be produced from heat, rather than from electricity and does not use harmful refrigerants. The absorption cooling technique demonstrated a higher gain in power output.

The analysis presented in this paper indicates that the absorption refrigeration unit linked with the combined cycles will perform in a satisfactory way. An examination of the gas turbine cycle has indicated that the performance is influenced by the ambient temperature. This study has shown that using the absorption chiller will improve the performance of a gas turbine during the hot ambient days. The inlet air temperature was cooled from 47 °C
to 10 °C and the attained power output and efficiency was 51.62 MW and 52.81% respectively.

Complex energy systems able to provide more than one product recently are the trend to save energy sources and restrain the environment from thermal pollution. Therefore, an economic study on the proposed energy system which will provide a clear insight into its economic validity among other combined systems is strongly recommended.

![Diagram of a combined cycle gas turbine power plant with absorption chiller](image)

**Figure 1:** Single shaft combined cycle gas turbine power plant with absorption chiller

**Reference**

SIMULATION OF WATER AND AIR COOLED LITHIUM-BROMIDE (LiBr) CHILLERS POWERED BY LOW-TEMPERATURE GEOTHERMAL HEAT SOURCE AT ARID-ZONE AREA

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Abstract

Remote communities, in arid zones such as Waddan city, 265 km south of Libyan north coast, could greatly benefit from their available high potential low-grade temperature geothermal sources. According to the literature survey, such sources located in desert areas are very attractive energy sources for absorption cooling and central heating. Lithium-bromide water mixture of single effect or half effect absorption chillers with different schematic configurations have been successfully modeled and simulated by many researchers. However, most of these types of chillers were directly powered by low input mass flow rates of high or low temperature heat sources such as steam or hot water.

Three Lithium Bromide models namely; water-cooled single effect, air-cooled single effect and water-cooled half effect were schematically built and mathematically simulated using the power plant and refrigeration modeling libraries of IPSEpro simulation modeling package. The chillers utilized a high potential artesian hot water supply well (1600 m deep) at a constant high flow rate of 114 kg/s and 73°C. Shallow cool water (25°C) reservoirs surrounding the geothermal source were used for cooling the absorber and the condenser. Weather data analysis has shown that, the average mean maximum dry bulb temperature of Waddan city was 40°C.

The simulated results show that this low-temperature geothermal source can be used as heat source for both half and single effect Lithium bromide water mixture absorption chillers. The cooling capacities and coefficients of performance (COP) were found to be within acceptable published limits. The usable chilled water temperature difference across the evaporators was found within the range of air-conditioning use in hot climate conditions. The comparison between the three different models has indicated that the high cooling capacity was produced by water-cooled half effect chiller without any significant economical benefit of the heat rejected from the desorber. The low cooling capacity was obtained from both air and water-cooled single effect chillers with significant outlet temperatures (65-68°C) which can be further used in central heating and domestic hot water supply.

Modeling summary

Figure 1 shows the schematic diagram for three modeled single/half effect chillers. All were simulated and parametrically studied over allowed water and lithium-bromide thermodynamic limits. The models were stable and validated in accordance to the Dühring Lithium Bromide pressure-temperature diagram (Keith E. Herold et al, 1996). The heat input temperature to the absorber, inlet chilled water temperature to the evaporator and ambient temperature were compared with other cycle parameters such as cooling capacity, COP and heat transfer across the main components of the chillers.
Brief conclusion

The main output results of the three different modeled chillers are clearly listed in table 1. Technically it can be concluded that, if only the cooling air-conditioning system is decided to be the most suitable choice for Wadan communities, without utilizing hot water energy, then water-cooled half effect chiller is probably the best system to be installed, and if air-conditioning along with hot water supply systems are desired, in summer and winter seasons, then the water cooled single effect chiller has to be selected instead.

<table>
<thead>
<tr>
<th>Chiller type</th>
<th>Water-cooled single effect</th>
<th>Air-cooled single effect</th>
<th>Water-cooled half effect</th>
</tr>
</thead>
<tbody>
<tr>
<td>COP</td>
<td>0.825</td>
<td>0.845</td>
<td>0.424</td>
</tr>
<tr>
<td>Cooling capacity (Ton)</td>
<td>564</td>
<td>628</td>
<td>1284</td>
</tr>
<tr>
<td>Desorbers outlet temperature (°C)</td>
<td>65</td>
<td>68</td>
<td>51</td>
</tr>
<tr>
<td>Heat input temperature difference across the desorbers $\Delta T$ (°C)</td>
<td>$(73-65) = 8$</td>
<td>$(73-68) = 5$</td>
<td>$(73-51) = 22$</td>
</tr>
<tr>
<td>Chilled water temperature difference across evaporators $\Delta T$ (°C)</td>
<td>$(12-5) = 7$</td>
<td>$(22-27) = 5$</td>
<td>$(12-5) = 7$</td>
</tr>
</tbody>
</table>

Table 1: Output results of modeled chillers

References

Abstract

Worldwide the energy consumption for cold and air-conditioning is rising rapidly. Usual electrically driven compressor chillers (split-units) have maximal energy consumptions in peak-load period during the summer. In the last few years even in Europe this regularly leads to overloaded electricity grids. The refrigerants that are currently used in the split-units do not have an ozone depletion potential (ODP) anymore, but they have a considerable global warming potential (GWP), because of leakages of the chiller in the area of 5 to 15 % per year. Particularly the sale figures of split-units with a cooling capacity range up to 5 kW are rising rapidly. The Japan Refrigeration and Air Conditioning Industry Association (JRAIA) has expected a worldwide sales of 82.3 million units in 2008. In Europe the number of sold units has risen about 37% from 6.3 million in 2004 to predicted 8.6 million in 2008 (JARN, 2008).

Thermal cooling by solar energy, district heat or waste heat from CHP units, biomass as well as processes could be lead to a considerable reduction of energy consumption. The sorption chillers use environmentally friendly refrigerants and have only very low electricity demand. Therefore the operating costs of these chillers are very low and the CO₂ balance compared to split-units is considerably better. The main advantage of solar cooling is the coincidence of solar irradiation and cooling demand. However, for thermal cooling e.g. by waste heat from a CHP unit the benefit is the longer operating time of the CHP unit itself and with that the increased electricity production. In general the market potential for small-scale solar cooling kits is very large. But so far, only a small number of companies develop and offer standardised small-scale solar cooling kits for the European market up to a 30 kW cooling capacity. At present SolarNext has realized over 20 small-scale systems worldwide (Canada, Europe, China and Australia). Further companies like e.g. Schüco, Solution and Phönix Sonnenwärme also offer solar cooling kits with the different chillers.

Description of the cooling kit

During the last two years SolarNext has developed different chillii® Solar Cooling Kits and Systems respectively based on the chillii® chillers (Table 1) and absorption chillers from the companies EAW and Yazaki. The solar cooling kits basically contain solar thermal collectors with attachment, hot water storage, pump-sets, a chiller, a re-cooler, partly cold water storage and a system controller. The cooling kits are developed for the European market, whereas other re-coolers can be offered according to the country.

For the development of standardized solar/thermal cooling kits it is indispensable to use a system controller for the complete system. The previous solar cooling demonstration and pilot projects are using several single controllers e.g. for the solar thermal system, for the chiller, for the recooler and for the cold or heat distribution, which are together cost intensive and are not always operating optimal together. The alternative was until now an expensive PLC controller which had to be programmed for each single case. Because of that the SolarNext has decided in the year 2007 to develop an own system controller for the whole system, which has an influence from the automotive sector and is cheap and system oriented. The functional range of the chillii® System Controller contains the control of different heat sources, the back-up system, the storage management, the domestic hot water, the chiller and the re-cooling as well as heating and cooling circuits. So the highest system efficiency is reached with the needed energy generation with priority in regenerative energy sources, optimized running of chillers as well as the re-cooling with speed control of the pumps and the re-cooling ventilator.
Table 1 – Small-scale sorption chillers for chillii® solar/thermal cooling kits

<table>
<thead>
<tr>
<th>Product name</th>
<th>chillii® STC8 / chillii® STC15</th>
<th>chillii® ISC10</th>
<th>chillii® PSC12</th>
<th>chillii® WFC18</th>
<th>chillii® ACC50</th>
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</thead>
<tbody>
<tr>
<td>Technology</td>
<td>adsorption</td>
<td>adsorption</td>
<td>absorption</td>
<td>absorption</td>
<td>absorption</td>
</tr>
<tr>
<td>Working pair</td>
<td>water/silica gel</td>
<td>water/zeolith</td>
<td>ammonia/water</td>
<td>water/LiBr</td>
<td>ammonia/water</td>
</tr>
<tr>
<td>Cooling capacity</td>
<td>7.5 and 15 kW</td>
<td>10 kW</td>
<td>12 kW</td>
<td>17.5 kW</td>
<td>50 kW</td>
</tr>
<tr>
<td>Heating temperature</td>
<td>75 / 68°C</td>
<td>85 / 77°C</td>
<td>85 / 78°C</td>
<td>88 / 83°C</td>
<td>115 / 105°C</td>
</tr>
<tr>
<td>Recooling temperature</td>
<td>27 / 32°C</td>
<td>27 / 32°C</td>
<td>24 / 29°C</td>
<td>31 / 35°C</td>
<td>25 / 30°C</td>
</tr>
<tr>
<td>Cold water temperature</td>
<td>18 / 15°C</td>
<td>18 / 15°C</td>
<td>12 / 6°C</td>
<td>12.5 / 7°C</td>
<td>-5 / -10°C</td>
</tr>
<tr>
<td>COP</td>
<td>0.56</td>
<td>0.50</td>
<td>0.62</td>
<td>0.70</td>
<td>0.55</td>
</tr>
<tr>
<td>Dimensions (LxDxH)</td>
<td>0.79 / 0.79 x</td>
<td>0.65 x</td>
<td>0.80 x</td>
<td>0.60 x</td>
<td>2.35 x</td>
</tr>
<tr>
<td></td>
<td>1.06 / 1.35 x</td>
<td>1.30 x</td>
<td>0.60 x</td>
<td>0.80 x</td>
<td>1.65 x</td>
</tr>
<tr>
<td></td>
<td>0.94 / 1.45 m³</td>
<td>1.65 m³</td>
<td>2.20 m³</td>
<td>1.77 m³</td>
<td>2.63 m³</td>
</tr>
<tr>
<td>Weight</td>
<td>260 / 510 kg</td>
<td>370 kg</td>
<td>350 kg</td>
<td>420 kg</td>
<td>1,600 kg</td>
</tr>
<tr>
<td>Electrical power</td>
<td>20 / 30 W</td>
<td>20 W</td>
<td>300 W</td>
<td>72 W</td>
<td>3,000 W</td>
</tr>
</tbody>
</table>

Findings and summary conclusions

In the small scale capacity range up to 30 kW several single-effect water/lithium bromide absorption chillers, one ammonia/water absorber as well as two water/silica gel and one water/zeolith adsorption chillers are market-ready available in Europe. All these chillers are specified as core components of solar/thermal cooling kits. Up to now over twenty chillii® Cooling Kits and chillii® Solar Cooling Kits respectively are in installed in Germany, Austria, Belgium, Spain, Italy, Malta, Romania, Canada, China and Australia. Different kinds of applications are realized like for residential buildings, retirement home, office buildings, bank, bakery, greenhouse and institutes. The first experiences and experimental results of the installed solar cooling kits showed that the chillers and the solar cooling system work very well. In case active cooling being necessary, the long running times of the chillers are the key for economic efficiency of solar or thermal cooling. For residential buildings in Central Europe only about 50 to 200 cooling hours occur, whereas in the southern Mediterranean area as well as for some industrial and office buildings approximately 1,000 full load hours are necessary.

Until now, the specific total costs of the solar cooling kits of SolarNext (without installation costs and cold distribution) have been between 5,000 and 8,000 EUR/kW cooling capacity. In 2008 average system costs of around 4,000 to 4,500 EUR/kW were reached depending on the application and the site. The average value of specific collector surface of all market available solar cooling kits is 4.2 m²/kW cooling capacity. SolarNext supplies the biggest collector areas with 4.5 m²/kW, Schüco the smallest areas with 3.2 m²/kW. The average value of all installed small to large-scale solar cooling systems in Europe until the year 2006 was 3.0 m²/kW. An all-season use of renewable energy sources for domestic hot water, space heating and solar cooling is here indispensable. The solar fraction for the solar cooling system should be more than 70%.

References

Abstract
Developing a database of adsorbents promising for adsorptive transformation of heat is very timely. This database would play an important role in unification of adsorbent properties, correct comparison of various adsorbents, theoretical analysis, mathematical modelling and brief estimation of heat transformation cycles. In this paper, we discuss principles of creating such database, consider the adsorbent properties which should be given there, and address the issues of their measurement and calculation. A tentative list of common and innovative adsorbents to be presented in the database is discussed as well.

Introduction
An adsorbent is a key element of an adsorptive heat transformer, AHT (heat pump, chiller, amplifier), and harmonization of the adsorbent with the cycle is of prime importance. It can be reached either by systematic screening of known adsorbents to select the best one for a given cycle or by tailoring of novel adsorbents adapted to this cycle (Aristov, 2007). Thermodynamic requirements to an optimal adsorbent were considered for a basic cycle in (Critoph, 1988; Aristov, 2007). First attempts to prognosticate which adsorbent is dynamically optimal for AHT have been done in (Aristov, 2009). Hence, it is well-timed to tabulate sorption properties of promising materials to make them available to anyone.

Principles of the database construction
The first approach bears in mind the adsorbent separately from the heat exchanger (HE) of AHT and tabulates those adsorbent properties which are necessary for modeling AHT cycle and calculating its main parameters. Because this methodology dominates at the moment, we first consider the structure of the database within this paradigm. For a given “sorbitive - adsorbent” pair, basic thermodynamic information can be obtained from the function q(P, T) which gives the equilibrium uptake at fixed pressure P and temperature T. The most popular models used to represent this function are considered. For each model a number of fitting parameters to be tabulated are discussed. An opportunity to use just one argument, the free sorption energy $\Delta F = -RT\ln(P/P_s)$, instead of the common two, P and T, is analyzed as well. The isosteric heat of adsorption $Q_{is}$ can be calculated from the equilibrium P and T or directly measured by the sorption-isosteric method. Restrictions of these methods are examined.

Another necessary thermodynamic function, a specific heat of adsorbent $C_p(q, T)$, can be measured calorimetrically at fixed q by a tight closing of measuring pan. Usually the efficient value of $C_p$ is represented as $(1-x)C_p$(dry adsorbent) + x $C_p$(adsorbate), where x is a weight fraction of adsorbate. The relation between $C_p$(adsorbate) in the adsorbed, gas and liquid states is discussed.

For a dynamic simulation of AHT cycle, three main functions have to be determined, specifically, the efficient diffusivity $D_{ef}(q, T)$, coefficient of heat transfer between the adsorbent and the HE wall $h(P, T)$, and thermal conductivity $\lambda(P, T, q)$ of adsorbent grain (layer). Methods for measuring these functions are reviewed and compared. Theoretical models for calculating these data are considered as well.

It is shown that the separate consideration of adsorbent and HE can be quite acceptable for estimating thermodynamic parameters of AHT cycles. However, for dynamic analysis this approach is not inherently consistent, in particular, the coefficient $h$ is not an intrinsic property of the adsorbent. It characterizes a layer configuration and cycle boundary conditions as well. A few reliable data on $D_{ef}$, $h$ and $\lambda$ are available in literature and their determination is extremely complex and time consuming. Moreover, these functions measured under quasi-equilibrium and real conditions of AHT cycle may differ.
**Alternative approach** to the dynamic characterization of AHT cycle is based on the idea that it is obligatory to take into account the adsorbent and heat exchanger as an integrated unit and tabulate dynamic properties of this unit measured at real operating conditions. Right away many questions arise. What cycles and boundary conditions should be considered as benchmarks? It is worthy to begin with a basic 3T cycle. No regeneration is assumed within this cycle. For 3T cycles, each set of boundary conditions can be designated as \((T_e, T_c, T_{HS})\), where \(T_e\), \(T_c\) and \(T_{HS}\) are the evaporation, condensation and heating temperatures (here we follow the nomenclature of (Pons, 1999)). Coming after this key review, we select four typical adsorptive applications, such as air-conditioning, ice making, heat pumping 1, and heat pumping 2. For each of these applications, two cases were suggested for the external cooling: for water-cooled units \(T_c = 40^\circ C\) and for air-cooled units \(T_c = 50^\circ C\) (or \(55^\circ C\) for heat pumping). The regeneration temperature \(T_{HS}\) can be chosen as \(T_{HS} = 80\) and \(100^\circ C\) keeping in mind utilization of low temperature heat. This results in four sets \((T_e, T_c, T_{HS})\) for each application, means, totally 16 sets. As a next step, other applications or boundary conditions may be considered.

**Which configurations of the adsorbent – heat exchanger (A-HE) unit should be selected as reference configurations?** The simplest, but practicable, configuration is one layer of grains loose-lying on a metal support. Other realistic configurations are two, three and so forth layers of loose grains. Then, several configurations of adsorbent grains consolidated to each other or and to a metal support by a binder should be considered at fixed binder nature and content. While choosing the basic configurations one has also to take into account a possibility of its correct mathematical modeling.

**Which dynamic properties of the A-HE unit should be measured and tabulated?** Isobaric stages of AHT cycle determine its dynamic behavior. The uptake evolution \(q(t)\) which is caused by a flash jump (drop) of the metal support temperature uniquely characterizes the dynamics of isobaric stages for given A-HE configuration and boundary conditions set. This function automatically takes into account all peculiarities of adsorption kinetics, heat and mass transfer in the A-HE system. Thus, the advantage of the second approach is the lack of necessity of performing complicated and time-consuming measurements of the dynamic parameters listed above. Moreover, if this function is determined, no sophisticated modelling is needed any more for analyzing dynamics of AHT units.

**How this dynamic function can be measured?** One of the ways to measure the function \(q(t)\) is given by a large temperature jump method which has recently been suggested in (Aristov et al., 2008). At each set of boundary conditions, the experimental function \(q(t)\) can be approximated analytically, and appropriate fitting parameters can be tabulated in the database. The first measurement for Fuji silica RD, FAM-Z02 and SWS-1L showed that for the simplest configuration “a monolayer of loose grains” \(q(t)\) is near-exponential and is described by a single characteristic time (Aristov, 2008; Glaznev, 2008)! Methods of measuring \(q(t)\) in middle and large-size AHT adsorbers are suggested and considered.

**Which pairs “sorbative - adsorbent” should be selected for this database?** The common working pairs “sorbive - adsorbent” promising for AHT applications were enumerated in (Pons et al., 1999). These are “\(H_2O - zeolite 4A\)”, “\(H_2O - zeolite NaX\)”, “methanol - active carbon” and “\(NH_3 - active carbon\)”. In addition, “\(H_2O - silica Fuji RD\)” has to be carefully characterized and tabulated together with several “new” pairs recently invented and studied for AHT application: “water - FAM-Z01, FAM-Z02, SWS-1L, SWS-2L”, “methanol – ACF, SWS-3L”, “ethanol – ACF”, “\(NH_3 - Busofit\)”, and “\(NH_3 - ACF\)”. This tentative list is of course a subject for further consideration and alteration.

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**References**


Abstract

In recent years, Ng et al. [1] have developed and experimented on the use of cost-effective and environmental friendly adsorption cycle for desalination (AD) that requires only low temperature waste heat to operate. Such waste heat is available in abundance from the exhaust of industrial processes, micro-turbines of co-generation plants, and renewable sources. Being waste heat-driven, useful effects are generated by the AD cycle with no additional burning of fuels and thus, mitigating the effects of global warming. The AD cycle has no major moving parts and it has low corrosion and fouling rates on the evaporator tubes as the saline solution evaporates at near ambient temperature as opposed to the conventional desalination systems. In addition to water production, the AD cycle offers cooling effect from the single energy input. The present paper describes the modeling and numerical simulation of the performance of an advanced adsorption desalination and cooling (AADC) cycle with internal heat recovery between the condenser and the evaporator. The heat recovery scheme enhances the water production by as much as twice the conventional rates while producing useful cooling energy.