# Abstracts of Heat Powered Cycles 2012

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DESIGN AND PERFORMANCE ANALYSIS OF A RESORPTION COGENERATION SYSTEM

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As a type of traditional cycle for the electricity generation, the main limitation of the Rankine cycle is the endothermic process for the evaporation of the water, which cannot always fit with the trend of heat source very well. The Kalina cycle could make up the efficiency loss by the working fluid of water-ammonia, which has a variable heating process by adjusting the concentration of ammonia in the absorbent of water. Based on the Kalina cycle a combined power and refrigeration generating cycle is proposed by Goswami, which generally has reasonable exergy efficiency but suffers from the low energy efficiency for the refrigeration process. In this study, a new type of the resorption cogeneration system is designed, which is shown in Figure 1. It mainly includes one high temperature salt (HTS) bed, one low temperature salt (LTS) bed, one turbine, one precooler, two heat exchangers, and one heating oil tank. Its working processes are:

Electrical generating and adsorption refrigeration process. In this process the HTS bed is heated by the low grade heat, the desorbed ammonia is expanded in the turbine and generates the electricity there. The expanded ammonia might provide the refrigeration in precooler if its temperature is lower than the refrigerating temperature. After that it will be adsorbed by the LTS bed.

The resorption refrigerating process. In this process the HTS bed is cooled by the environmental medium such as the cooling water, and it will adsorb the refrigerant, i.e. ammonia from the LTS bed when its pressure is lower than that of LTS bed. The desorption process of LTS will generate the refrigeration by desorption process.

The bed for the resorption system is designed as tube-shell heat exchanger. The consolidated adsorbent of chloride and expanded natural graphite, which is packed on the heat transfer pipes, is utilized in the system. The bed is designed for the 15 kW refrigeration capacity, and the highest pressure designed for the system is 3.5MPa, which is 0.5MPa higher than the working pressure of 3MPa.

When the BaCl₂ is chosen as the LTS the performances of different working pairs are analysed by fundamental equations for the cycle. For the system without heat recovery the results are shown in

![Figure 1: The system design](image-url)
Figure 2. $\eta_{El}$ and $COP_{ref}$ ranges between 0.072-0.126 and 0.33-0.53, respectively. The performance for the heat recovery cycle is shown in Figure 3. The $\eta_{El}$ and $COP_{ref}$ range between 0.095-0.158 and 0.416-0.691, respectively. The exergy efficiency of the system is also analysed, and the highest value is about 0.82, which is improved by 50% if compared with that of the Goswami cycle.

![Figure 2: The performance for the system without heat recovery](image)

(a) $\eta_{El}$ vs. superheating temperature, (b) $COP_{ref}$ vs. superheating temperature

![Figure 3: The performance for the system with heat recovery](image)

(a) $\eta_{El}$ vs. superheating temperature, (b) $COP_{ref}$ vs. superheating temperature

Considering the heat and mass transfer performance, the energy balancing equations were set up, and the results of 5 kinds of adsorbents and four working pairs were analysed, respectively. By analyzing the refrigeration performance of adsorbents, results showed that BaCl$_2$ compound adsorbent has the highest SCP and COP, which are 0.2138 and 79.4085 W/kg, respectively for one adsorbent tube. By analyzing the cogeneration performance for different working pairs with BaCl$_2$ as the LTS (Table 1), FeCl$_2$-BaCl$_2$ showed the best performance with SCP 54.89 W/kg, COP 0.271 and power generation of 0.452 kW. For different working pairs, SCP, COP, and power ranged from 40.15 to 54.89 W/kg, 0.2211 to 0.271, and 0.3346-0.452 kW, respectively.

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<td>18.28</td>
<td>54.89</td>
<td>0.271</td>
<td>0.452</td>
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ABSTRACT

A numerical analysis of the heat and mass transfer in an adsorbent bed during an isobaric desorption process of adsorption heat pump cycle will be simulated for varies power of microwave (180 and 360). The influence of power of microwave on the regeneration of adsorbent bed during isobaric desorption process will be investigated.

INTRODUCTION

The adsorbent that is used in the adsorption heat pump have a poor thermal conductivity. The poor thermal conductivity of the adsorbents might have a direct influence on the heat transfer. A survey of the literature revealed that the majority of the studies focused on eliminating the abovementioned problem. In these studies, although the thermal conductivity of the adsorbent was able to be improved by different methods, the performance of the adsorption heat pump could not be improved by using these new high thermal conductive adsorbents. Thus, the main obstacle in the practical application of the adsorption heat pump still remains (Demir et al., 2008).

Nowadays, dielectric heating or microwave heating has found a wide place in the application areas such as food drying, mineral drying and regeneration of the adsorbent used for the adsorption of a volatile organic compound. Microwave heating have many advantages over conventional heating such as:

- high heating rate
- material selective heating
- non-contact heating
- precise and controllable heating
- energy transferred rather than heat transfer
- compact equipment (Haque, 1999)

Ania et al., (1999) comprised the effect of microwave and conventional regeneration on the adsorption capacity of the activated carbons. The results showed that the microwave heating increased the phenol adsorption capacity of the activated carbons in comparison to heating with the electrical furnace. They explained the results as the microwave heating improves the stability and structure of the activated carbons. Kumja et al., (2008) modelled the microwave regenerated rectangular plate adsorbent bed. Their model revealed that the COP of the microwave driven cycle was higher than the conventional one. More applications of the microwave heating on desorption processes can be found at Cherbanski and Molga (2009).

RESULT AND DISCUSSION

The electric field over silica gel bed was simulated by using ANSYS 13.0 multiphysics software as shown in Figure 1. The rectangular box was used as the microwave cavity (30x18x30cm) and the silica gel bed was put in the middle of the box. One magnetron was used as electromagnetic source and the type of the waveguide was WR340.
**Figure 1: Distribution of electromagnetic field on silica gel bed for 360W power.**

The distribution of electromagnetic field inside the cavity and over the silica gel was obtained at the power of 360W. The results of the electromagnetic were integrated and averaged for each element (E). The defined electromagnetic field intensity will be used for calculating the heat generation term, $Po$, at Equation (1) which can be embedded into the energy balance equation (Eqn(2)) (Pere et al., 2001). The distribution of temperature, pressure and adsorbate concentration on adsorbent bed during isobaric desorption process will be simulated for different power of microwave.

$$Po = 2 \times 10^{-3} \pi f \varepsilon_0 \varepsilon_r E$$  \hspace{1cm} [1]

$$\left( \rho C_p \right)_{\text{eff}} + \rho_s C_p \frac{\partial W}{\partial t} = \lambda_{\text{eff}} \frac{1}{R} \frac{\partial}{\partial R} \left( R \frac{\partial T}{\partial R} \right) - \frac{1}{R} \frac{\partial}{\partial R} \left( \rho_s C_p R \frac{\partial T}{\partial R} \right) + \rho_s \Delta H_a \frac{\partial W}{\partial t} + \gamma \rho_0$$  \hspace{1cm} [2]

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MAGNESIUM HYDROXIDE - EXPANDED GRAPHITE COMPOSITE PELLETS FOR A PACKED BED REACTOR CHEMICAL HEAT PUMP

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INTRODUCTION

The chemical heat pump (CHP) is a promising technology for the recovery of waste heat from industrial processes or cogeneration systems. It can be used for storing the surplus heat during low demand periods and release it for shaving the heat peak demands, with a benefit for the overall system efficiency. In this work, a packed bed reactor (PBR) CHP based on the dehydration and hydration of magnesium hydroxide (Mg(OH)₂) has been investigated. Due to its high thermal conductivity, expanded graphite (EG) was mixed with Mg(OH)₂ to enhance heat transfer. The composite material, named EM, was developed and tested experimentally in order to understand the effects of EG on the chemical reactions occurring in the PBR.

PRINCIPLES AND GOAL OF THIS STUDY

In this work the CHP is based on magnesium oxide (MgO) and water (H₂O) adsorption reaction. Its operability was already demonstrated in previous work from Y. Kato (2003). It is based on the following equilibria:

\[ \begin{align*}
\text{MgO(s) + H}_2\text{O(g)} & \leftrightarrow \text{Mg(OH)}_2(s) \quad \Delta H^\circ_1 = -81.0 \text{ kJ mol}^{-1} \quad [1] \\
\text{H}_2\text{O(g)} & \leftrightarrow \text{H}_2\text{O(L)} \quad \Delta H^\circ_2 = -41.0 \text{ kJ mol}^{-1} \quad [2]
\end{align*} \]

The right direction reaction in Eq. [1] is called hydration. It is an exothermic reaction and corresponds to the heat output mode of the CHP. The left direction reaction in Eq. [1] is called dehydration. It is an endothermic reaction and corresponds to the heat storage mode. Figure 1 shows a schematization of the CHP cyclic reactions. Mg(OH)₂ and MgO have poor heat transfer properties; it means that for using the CHP technology, large surface for heat exchangers and sophisticated engineering are required, therefore high costs. EG is a commonly employed material to enhance the heat transfer properties of adsorption salts: it has high thermal conductivity, chemical stability and does not react with adsorbents. A composite material, named EM, was obtained by drying a mixture of Mg(OH)₂ powder, EG and water. EM was compressed into pellet shape (diameter \( d = 7.1 \text{ mm} \), thickness \( h = 3.8 \text{ mm} \)). The mass ratio of Mg(OH)₂ to EG was fixed as 4:1, therefore the composite was called EM4.

EXPERIMENTAL APPARATUS AND METHOD

The experimental apparatus is an assembly of a vacuum chamber and a water reservoir (WR); they are linked by a heated tube. Inside of the vacuum chamber, the PBR finds its position. Heat is supplied through a sheath heater wrapped on the side surface of the reactor. The chamber is mounted over an electric balance: the reactor’s weight change is caused by the movement of water vapor from the packed bed to the WR or vice versa during the chemical reactions. From periodic recording of weight it is possible to calculate the reacted fraction change \( \Delta x [-] \), observe the reaction progress and calculate the stored energy. Dehydration consists in heating the PBR at 400°C until a steady state of temperatures and weight change is reached. Generally the reaction was concluded after at least 90 minutes.
RESULTS AND DISCUSSION

Figure 2 shows the time dependent temperature at the center of the PBR and the reacted fraction change $\Delta x$. Compared to Mg(OH)$_2$, EM4 is characterized by a steeper temperature profile curve, which means that EG enhanced the heat transfer into the packed bed. This benefit is translated in a faster reaction rate, therefore energy can be stored in a shorter time. It was observed that for EM4 composite the dehydration could start within 5 minutes of heat supply. From the reacted fraction change $\Delta x$, the stored energy per unit mass, $Q_m$, could be obtained. Results are shown on Figure 3, in particular evaluated after 5, 10 and 30 minutes of reaction progress. It was calculated that EM4 could store 143 and 553 kJ kg-pellet$^{-1}$ of thermal energies after 10 and 30 min, respectively. Those values corresponded to 7.7 and 1.3 times of stored thermal energy for Mg(OH)$_2$ pellet, respectively.

CONCLUSION

A composite material made of Mg(OH)$_2$ and EG, called EM, was developed and tested on an CHP apparatus. EG enhanced the thermal conductivity and allowed accelerating the dehydration reaction. EM4 pellets began to store heat earlier than pure Mg(OH)$_2$ ones and could accumulate a larger amount of thermal energy per unit mass. Further investigation on optimization of mixing mass ratio was required in order to fully understand the effects of EG on the chemical reactions in the packed bed.

REFERENCES

HEAT-POWERED CYCLES: ARE THE PROCESS INDUSTRIES 'MISSING THE BOAT'?

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SETTING THE SCENE

The process industries are striving to reduce energy costs, and engaged in such activities for decades. Oliver Lyle’s ‘The Efficient Use of Steam’, first published in 1947 when Britain was in the throes of a severe post-war energy shortage, was regarded as the steam-users’ bible if one was striving for energy efficiency. Waste heat recovery, including the use of excess steam, featured here, as it did a year earlier (1946) in a conference organised by The Institute of Fuel (now The Energy Institute) on 'Waste Heat Recovery from Industrial Furnaces'. The theme was continued in 1961 where 21 papers were presented in Bournemouth, UK on topics ranging from waste heat boilers to waste-heat recovery in the ceramics industry. Interestingly, at this time coal provided 70% of the UK's energy needs).

It was the heavy industry sectors - nitric acid production, for example - that featured in the 1960s as users of heat-powered cycles. In this example both gas turbines (using let-down pressure recovery from hot gases) and steam turbines (fed by waste-heat boilers) that featured strongly. On another plant, vapour recompression on an ethane/ethylene distillation column was used to provide high-grade heat for the reboiler.

If we jump forward another 15 years to the oil crises of the 1970s we again find a substantial literature on heat-powered cycles directed at industrial processes. R&D was typified by the efforts of the European Commission in their 4th Framework Programme, where I recall many projects on organic Rankine cycle (ORC) machines, as well as absorption cycle machines, and subsequently, process heat transformers (Reay, 1982). One of the highlights was a study by MBB in Germany of the potential for ORC units in the chemical industry - a potential that was never realised.

There were a few highlights in the European HPC scene in 2011. The Delft ORC Seminar in September attracted over 200 delegates and many papers (Web 1), including several process-related papers, and the Nuremberg Heat Pump Summit, also in September, featured several contributions related to the ever-interesting topic of industrial (including high temperature) heat pumps (see for example Paaske, 2011). Nevertheless, heat-powered cycles are now more commonly associated with domestic combined heat and power (DCHP) or micro-chip cooling! This neglects a major market - the process industries - that micro-systems and decentralisation will not satisfy in the near future.

PROPOSING SOLUTIONS

If we look at Figure 1, (Law et al, 2011), it can be seen that capital outlay varies depending upon the form of energy recovery in which we wish to invest. Heat pumps, ORC machines and absorption (not to forget adsorption) cycles are third in the pecking order, as determined by their capital (or installed) cost. Regardless of the perceived benefits in terms of energy cost savings, the first cost is often the headline figure that attracts the accountants' eyes. Heat-powered cycles unfortunately tend to be expensive compared to other energy-saving equipment, and while in the UK we used to enjoy good financial support for improved process technologies from Government, the trend now is mean’, not 'green'!

SO WHAT CAN WE DO TO ADDRESS THE COST DISADVANTAGE OF HPCS?

In this paper it is proposed to examine the opportunities for reducing the capital and installed cost of the equipment in a number of ways, including the use of compact heat exchangers, as suggested by Kew et al (2011). Has miniaturisation got a role to play in the process sector, as opposed to electronics thermal management etc, (Determan et al, 2001)? Going the extra mile - can process
intensification of the HPC unit operations (Reay et al, 2008), or more extensive integration with the principal processes in the factory, help to overcome the first cost barrier.

The technology is available and options will be presented in the full paper.

![Diagram of heat recovery options](image)

**Figure 1:** Capital outlay of various heat recovery options, (Law et al, 2011).

**REFERENCES**


PRIME MOVER SIZING IN A MICROTURBINE-BASED CHP SYSTEM WITH DIFFERENT OFF-DESIGN PERFORMANCE STRATEGIES

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Combined Heat and Power (CHP) systems emit less pollution in comparison with a separate production of electricity and heat. Optimum sizing of CHP systems is required to maximize the benefits of these systems. To select the optimum prime mover of a combined heat and power system, performance characteristics of engine as well as economic parameters should be taken into consideration.

Microturbines are the new generation of small gas turbines which have entered the market recently. They operate on the same thermodynamic cycle as gas turbines (i.e. Brayton cycle). To decrease the fuel consumption and consequently increase the thermal efficiency, most microturbines are equipped with a recuperator which uses hot turbine exhaust gas to preheat the compressed air going into combustor. However, applying recuperator reduces the exhaust temperature and microturbine’s effectiveness when utilised as the prime mover of a combined heat and power system. Microturbines are compact in size, can be brought on-line quickly, and offer fuel flexibility; therefore, they are one of the leading prime movers of CHP systems, particularly in commercial and residential applications. In order to supply larger demand, microturbines are stacked in parallel.

This study aims to further develop a thermo-economic method for selecting the optimum number and nominal power as well as planning the operational strategy of microturbines as the prime movers of small scale combined heat and power systems (capacities up to 500 kW) (Aghaei Meybodi, 2011). We consider the impact of two different strategies of working in off-design conditions. In our previous study (which is referred to as case 1), all microturbine units worked at the same partial load (in the range of 20% to 100%) while in this study (case 2), working condition of units is different; one unit is set to accommodate load variation and other units work at full load. If load variations lead to working at partial loads below 20%, the unit is turned off and another microturbine starts to work in partial load condition.

The economic method for developing the objective function and comparing various cases is the Net Present Worth (NPW) approach. In the proposed selection procedure both performance characteristics of the prime mover and economical parameters (i.e. capital and maintenance costs) are taken into account. The impact of carbon tax on the economics of system has also been considered. The studied modes of operation are: one-way connection (OWC) mode, two-way connection (TWC) mode, and heat demand following (HDF) mode. Buying electricity from the grid is allowed in all three operational modes, while selling excess produced electricity to the grid is only possible in TWC and HDF modes of operation. With HDF mode, in order to minimize the waste of energy, prime movers work in a condition at which the excess produced heat is minimal. It is noted that the decision as to which operational strategy could be adopted depends mainly on the method of connection to the grid (i.e. the possibility of selling electricity to the grid).

As a case study, a local aquatic centre has been considered. The results showed that in all operational modes, the optimum nominal power and number of units is the same for both cases. In TWC and OWC modes, for all studied microturbine models NPW values associated with case 1 are higher than those of case 2 and therefore case 1 is more economical. With HDF mode, case 2 results in higher NPW values and consequently is the preferred off-design performance strategy. It is noted that despite leading to higher NPW values, similar to case 1, NPW values are negative and HDF mode is uneconomical.
REFERENCES

OPTIMIZATION OF A COGENERATION CYCLE BY GENETIC ALGORITHM AND EXERGOECONOMIC ANALYSIS OF THE CYCLE USING NEW EXERGETIC CONCEPTS

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ABSTRACT

In recent decades, to use waste heat of power generating systems and reversible energy resources, reducing fossil fuel consumption and harmful effects of these systems on the environment, also because of decreasing the fossil fuel resources, use of efficient energy systems such as Co-generation systems that capable to produce different types of energy including power and heat or cold simultaneously, more attention is placed. Besides, in the past few years, new exergetic concepts such as endogenous/exogenous and avoidable/unavoidable exergy destruction and investment costs have been introduced. These concepts provide valuable information about the potential of optimization of the system. In every thermodynamic component, the part of exergy destruction which is due to internal irreversibilities of components, is endogenous exergy destruction and the other part which is due to effect of internal irreversibilities of other components, is exogenous exergy destruction. On the other hand, in optimization process of the system, because of technical and economical limitations, only a part of exergy destruction and investment costs in every component can be avoided and the other part is unavoidable. These two concepts can be combined and this privilege increases the accuracy of analysis of the system.

In present work, in the first step, the co-generation cycle has been analyzed by conventional and new exergetic concepts. Later the cycle has been analyzed by economics concepts and conventional and new exergoeconomics concepts. Results indicate that, the highest exergy destruction occur in the combustion chamber. Also results show that 32.6 percent of exergy destruction and 73 percent of investment cost can be avoided by optimization.

Finally, the co-generation cycle is optimized using Multi-objective Genetic algorithm. Results indicate that exergy efficiency of cycle is improved while that cost of production is lower than base state.

DESCRIPTION OF THE CYCLE

In the present work, optimization of a co-generation cycle which is illustrated in Figure 1, is investigated by considering two different target functions. These functions include second law efficiency and the operational cost rate of the cycle. The decision variables considered in this study are as follows: compressor pressure ratio, compressor isentropic efficiency, gas turbine isentropic efficiency, combustion chamber inlet temperature, and turbine inlet temperature. The following physical constraints should be satisfied in heat exchangers:

\[ T_c > T_{in}, \quad T_r > T_{in} ; \quad \Delta T_{rw} > 0, \quad T_{in} \geq T + \Delta T_{rw}, \quad T_{in} \geq T_{in} + \Delta T_{fw}, T_{in} \geq T_{in}, T_{in} \geq 378.15 \]

This cycle produces 30 MW of electrical power and 14 kg/s of saturated steam at 20 bar at base conditions. In this cycle, the pressure ratio, \( r_p \), of the compressor is equal to 10. The isentropic efficiencies of compressor and gas turbine, \( \eta_c \), are assumed 86 percent. The turbine inlet temperature is 1520 K.
ADVANCED EXERGY ANALYSIS BASED ON SPLITTING EXERGY DESTRUCTION OF EVERY COMPONENT TO ENDGENOUS, EXOGENOUS, AVOIDABLE AND UNAVOIDABLE PARTS IS ADDED TO THE ANALYSIS OF OPTIMIZED CYCLE WHICH DEFINES THE POTENTIAL OF OPTIMIZATION OF EACH CYCLE COMPONENTS. MAIN RESULTS OF THE ANALYSIS CAN BE CONCLUDED AS FOLLOWING ISSUES:

OPTIMIZATION RESULTS SHOW THAT THE PERFORMANCE OF THE CYCLE IS IMPROVED SO THAT EXERGETIC EFFICIENCY IS INCREASED IN ABOUT 2.94 PERCENT AND TOTAL COST RATE IS DECREASED IN ABOUT 10.14 PERCENT IN COMPARISON WITH BASE CYCLE.

ADVANCED EXERGY ANALYSIS REVEALS THAT UNAVOIDABLE EXERGY DESTRUCTIONS OF THE OPTIMIZED CYCLE COMPONENTS ARE REDUCED IN COMPARISON WITH BASE CYCLE. AS MORE THAN OF 31 PERCENT OF THE EXERGY DESTRUCTIONS OF THE OPTIMIZED CYCLE CAN BE AVOIDED.

ADVANCED EXERGEOECONOMIC ANALYSIS SHOW THAT AVOIDABLE COST RATE RELATIVE TO THE TOTAL COST RATE OF THE CYCLE CAN BE INCREASED FROM 54 PERCENT IN THE BASE CYCLE TO THE 61 PERCENT IN THE OPTIMIZED CYCLE.

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ABSTRACT

Reducing energy consumption is becoming an increasingly important issue in UK industry because of two key factors. Firstly, the ever-increasing cost of electricity and fossil fuel resources provides a growing monetary incentive for change. Secondly, the stringent, self-imposed government legislation in the Climate Change Act, 2008, provides a legal obligation to reduce greenhouse gas emissions according to ambitious goals.

Given the ever-rising UK population, the demand for industrial produce is unlikely to decrease. Therefore, the emphasis is on reducing industrial energy consumption by increasing energy efficiency. A key way of achieving this is by recovering waste heat streams.

Heat pumps and organic Rankine cycles can prove useful technologies for the recovery of waste heat in certain circumstances. While the most economical solution for the recovery of waste heat is often by simple heat transfer between heat source and sink, frequently the quantity of waste heat available outweighs the required heating duty of suitable heat sinks, particularly in the recovery of low-grade waste heat. In such cases, heat pumps and organic Rankine cycles should be considered (Law et al., 2011).

Both of these technologies are yet to be implemented on a large scale in the UK, with UK engineers citing high risk of investment and uncertainty as potential reasons (Sinclair, 2001) despite the common uptake of such projects in Europe.

A techno-economic comparison of both an organic Rankine cycle and a high temperature heat pump for low-grade waste heat recovery for a specific UK industry case study is carried out. The case study in question has a surplus of low-grade waste heat available thereby preventing the use of simple heat transfer methods for complete waste heat utilisation. The plant does, however, have a large electricity demand which an organic Rankine cycle may help to reduce, and a low-pressure steam demand which may be satisfied using a high temperature heat pump. Therefore both solutions appear to offer a credible approach for the recovery of low-grade heat.

Technically, one would expect the benefits of a heat pump to far exceed that of the organic Rankine cycle alternative due to the much greater heat recovery efficiency of heat pump systems (ORCs achieve only around an 18% generation of useful work compared to the duty of heat recovered). However, the UK provides a unique case study in comparison to many EU countries in that the cost ratio of electricity to gas exceeds 3:1 (Department of Energy and Climate Change, 2011), and also that greenhouse gas emissions associated with standard electricity generation is uncommonly high at around 0.52kg CO₂-equivalent per kWh (Carbon Trust, 2011) due to the outdated nature of the power generation infrastructure and the large number of coal-brining plants. Therefore, in terms of economic and environmental performance, a closer comparison of ORCs and heat pumps is expected for low-grade heat recovery than would be in other countries across Europe.

The two technologies are theoretically modelled using the Aspen Plus software package according to Figure 1 and 2 below. R-245fa is found to be the most suitable working fluid for both technologies in this case study and is used in the models.
The model results are analysed for potential cost savings and reduction in greenhouse gas emission. Estimates of the capital cost of each technology are also provided to give an estimation of simple pay back times. Recommendations are made for the most appropriate type of waste heat recovery for this particular case study.

REFERENCES

**INNOVATIVE WATER VAPOR PERMEABLE COATINGS SUITABLE FOR THE USE IN ADSORPTION CHILLERS AND HEAT PUMPS**

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Direct coating of an adsorbent on a heat exchanger surface is an excellent possibility to enhance heat and mass transfer within adsorption chillers and heat pumps. Starting from the well-known zeolite bed configuration, a poor thermal contact between the pellets and the heat exchanger wall results in slow adsorption dynamic and therefore long cycle times, as has been reported before (e.g. Dawoud 2010; Freni 2010). In this work we report both, equilibrium and dynamic water uptake results on a new developed coating with a broad variety of binder/adsorbent ratio.

**INTRODUCTION**

Within the last years, several coating techniques with regard to the application in adsorption heat pumps or chillers have been reported. In principle, the different techniques can be divided into direct crystallization (e.g. Bauer 2009; Bonaccorsi 2006) or binder based (Dawoud 2010) coatings. In this work a novel binding system has been used to prepare coated metal plates. Unlike common known binder based coatings, as reported in Bauer 2009, this novel system shows no influence on the material properties at all. As verified by equilibrium measurements (see Figure 1) there is now pore plugging and hence loss of accessibility and no organic contents which may cause problems in vacuum setups. Instead the coatings show an excellent thermal and mechanical stability and a wide compatibility to various adsorbent materials. In addition, the binder shows a huge flexibility which is demonstrated by the preparation of five different dispersions with varying binder ratios from 100 wt% down to 2.5 wt%. As adsorbent a SAPO with chabazite framework type was used, as this material shows a good water uptake and hydrothermal stability (Henninger, 2010). The dispersions were manually coated on 50 × 50 mm aluminum-plates and dried at 200°C. The achieved thickness of the film is about 300 µm.

**WATER ADSORPTION PERFORMANCE**

The water adsorption characteristics of the coatings were measured in a Rubotherm thermobalance under pure water vapor atmosphere. The maximum water uptake of the pure adsorbent material is 0.318 g/g, as can be seen in Figure 1. The binder itself shows no water uptake. Furthermore, the equilibrium water uptake is not influenced by the binder, as the isobars are shifted exactly to lower loadings according to the binder amount. For example, the maximum loading for a ratio of 2.5 wt% binder is 0.310 g/g whereas the maximum loading for a ratio of 25wt% is 0.227 g/g.

The adsorption kinetics have been measured by a pressure jump method in a self-made apparatus (as described in Schnabel 2010). To quantify the adsorption dynamics, the adsorption half time and the rise up time according to Dawoud (2010) has been calculated. For the samples, the half time 

$t_{0.5}$

is quite similar with only 10s which is 10-times faster compared to the 300 µm layer reported in Dawoud (2010).

In addition the rise up time $t_{0.8} - t_{0.15}$ is 30s for 2.5wt% and 38s for 25wt% respectively and therefore 6-times faster compared to the results of Dawoud (2010).
CONCLUSION

A novel binding system for the use in adsorption heat pumps and chillers has been developed and various small metal plates have been successfully coated. The binder is compatible to various zeolites like materials and forms mechanical stable coatings even with 2.5 wt% binder/adsorbent ratio. In addition the coating shows a good thermal stability and enhanced heat and mass transfer properties. Out of the first results, no influence on the water adsorption properties has been observed. The kinetic of the adsorption is very fast and takes place within under 100 s. Out of these promising results no limitation due to reduced water vapor transport through the coating can be seen.

The introduced coating has a huge potential for improving the power density and reduce the production costs of adsorption heat pumps and chillers.

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A NOVEL DEHUMIDIFICATION AND COOLING SYSTEM DRIVEN BY AN EVACUATED TUBE SOLAR AIR COLLECTOR

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Due to enhanced comfort requirements and rising temperatures the energy demand spent on cooling of buildings is increasing significantly. In the given situation the integration of renewable energy for building HVAC is of great importance for reducing CO₂ emissions in that sector. As the so-called latent cooling is consuming the biggest share of energy in state of the art compressor cooling systems in maritime and tropical climates, simultaneous dehumidification of ambient air and cooling driven by solar heat can reduce the primary energy demand and reduce peak loads in power grids. Due to that background, Fraunhofer ISE is developing a solar driven dehumidification and cooling system in a collaborative project with the partners Kollektorfabrik GmbH & Co. KG and Contherm Wärmedämmsysteme GmbH.

SYSTEM DESCRIPTION

The HVAC system, which is currently under development, consists of an evacuated tube solar air collector from the Kollektorfabrik with an open dehumidification and cooling system. The setup is an ECOS-type using an evaporatively cooled sorptive coated cross-stream heat exchanger (Motta 2005, Bongs 2011). Since space for rooftop installation is rare due to the competition with several utilisations like photovoltaics, the collector will be developed to a facade integrated type supplying the driving heat with air temperatures up to 150°C.

The adsorbent material, which is coated on the surface of the sorptive heat exchanger channel, is regenerated in the desorption stage by means of hot air from the collector. In the adsorption stage, after a short pre-cooling of heat exchanger and housing, ambient air is dehumidified in the sorptive channels and simultaneously cooled down by indirect evaporative cooling. For this purpose the waste air stream in the cooling channels is saturated by water. In difference to the ECOS-system, the air handling unit will be reduced to one heat exchanger for reasons of compactness as well as installation cost. This will lead to a discontinuous cooling supply with full-cycle durations of approx. 20 minutes, circulating the air in the meantime using the room as a buffer.

Figure 1: Scheme of the HVAC system with detailed view of a rooftop mounted collector and a CAD model of the cross-stream heat exchanger
DEVELOPMENT OF THE ADSORPTIVE COATED HEAT EXCHANGER

The aluminum heat exchanger with dimensions of approx. 400x400x400 mm\(^3\) will be manufactured by brazing to ensure good leakage tightness against air, water and vapour. Different adsorbents have been evaluated for the given boundary conditions of the application. The characteristics of some materials in focus with highlighting the region of interest and maximum uptake reachable under equilibrium conditions are given in Figure (adapted from Bongs 2009). The chosen adsorbent will be attached to the heat exchanger surface by a new developed coating technique.

![Figure 2: Characteristic uptakes of suitable adsorbents for the DEC-application with the evacuated tube solar air collector: The normalized uptake of a Silicagel, Zeolite-Y, and SAPO-34 are compared; Silicagel and SAPO-34 should perform best under these conditions.](image)

Besides estimations from static equilibrium data a dynamic, validated 2-dimensional model implemented in Modelica (Bongs 2011) will be used for a performance prediction of different adsorbents. The air handling unit will be installed in summer 2012, performance measurements and monitoring will begin accordingly. As a goal assumed from pre-calculations a dehumidification performance of 5g/m\(^3\) at a flow of 200 m\(^3\)/h is set.

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REFERENCES

NANOTAILORING OF COMPOSITE SORBENTS OF AMMONIA FOR ICE MAKING AND AIR CONDITIONING APPLICATIONS

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INTRODUCTION

Performance of an adsorption heat transformer (AHT) is strongly affected by adsorption equilibrium of the working pair “adsorbent – adsorbate”. The selection or synthesis of adsorbents, whose properties match the operating conditions of the particular AHT cycle, is one of the encouraging ways to enhance the AHT performance. Composites “salt inside porous matrix” have been proposed as water, methanol and ammonia sorbents for AHT [Aristov et al, 2002]. The salt (S) is the active component, which reacts with refrigerant vapor (V) forming the complexes

\[ S + NV = S\cdot NV \] [1]

That results in large sorption capacity of the composite. Furthermore, the sorption equilibrium of composites with the refrigerants can be intently modified that gives a wide scope for designing the sorbent optimal for the particular cycle. The confinement into matrix’s pores of two salts, which affect each other, is an efficient tool for managing the composites sorption properties [Gordeeva et al, 2012]. The aim of this communication is the intent design of composite ammonia sorbents specialized for adsorption ice making (IM) and air conditioning (AC) sorption cycles. The requirements of the cycles for the optimal adsorbents were formulated; the real sorbents based on barium halides in the vermiculite pores were intently prepared and studied.

DESIGN OF THE ADSORBENTS

The typical operating conditions of IM cycle driven by low temperature heat can be determined as follows: the temperature of cooling water \( T_{\text{cool}} \) and of condenser \( T_{\text{con}}=30^\circ\text{C} \), the evaporator temperature \( T_{\text{ev}}=5^\circ\text{C} \), the heating water temperature \( T_{\text{heat}}=90-100^\circ\text{C} \). When the composite is used as the sorbent some overcooling below the equilibrium reaction temperature \( T'(P) \Delta T = T'(P_{\text{ev}}) - T_{\text{cool}} \Delta T \) is necessary to provide a reasonable driving force for reaction [1] [Veselovskaya and Tokarev, 2011]. Thus the “optimal” sorbent for considered IM cycle has to react with ammonia at \( T'(P_{\text{ev}}) = T_{\text{cool}} + \Delta T \approx 40^\circ\text{C} \) at \( P_{\text{ev}} = 3.5 \) bar. The equilibrium temperature of the complex decomposition at \( P_{\text{con}} = 11.2 \) bar can be evaluated as \( T'(P_{\text{con}}) \approx 80-90^\circ\text{C} \). Following the same logic the requirements for the optimal adsorbent for AC cycle \( T_{\text{cool}} = T_{\text{con}} = 35-40^\circ\text{C}, T_{\text{ev}} = 10^\circ\text{C}, T_{\text{heat}} = 90-100^\circ\text{C} \) are formulated: \( T'(P_{\text{ev}}) \approx 45-50^\circ\text{C} \) at \( P_{\text{ev}} = 5.8 \) bar and \( T'(P_{\text{con}}) \approx 80-90^\circ\text{C} \).

BaCl\(_2\) and BaBr\(_2\) react with 8 molecules of ammonia forming complexes BaHal\(_2\)-8NH\(_3\) that allows sorption \( \Delta w = 0.65 \) and 0.46 g/g, respectively [Touzain, 1999]. However, the formation of BaCl\(_2\)-8NH\(_3\) occurs at lower temperature than that estimated for the optimal adsorbent. On the contrast, the decomposition temperature of complexes BaBr\(_2\)-NNH\(_3\) is too high. Hence, both single salts do not meet the formulated requirements. It has been shown [Gordeeva et al, 2012] that the binary system (BaCl\(_2\)+BaBr\(_2\)) in silica gel pores reacts with ammonia at intermediate temperature between those for BaCl\(_2\) and BaBr\(_2\) due to the formation of the salts solid solution. For shifting the temperature \( T'(P) \) to the required value we used the binary system (BaCl\(_2\)+BaBr\(_2\)) as the active component of the composites. Keeping in the mind the formulated requirements and the dependence \( P_{\text{sat}}(T,C_{\text{BaBr2}}) \) observed in [Gordeeva et al, 2012] the optimal molar salts ratio in the composite was estimated as \( C_{\text{BaCl2}}/C_{\text{BaBr2}} = 1/1.2 \) and 3/2 for IM and AC, respectively. Composites BaClBr\(_2\)-V-1/1.2, BaClBr\(_2\)+Br\(_2\)-V-3/2 \((n/m\) indicates the Cl/Br ratio\), BaCl\(_2\)\(_2\) and BaBr\(_2\)\(_2\) were synthesized by dry impregnation method.
MAIN FINDINGS AND CONCLUSIONS

The data of temperature programmed desorption TPD (Figure 1) and the isosteric charts of the composites demonstrate that ammonia adsorption on BaClBrV-1/1.2 and BaClBrV-3/2 occurs at temperature ranges intermediate between those for single salt composites BaCl\(_2\)V and BaBr\(_2\)V. The larger the bromine content the higher the decomposition temperature. Thus, the addition of BaBr\(_2\) to BaCl\(_2\) allows the raise of BaHal\(_2\)-NH\(_3\) formation temperature. Dynamics of ammonia ad/desorption was studied by Large Temperature Jump method [Aristov et al, 2008] under conditions of isobaric stages of the cycles. Dynamic curves of ammonia sorption on BaClBrV-1/1.2 and BaClBrV-3/2 (Figure 2) obey the exponential equation

\[
\Delta w(t) = \Delta w_{\text{max}} \left( 1 - \exp\left(-t/\tau\right) \right)
\]

The data obtained allows an estimation of the BaHal\(_2\)-NH\(_3\) formation temperature. For composite BaClBrV-1/1.2 \(T^*\)(Pev) = 39°C at Pev = 3.5 and for BaClBrV-3/2 \(T^*\)(Pev) = 48°C at Pev=5.8 bar, respectively. Both composites are regenerated at Theat = 85-90°C. Thus the properties of the composites BaClBrV-1/1.2 and BaClBrV-3/2 intently prepared for IM and AC cycles, respectively, match the formulated requirements.

The cycles Specific Cooling Power (SCP) and Coefficient of Performance (COP) were evaluated as function of the cycle duration on the base of the data obtained. The maximum cooling power, realized at the beginning of adsorption, ranges between 1-2 kW/kg for both composites. At complete conversion (or at infinite cycle duration) COP reaches 0.58, however SCP is very low. The restriction of the cycle duration allows realization of the cycles with SCP = 350-540 W/kg and
COP = 0.53-0.54. The specific IM rate can be estimated as 2 kg/(kg-h) at the conversion equal to 0.9 that is much larger than that for the cycles based on physical adsorbents (0.02-0.18 kg/ (kg-h)). It is worth noting that each the composite provides better efficiency under conditions of the cycle for which it was tailored. This demonstrates encouraging opportunities of the target-oriented design of composites based on binary salt systems with predetermined properties for the particular adsorption cooling cycles.

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EXPERIMENTAL STUDY ON HEAT AND MASS TRANSFER IN A GRANULAR ADSORBENT BED

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Keywords: adsorption heat pump, adsorbent bed, silica gel, heat and mass transfer

Thermal heat pumps have gained attentions of researchers in recent years due to limited conventional energy sources and environmental issues. Adsorption heat pump (AHP), which is a kind of thermal heat pump, can recover heat at low temperature levels and provide cooling effect (Ülkü, 1986). It can operate with solar or geothermal energy, and utilize waste heat. Moreover, it has ability to store energy which can be utilized later on. As a disadvantage, it has lower COP values compared to the mechanical heat pump.

An adsorption heat pump contains four main components as adsorbent bed, condenser, evaporator, and expansion valve. The adsorbent bed is the crucial component of adsorption heat pump. Slow heat and mass transfer process in the adsorbent bed is the main drawback of the adsorption heat pump. The enhancement of heat and mass transfer rate reduces the period of cycle and consequently increases its power.

An experimental setup was designed and constructed to investigate heat and mass transfer in an adsorbent bed during the adsorption process. The setup was mainly composed of an adsorbent bed and an adsorbate container. The experiments were performed by silica gel - water as an adsorbent - adsorbate pair. 12 thermocouples were placed in the bed, in order to measure the temperature variation within the adsorbent bed. The components of setup, the location of thermocouples and pressure transducers are shown in Figure1, schematically. The adsorbent bed was constructed from a stainless steel pipe with 4 mm thickness and 309 mm inner diameter. The height of adsorbent bed is 290 mm. The bed was divided into three parts and the silica gel granules were located in the middle of the adsorbent bed. The upper and lower parts of adsorbent bed are filled with insulation materials to provide radial heat and mass transfer. Thermocouples were located at 0, 120, and 240 degrees in θ direction. This configuration provides measurement of temperature at 6 points in the adsorbent bed. In the middle of the adsorbent bed, a free space with 80 mm diameter in vertical direction was provided to remove flow resistance in axial direction. The level of adsorbate was measured from a sight glass mounted in the casing of adsorbate container. The adsorbent bed was heated by electrical resistance around the outer surface. Two identical pressure transducers were used on the setup to measure the pressures of the evaporator and the adsorbent bed. The temperature and pressure were measured and acquired by using data loggers.

Many experiments were performed; however the results of two experiments are presented. By measuring the temperatures and pressure of different points in the adsorbent bed, the temperature and bed pressure profiles during the adsorption process were obtained and plotted. The change of amount of adsorption in the silica gel with time was measured and plotted for both experiments. The evaporator temperature and pressure were also plotted and necessary discussions were performed. The obtained results show that for the designed adsorbent bed, heat and mass transfer does not depend on angular direction; however the small changes in the vertical direction was seen due to heat loss from the top and the bottom of the adsorbent bed. One of important results of this study is that thermal equilibrium model is valid for the designed system.
Figure 1: The experimental setup, (a) schematic view, (b) the silica gel granules in the adsorbent bed.

REFERENCES

SUPERCritical ORGANIC RANKINE CYCLE FOR GEOTHERMAL APPLICATION

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MOTIVATION

In 2010 the proportion of geothermal applications on the overall power generation within Germany is less than 0.01 %. In contrast the federal government aims for 20 % power generation out of renewable energy sources. As geothermal power plants are base-loadable compared to other sources like wind and solar energy the technology shows high potential for renewable power production.

OBJECTIVE

Geothermal heat source temperatures vary due to geological conditions in the range of 100 °C to 190 °C. Binary power plants like the Organic Rankine Cycle (ORC) or the Kalina Cycle (KC) are predominantly chosen for energy conversion in geothermal applications. Figure 1 shows a scheme of a standard ORC with internal recuperator.

One way to increase exergetic efficiency as well as power output of a standard subcritical ORC is the usage of fluid mixtures within the plant. Due to non-isothermal phase-change, efficiencies can be increased by 15 % (Heberle et al., 2012). To avoid pinch point restrictions within the plant, a supercritical mode of operation can be used. Saleh et al. (2007) concluded that for the working fluid R134a a raise in thermal efficiency of 17 % can be reached, Schuster et al. (2010) specified a potential of 8 % increase in exergetic efficiency for supercritical mode of operation. Both studies are based mainly on thermodynamic analysis and without a special application. Within the present work exergetic efficiency as well as power output will be investigated for geothermal applications and the effects on payback time and cashflow will be discussed based on the special boundary conditions of geothermal power plants (high development costs, etc.).
RESULTS

Eight different fluids out of two chemical classes are investigated. Refrigerants R227ea, R236ea, R236fa, R245fa and RC318 as well as hydrocarbons isobutane, isobutene and isopentane are considered. Geothermal temperature is varied between 100 °C and 190 °C in steps of 10 K. Subcritical and supercritical mode of operation are compared with regard to exergetic and thermal efficiency as well as gross and net power. The results show that the supercritical ORC does not offer advantages for heat source temperatures lower than 130 °C, though it improves the net and the gross power output above 130 °C.

Furthermore heat source temperature is a crucial factor for the choice of the working fluid. R227ea shows highest power output at heat source temperatures lower than 140 °C, RC318 at 150 °C and 160 °C. At higher temperatures R236fa (170 °C) and isobutane (180-190 °C) performed best. Highest improvements within one fluid are achieved at 130 °C with an increase in gross power of up to 13.4 % for R227ea as a working fluid. Though one has to consider that the increase in gross power output is linked to an increase in heat transfer capacity $UA$. Therefore an economic analysis for a geothermal power plant in Germany is carried out. The heat source temperature used in this case study is 150 °C which is typical for the Upper Rhine Graben located between Frankfurt and Basel. The effect of investment costs of the power plant per kW$_{el}$ as well as interest rate on payback period and cashflow are calculated.

![Figure 2: Exergetic efficiency for a brine temperature of 150 °C (a) and 190 °C (b)](image)

CONCLUSION

Supercritical mode of operation is one way to increase ORC efficiency in geothermal applications. The present work investigated eight fluids and compared efficiencies as well as power output of subcritical and supercritical ORC. Supercritical ORC shows a potential of up to 15.4 % increase in power output though the heat source temperature is a crucial factor for possible improvements. All in all one has to consider thermodynamic as well as economic aspects to allow for a holistic comparison of both technologies.

REFERENCES


CXFD ANALYSIS OF A VARIABLE GEOMETRY EJECTOR CONCEPT FOR COOLING USING R600A AS WORKING FLUID

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ABSTRACT

It is well known that optimal ejector geometry depends strongly on the operating conditions. Any change in the upstream and downstream pressures (temperatures) would require a different ejector to provide optimal cooling performance. One of the most important geometrical features of an ejector is the area ratio ($r_A$) between the constant area section and the primary nozzle throat. In the present work, a spindle in the primary nozzle is proposed in order to actively control $r_A$ such that a single ejector can be brought to operate in critical mode for different operating conditions.

The present theoretical analysis was carried out for a 1kW rated capacity ejector using R600a as working fluid. Operating conditions were selected in a range that would be suitable for an air-conditioning application using evacuated tube solar collectors as heat source. Generator and condenser temperatures ranged from 80 – 90 ºC and 25 – 41 ºC respectively. The base line ejector geometry was designed by a constant pressure mixing ejector model for operating temperatures of 10ºC, 37ºC and 80ºC for the evaporator, the condenser and the generator respectively. R600a was selected as a working fluid because it is environmental friendly and has been recently demonstrated to operate with high COP in ejector cooling systems. The entrainment ratio ($\lambda$) was used as performance indicator.

The developed CFD model was based on the compressible axisymmetric Navier-Stokes equations where turbulent behaviour was treated by the RNG formulation of the Reynolds averaging principle. The simulations were carried out using ANSYS/FLUENT, considering adiabatic ejector walls and real fluid properties for the refrigerant. The computational domain consisted of a structured mesh with approximately 20 thousand control volumes. A large number of simulations were performed for the set of operating conditions indicated above and for different spindle tip positions from closed to fully open. Figure 1 show the influence of the spindle position on the primary flow rate for 3 different generator temperatures. It can be clearly seen from the figure that the primary flow rate can be controlled by the spindle tip travel.

Ejector performance curves were constructed for $\lambda$ and the critical operating conditions were identified as shown in Figure 2. This diagram can be used to find optimal spindle tip position for highest entrainment ratio. E.g. for a condenser pressure of 400kPa (~30ºC) (dashed arrows) and generator temperature of 80ºC, optimal spindle tip position would be approximately 3.2 mm from fully closed, resulting in $\lambda$=0.38 against 0.26 that would be the performance of a constant geometry ejector designed under the same operating conditions.
Figure 1: The influence of spindle position on the primary flow rate in the ejector for 3 different generator temperatures.

Figure 2: Critical ejector operation as a function of condenser pressure ($p_c$) for different generator temperatures ($T_g$) and spindle tip positions (1 – 4 mm).
SURVEILLANCE SYSTEM HEAT TRANSFER ENHANCEMENT USING SINGLE-PHASE FLOW WITH NANOFLUID

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ABSTRACT

Following today’s needs for improvement on heat transfer, new technologies and innovative solutions must be found in order to meet current requirements for both active and passive thermal control. Independently on the application, there has been substantial growth on the heat fluxes that need to be dissipated, which require different approaches from designers as well as manufacturers of heat exchangers and heat management devices, especially those designed for defense purposes. With the increase of heat dissipation needs, conventional designs are not suitable due to several factors such as operation in hostile environments, high density of electronics that need their temperature to be controlled, which require innovative designs. Thus, bigger equipments are required to meet the expectations of heat transfer that directly influence not only their final price and the required investment needed to be done by the consumer, but also may find a serious impact on the footprint necessary to install such equipment in reduced areas. When such constraint is found, new and innovative solutions must be found. Combination of different working fluids has already been used in the past as well as different configurations of heat exchangers to meet the requirements of heat dissipation. However, in most cases, high costs are involved which make the design neither economically nor technically viable.

One important approach that has been presented over the last years that could help on the enhancement of heat transfer capability is directly related to the use of nanofluids. Nanofluids are composed of a regular base fluid (like water, for example) with an addition of solid nanoparticles with sizes below 100 nm mixed at a fraction of the fluid’s mass. Nanofluids were early presented by Xuan and Li (2000) and Xuan and Roetzel (2000) and have been extensively investigated over the last years with important contributions to several areas, like aerospace, electronics and industry, even though the application on those areas is still in the beginning. Some examples for aerospace applications have been presented focusing on using nanofluids in single-phase and passive thermal control systems using heat pipes, as presented by Riehl and Santos (2011). In this case, the heat source temperature can be kept at lower levels when compared to regular working fluid application, which leads to the possibility of using the nanofluid to dissipate higher levels of heat. Upon using single-phase cooling systems, the heat transfer can be enhanced as the nanoparticles addition to the base fluid is highly influenced by the presence of more liquid. When the application shows that more vapor is presented, the nanoparticles addition to the base fluid do not show important enhancement on the heat transfer process. Since in aerospace the use of passive thermal control devices is important and their sizes and footprint are a major issue due to limited area available, the use of nanofluids present to be an important approach to enhance the heat transfer capability of heat pipes and loop heat pipes systems, which has already been proven in laboratory conditions, but still requires a full qualification program for space application.

An important application for today’s needs for heat dissipation is related to surveillance systems designed for defense purposes. As more compact and powerful defense equipments are necessary, higher heat fluxes need to be properly addressed efficiently. Considering that the systems usually operate with single-phase flow, more efficient thermal management devices need to be designed and the use of nanofluids can be a powerful solution. A specific design for a surveillance equipment has been conceived to operate in hostile environments where the ambient temperatures can range from +5 to +50 °C. In this case, a single-phase thermal control loop has been designed with potential use of nanofluids, which would present a forced circulation using a pump to move the working fluid throughout the architecture of the circuit to remove heat from the electronic components, rejecting to the environment by a fan cooling system. Details of the surveillance
equipment and its architecture are considered proprietary information and cannot be disclosed. However, the system needs to reject up to 50 kW of heat to the environment, circulating the working fluid in a diameter of 11 mm through 45 m of tubes with a required temperature difference between the inlet and outlet of the fan cooling system of 15 °C, for a total internal volume of 8 liters. Figures 1a and 1b present some results for the pressure drop and heat transfer coefficients, respectively, on a comparison between the use of pure water and the addition of copper nanoparticles at different concentrations \( f \) by mass percentage of working fluid in the system. It is clear that as the concentration increases, the pressure drop also increases up to 32% for \( f=20\% \), as more solid nanoparticles are present in the system and the pump needs to overcome the extra resistance, as the transport properties are changed with the addition of the nanoparticles. However, the increase of the heat transfer coefficients is also clear and can represent a gain around 12% for the same \( f=20\% \) which cannot be neglected.

![Figure 1](image)

**Figure 1: Analytical results for (a) pressure drop and (b) heat transfer coefficient.**

Even though the use of nanofluids can represent a gain on the heat transfer coefficient and thus a new option for the design of heat management devices, it must be carefully considered regarding the pumping requirements. However, apparently the use of nanofluids present gains that can lead to reducing the size of heat exchangers while directly improving their heat dissipation. A correct combination between the base fluid and the solid nanoparticles must be carefully considered in order to avoid chemical reactions and incompatibilities issues.

**REFERENCES**


COMPARATIVE ANALYSIS OF ORC SYSTEM DESIGN USING VARIOUS WORKING FLUIDS FOR MEDIUM GRADE HEAT RECOVERY

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The ANR-EESI\textsuperscript{1} project Enerco_LT\textsuperscript{2} is a waste heat recovery project that aims to reduce energy consumption in industrial gas production sites, by producing electrical power from exothermic processes discharges at low and medium temperatures. Two promising thermal sources, consisting in i) dry gas flow at 165°C and ii) moist gas flow at 150°C with dew point at 60°C, are then investigated.

The optimized recovery solution can be identified by an appropriate system design, by an adequate working fluid and under suitable operating conditions. Combined exergy analysis and pinch optimization have been performed to identify the potential of various working fluids to overcome the overall irreversibility within various system conceptions, i.e. simple cycles, cascading cycles and regenerative cascading cycles either subcritical or transcritical.

Two recovery solutions based on 10K pinch are detailed and compared for each resource by Ayachi et al, 2011. In this paper, various fluids are investigated. The results (Figures 1 and 2) show that global exergy efficiency is strongly related to the working fluid critical temperature. This leads to distinguish an appropriate range of fluid critical temperature that would most suits the requirements of the energy recovery conditions. Besides, a sensitivity study on the appropriate range evolution according to the pinch value (25K, 10K and 5K) has been performed and indicates that it is appropriate to associate the fluid choice to the pinch value.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure1.png}
\caption{Dry resource: System performance – 10K Pinch}
\end{figure}

\textsuperscript{1} Agence Nationale de la Recherche.
\textsuperscript{2} ENERGY ReCOvery from Low Temperature heat sources.
The solutions based on pure fluids are subject to constraints predominately technological and environmental.

For the dry effluent case (Figure 1), R-236fa is proved as the best solution but cannot be used in practice, due to its high Global Warming Potential value. Other alternatives such as R-1234yf could raise the global efficiency but at the expense of supercritical operating conditions or pinch diminishing. These constraints could be sidestepped when using working blends. Some binary blends are investigated and seem to bring promising results regarding system performances. Temperature glide can contribute according to the molar blend composition to reduce the exergy destruction into the condenser.

![Diagram showing fluid critical temperature and global exergy efficiency](image)

**Figure 2: Moist resource: System performance – 10K and 5K Pinch**
(topping cycle working fluid is indicated at the right and bottoming cycle working fluid is varied according to the critical temperature)

For the moist effluent case (Figure 2), single cycles, even transcritical, show low global exergy efficiencies. Two-cycle cascade is identified as the appropriate recovery design from the moist resource. The tendency curves indicate that global exergy efficiency would raise as well as the bottoming cycle fluid critical temperature is lower than the dew point of the moist resource. Carbon dioxide could be the best candidate but it should be noted that the optimum corresponds to a supercritical pressure far exceeding the fixed limit of 70 bar.

Heat regeneration contribution is not very significant since the global exergy efficiency raise is less than 1.5%.

For each type of heat source available at medium temperature, the appropriate design and working fluid(s) are identified taking into account technological and environmental constraints.

**REFERENCES**

THERMODYNAMIC COMPARISON OF ORGANIC RANKINE CYCLE WITH LIQUID FLOODED EXPANSION AND WITH SOLUTION CIRCUIT

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ABSTRACT

The competing pressures of energy security and climate change mitigation have spurred the development of Organic Rankine Cycle (ORC) technology to harness low grade heat (80°C – 200°C) as a viable energy source. Organic Rankine Cycles differ from traditional steam Rankine Cycles in the use of an organic liquid as opposed to steam as the working fluid. The term ORC is also applied generally to any Rankine cycle with a low-grade heat source. Due to the low-temperature heat sources, the theoretical (Carnot) efficiency limit is itself relatively low. Therefore, achieving cycle efficiencies as close to the Carnot limit as possible is important for ensuring the economic feasibility of the technology. In this view, the thermodynamic performance of two novel modifications to a traditional Organic Rankine Cycle is investigated: Organic Rankine Cycle with Liquid Flooded Expansion (ORCLFE), and Organic Rankine Cycle with Solution Circuit (ORCSC).

The ORCLFE involves “flooding” the expansion device with a liquid that is in thermal equilibrium with the primary working fluid, while simultaneously expanding the primary working fluid through the same device. This process is readily accommodated using a positive displacement expander such as a scroll or screw machine. The ultimate result is that the expansion of the primary working fluid takes place in a more isothermal manner. The ORCSC, on the other hand, employs azeotropic mixture of the primary working fluid and an absorbent characterized by a large boiling point difference. This enables the separation of the more volatile component (the primary working fluid) in the vapor phase from the solution in the liquid phase. The working fluid vapor then flows through the expansion device, whereas the liquid absorbent gives rise to the solution circuit. Due to the cycle configuration, there is significant regenerative potential that can be used to enhance the overall system performance.

Figure 1 shows the Second Law efficiency (ratio of thermal efficiency and Carnot efficiency) for varying source temperatures of a conventional ORC with regeneration between the expander exhaust and pump discharge streams. In comparing a conventional ORC with an ORCSC and ORCLFE in Figures 1 and 2 below, it can be seen that purely on thermodynamic considerations, a simple ORC using water as the working fluid is more efficient than most combinations of working fluids used in the enhanced cycles. However, practical concerns such as expander exhaust quality and pressure, expander pressure ratio, and other issues related to capacity should be considered in addition to the thermodynamic results.

Due to the challenges associated with achieving the theoretical performance in a simple ORC, the practical advantages of the ORCLFE and the ORCSC render them both attractive for different reasons. In addition to marginally superior thermodynamic performance, the ORCLFE allows for a variable expansion ratio on a positive displacement machine. This may be beneficial when the temperature of the heat source varies significantly or when it is too costly to build an expander with the desired intrinsic expansion ratio. The expander exhaust quality concerns are eliminated with the ORCLFE due to the more isothermal nature of the expansion process. A boost in efficiency also results when a regenerator is employed between the expander exhaust and pump discharge streams (Woodland et al., 2010). The ORCSC, on the other hand, allows for temperature glides that can be adjusted over a wide range to match the temperature profiles of the heat source and heat sink media. Capacity control in the ORCSC can be achieved simply by changing the concentration of the working fluid mixture. The ORCSC also offers significantly lower working pressures than use of the pure absorbate as the working fluid due to the solution in the cycle (Krishna et al., 2011).
Figure 1: Second Law efficiency as a function of source temperature for an ORC with a regenerator

Figure 2: Second Law efficiency as a function of source temperature for the ORCLFE and ORCSC (Ammonia-Water and CO$_2$-Acetone mixtures pertain to the ORCSC. All others pertain to the ORCLFE).

In the above context, the decision of a truly optimal choice of cycle configuration and working fluid may depend strongly on the application. However, this work primarily seeks to compare the thermodynamic potential of the cycles, along with a qualitative assessment of the tradeoffs associated with the implementation of each the cycles and working fluids. The design space is explored using different primary working fluids and absorbents under a range of heat source temperatures. The results can be used as a guide to identify areas that need further development efforts. Furthermore, three experimental test stands consisting of a traditional ORC, an ORCSC, and an ORCLFE have been constructed in order to provide further insight into the feasibility of these technologies and validate the results of this study.

REFERENCES


EFFECT OF LITHIUM BROMIDE ON CHEMICAL HEAT STORAGE MATERIAL WITH EXPANDED GRAPHITE

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INTRODUCTION

The increase of energy consumption in every society leads to serious environmental problems, especially global warming and natural resource depletion. In minimizing those problems, more efficient ways to use heat have to be explored. A chemical heat pump (CHP) is one of the desired heat storage systems using a reversible chemical reaction. In this study a magnesium oxide/water (MgO/H_2O) CHP was examined. It can store heat at around 350°C and can recover waste heat efficiently. It is based on the following equation (Kato, 1996),

\[ \text{MgO (s) + H}_2\text{O (g)} \leftrightarrow \text{Mg(OH)}_2(s) \quad \Delta H^0 = -81.02 \text{ kJ mol}^{-1} \quad [1] \]

The right direction in Eq.1 is hydration corresponding to heat output operation for CHP and the left one is dehydration corresponding to the heat storage operation. The main aim of this study was to enhance the reactor performance by introducing lithium bromide (LiBr) into heat storage material used for MgO/H_2O CHP due to hydrophilic ability of LiBr. Expanded graphite (EG) was known to enhance the thermal conductivity of the storage material. Then a composite of LiBr, EG and magnesium hydroxide (Mg(OH)_2), which named as EML, was synthesized. The kinetic studies of dehydration and hydration of synthesized material were investigated by using thermobalance (TG). The kinetic parameter as the reaction rate constant of EML composites at dehydration was analyzed.

EXPERIMENTAL

In this work, Mg(OH)_2 powder and EML (\( \alpha = 0.10, 0.01 \) under \( w = 1:1 \)) composites were employed, in where \( \alpha \) and \( w \) indicate mixing molar ratio between Mg(OH)_2 and LiBr, and mass ratio between Mg(OH)_2 and EG, respectively. EML composite was prepared by an impregnation method which allows particles of Mg(OH)_2 were dispersed well on the graphite matrix. Sample initially charged in a cell made by platinum (Pt) with inner diameter of 7.5 mm and height of 10 mm which is placed in TG. The sample was dried at 110°C for 60 min under 100mL min\(^{-1}\) of Ar flow to remove physically adsorbed water in sample. Then temperature was raised at 20°C min\(^{-1}\) to 300°C and it was dehydrated at 300°C for 120 min. After dehydration completion, it was hydrated at 110°C for 130 min with mixed gas flow of 35mL min\(^{-1}\) of Ar flow as carrier gas and of an amount of water supplied by a water micro-feeder for corresponding to vapor pressure of 57.9 kPa. After stopping water vapor supply, sample was kept at 110°C under Ar flow of 100mL min\(^{-1}\) to estimate amount of adsorbed water in LiBr. Hydrated water was desorbed under this condition with weight decrease.

REACTED FRACTION

The sample mass change during the dehydration/hydration was caused by the movement of water vapor. By heating the sample by electric furnace, the sample starts to lose its mass due to Mg(OH)_2 dehydration, in order words, water vapor as reaction product is released from the sample. Temporal change in mass is measured continuously with temperature. In contrast, mass of sample is increased at MgO hydration. Thus, the mole reacted fraction, \( x \)% is calculated by:

\[ x = 1 + \frac{\Delta m_{\text{H}_2\text{O}}}{M_{\text{Mg(OH)}_2}/M_{\text{H}_2\text{O}}} \cdot 100 \quad [2] \]

in where \( \Delta m_{\text{H}_2\text{O}} \) [g] is mass change of the sample caused by reaction, \( m_{\text{Mg(OH)}_2} \) [g] is initial charged mass of Mg(OH)_2 and \( M \) [g mol\(^{-1}\)] are molecular masses of Mg(OH)_2 and H_2O, respectively.
RESULTS AND DISCUSSIONS

Thermogravimetric Analysis of EML composites and Mg(OH)$_2$ powder

Figure 1 shows the mole reacted fraction changes of dehydration and hydration of EMLs with two different mixing molar ratios under constant mixing mass ratio, and Mg(OH)$_2$ powder.

EML composite with $\alpha = 0.10$ had higher hydration reactivity than that of Mg(OH)$_2$ powder. Hydration rate of EML composite was accelerated with increase of mixing molar ratio. Both EMLs had higher reaction rate for dehydration in comparison with Mg(OH)$_2$ powder. The degree of dehydration rate depended on amount of LiBr in the composite. It was assumed that dehydration of Mg(OH)$_2$ proceeds as first order reaction and the rate equation was proposed as following,

$$-\frac{dx}{dt_d} = k_d x$$  \[3\]

where $k_d$ [min$^{-1}$] and $t_d$ [min] are a dehydration rate constant and time, respectively. Eq.3 can be integrated by initial condition ($x = x_0$ at $t_d = 0$) as following,

$$ln x = k_d t_d + ln x_0$$  \[4\]

In Figure2 the dehydration rate analysis is represented the dehydration time $t_d = 0$ corresponds to the time which the cell temperature reached at 300°C in Figure 1. Initial conversion as $x_0$ is assumed to be 0.9 in order to remove effect of physically adsorbed water. Relation between $t_d$ and $ln x$ for dehydration of EML composites and Mg(OH)$_2$ powder was shown in Figure 2; markers represent measured values and solid line shows linear regression fit of those values, respectively. The plot indicates well linear trend according to Eq. 4. The slope of the linear plot corresponds with $k_d$. It was thought that variation of $k_d$ value was involved to changes in the mixing molar ratio. The experimental value of $k_d$ for EML($\alpha = 0.10$, $w = 1:1$) was 0.24 min$^{-1}$ at 300°C.

CONCLUSIONS

LiBr was employed as reactivity enhancer for a MgO/H$_2$O CHP. Dehydration rate of EML composites was greater than that of Mg(OH)$_2$ powder and then, degree of dehydration depended on amount of LiBr in the composite. It was expected that EML composite could be utilized as a new heat storage material for relatively low temperature heat utilization at 200~250°C.

REFERENCES

DYNAMIC PERMEABILITY OF COMPOSITE REACTANT OF CALCIUM CHLORIDE AND EXPANDED GRAPHITE

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ABSTRACT

The composite reactant of calcium chloride and expanded graphite is a solid material used in chemical heat pumps. Its advantage compared to untreated inorganic salt-reactants is that it can enhance heat transfer in reactor beds without decreasing mass transfer. However, in the operation cycles of heat pumps which work at low-temperature using water as a refrigerant, permeability of working fluid is still the rate-controlling factor because of the low vapor pressure incidental to such cycles. In this study, the dynamic permeability of the composite reactant has been examined experimentally in order to obtain information for the optimum preparation condition of the composite reactant.

EXPERIMENT

Experiment has been conducted using an experimental set-up which consists of a grass vessel (with a diameter of 30 mm and a thickness of 17 mm), an evaporator and a differential pressure gauge. The composite reactant was prepared according to the procedure described in our previous study (Fujioka et al., 2008) and packed in the vessel. The vessel was put in a constant-temperature bath and water vapor was introduced in the vessel to progress hydration reaction. The pressure drop of the water vapor across the sample layer was measured at 30°C. The amount of water absorbed in the sample was evaluated by the change in the water level of the evaporator.

Permeability of each sample was obtained from the measured pressure drop using the Darcy’s law expressed by the following equation.

\[ 0 = -\nabla P + \rho g - \frac{\mu v_s}{K} \]

where \( \nabla P \) is the pressure gradient across the sample [Pa/m], \( \rho \) is the gas density [kg/m³], \( g \) is the acceleration of gravity [m/s²], \( \mu \) is the gas viscosity [Pa·s], \( K \) is permeability [m²], and \( v_s \) is the surface velocity [m/s].

Measurements using nitrogen gas instead of water vapor have also been carried out.

TIME VARIATION OF PERMEABILITY AND REACTION PROGRESS

Figure 1 is time variations of reaction progress, which expressed by the molar ratio of water to calcium chloride in the composite reactant, \( n \), and the permeability, \( K \), for the case the initial void fraction, \( \varepsilon_0 \), is 0.84. Both \( n \) and the permeability decreases with time. After 20 minutes, increase in \( n \) becomes slow, but the permeability decreases with almost the same rate as in the first stage. It may indicate that the hydration rate decreases because of the rapid decrease in the permeability after 20 minutes. The similar tendency was observed in all the samples examined.

PARTICLE SIZE

Figure 2 shows the relationship between pressure drop and void fraction obtained by the measurements using nitrogen gas as well as calculated value using the the Kozeny-Carman equation.

\[ \Delta P = 180 \left( \frac{1 - \varepsilon}{\varepsilon} \right)^2 \frac{\mu v_s}{d_p^2} L \]

where \( \Delta P \) is the pressure drop, \( d_p \) is particle diameter and \( L \) is the sample thickness.
The calculated value shows good agreement when the particle diameter is assumed to be 120 μm as shown in figure 2 by a solid line. The composite reactant before packed in the vessel has approximately 200 μm of diameter and one to two mm of length. The calculation result that the particle diameter of 120 μm can fit the measured pressure drop suggests that the composite reactant breaks into pieces around 120 μm. The change in particle size would be one of the reasons of decreasing permeability.

![Figure 1: Time variations of reaction progress and permeability ($\varepsilon_0=0.84$)](image1)

![Figure 2: Relationship between pressure drop and void fraction](image2)

**REFERENCES**

EXPERIMENTAL TESTING OF A COATED ADSORBER

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INTRODUCTION

Intensification of the heat transfer quality in adsorbers is a key-factor for development of dynamically efficient adsorption refrigeration and heat pump systems. The most recent approach to design advanced adsorbers, is the use of large surface area heat exchangers coated by a thin layer of adsorbent material. Different coating methods have been proposed, including in-situ zeolite crystallization (Bauer J. et. al, 2009), adhesive coating (Dawoud B. et. al, 2007) and dip coating processes (K. Okamoto et al., 2010). Experimental studies on full-scale coated adsorbers returned encouraging results, especially in terms of reduced adsorption cycle time (Dawoud B., 2012), that can be translated to an elevated specific power.

The aim of this work was the experimental testing of two dip-coated adsorbers with different amount of zeolite and the comparison with a granular adsorber, both concepts employing the same type of heat exchanger. Dynamic performance of the adsorbers were measured by a lab-scale testing facility under realistic operating conditions of an adsorption chiller.

EXPERIMENTAL

\textit{Figure 1a, b} shows a realized coated adsorber, which was prepared \textit{via} dip coating method, starting from a liquid solution of SAPO34 zeolite and a proper binder. The coating was applied on a compact finned flat-tube aluminium heat exchanger. Two coated adsorbers were prepared, having the metal to adsorbent mass ratio $M_{\text{met}}/M_{\text{ads}} = 3.8$ and 2.2. The corresponding the coating thickness was 0.3 mm and 0.5 mm, respectively. The so-realized adsorbers were tested by means of a testing facility consisting of an intermittent single bed adsorption unit (power of about 1 kW) and a test bench.

The adsorption unit consists of a single testing chamber where the adsorber is placed, connected to an evaporator and a condenser, operating with an intermittent mode. The adsorption unit is connected to the test bench specifically devoted to provide external heating and cooling energy for adsorption unit operation as well as to automatically control and manage the system. The test bench is able to reproduce a wide range of operating conditions and to measure the performance obtained in terms of Coefficient Of Performance (COP) and delivered power both in heat pumping and cooling mode.

Moreover, for comparison purpose, a granular adsorber was realized by integrating loose grains of the same adsorbent (0.3-0.35 mm in size) in the same heat exchanger type ($M_{\text{met}}/M_{\text{ads}} = 1.2$).
Figure 2a, b. The coated adsorber: a) detailed view, b) the adsorber inside the testing chamber.

RESULTS

Experiments were carried out under the following operating conditions: $T_{ev} = 15^\circ$C, $T_{con} = 35^\circ$C, $T_{ads} = 35^\circ$C $T_{des} = 75 – 90^\circ$C. COP and delivered power were evaluated in cooling mode. Tab I resumes the performance achieved at $T_{des} = 90^\circ$C in terms of cooling COP, adsorbent mass and volume Specific Cooling Power; $SCP_{M}$, $SCP_{V}$. Typical experimental error was less than 10%.

<table>
<thead>
<tr>
<th>Tabel 1: Performance measured at $T_{ev} = 15^\circ$C, $T_{con} = 35^\circ$C, $T_{ads} = 35^\circ$C, $T_{des} = 90^\circ$C</th>
<th>$M_{met}/M_{ads}$</th>
<th>Cooling COP</th>
<th>Mass Spec. Cooling Power $SCP_{M}$ [W/kg adsorbent]</th>
<th>Vol. Spec. Cooling Power $SCP_{V}$ [W/dm$^3$ adsorber]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Granular adsorber</td>
<td>1.2</td>
<td>0.30-0.41</td>
<td>202- 295</td>
<td>81-115</td>
</tr>
<tr>
<td>Coated adsorber 1</td>
<td>3.8</td>
<td>0.24-0.35</td>
<td>405- 703</td>
<td>51-88</td>
</tr>
<tr>
<td>Coated adsorber 2</td>
<td>2.2</td>
<td>0.25-0.39</td>
<td>260- 401</td>
<td>73-95</td>
</tr>
</tbody>
</table>

Primarily, all adsorbers performed rapid adsorption cycles (3-15 min), thanks to the selected heat exchanger, which is compact, lightweight and offers high surface area. The coated adsorbers attractively delivered a Mass Specific Cooling Power $SCP_{M}$ of several hundred watts per kilogram of dry adsorbent, confirming that the coating process allows to enhance the heat transfer rate. COP and $SCP_{V}$ values measured for the coated adsorber 1 were lower than those measured for the granular adsorber, due to the less favorable metal to adsorbent mass ratio. Differently, the coated adsorber 2, performed better thanks to the higher zeolite loading, indicating that the proper selection of the coating characteristics is of primary importance for optimization of adsorption dynamics in adsorption systems.

REFERENCES

EXTENDED ABSTRACT

In the recent years, the ever increase in global energy demand and greenhouse gas emissions in the recent years necessitates for more and more utilization of sustainable energy sources such as solar energy, biomass and waste heat energy for power and cooling applications. The use of low and medium temperature thermal sources such as thermal solar energy, biomass or biogas to produce refrigeration and mechanical power is well known; absorption refrigeration cycles and Kalina cycles both using ammonia/water, are good examples of these systems. Absorption cycles for mechanical power such as the Kalina cycle consist of components used also in absorption refrigeration cycles as absorbers, generators thus the combination of both systems is not only possible but also presents some advantages from the technical performance perspective with a higher efficiency than the separated production of power and refrigeration which means a better use of primary energy resources.

The objective of this paper is to systematically study a modified basic absorption cycle configuration using ammonia as refrigerant and water as absorbent is to simultaneously produce refrigeration and mechanical power using renewable resources such as thermal solar energy, biomass or biogas with a single system able to cover different demand profiles of refrigeration and power. This kind of cycles is the result of the combination of absorption refrigeration and Kalina power cycles. Figure 1 shows the modified ammonia-water absorption cycle for combined power and cooling applications.

Figure 1. Modified ammonia-water cycle for combined power and cooling applications

Goswami, 1995 proposed and Sadrameli analyzed [2007] the cycle intended primarily for power generation while simultaneously producing a cooling output. The combined cooling output is gained from a heat exchanger following the expander. The expander exhaust is cooled by expanding the vapour to sub-ambient temperatures. Application of low heat-source temperatures below 200ºC is one of the characteristics of this cycle. Since this cycle employs the turbine exhaust
gases through a cooler transferring sensible heat to the chilled water, the refrigeration output is relatively small. Erickson et al., 2004 proposed an ammonia-water absorption cycle configuration which produces power and refrigeration interchangeably. The dual function cycle consists of a heat recovery unit, desorber, recuperator, absorber, turbine, generator, condenser and evaporator. The power and absorption cycle use the same absorption unit that improves the economics of recovering low grade energy. A novel form of the basic absorption cooling cycle has been presented by Ziegler, 2007. The cooling and power cycle derived from a double absorption chiller. It is estimated that with this cycle configuration it is possible to produce 1.2 MW of cooling and 100 kW of power from a single source of thermal energy of 1 MW. The main advantage of these configurations is the possibility to use low grade heat such as solar energy or waste heat.

The modified cycle shown in Figure 1 enables to utilize the waste heat temperatures up to 150°C for their operation. Ratio between power output and cooling can be controlled depends upon the demand on power/cooling. Depending on the requirement of refrigeration to power ratio, one can adjust the Stream 17 temperature. Assuming the heat source is hot water, it will go in series through super heater, then Generator 2 or Dist.2, then to generator 1 or Dist.1. For higher refrigeration requirement, more heat input is required in Dist-1 by partly bypassing heat to Dist.2. Otherwise, power generation gets priority, and the residual heat is converted to refrigeration. The variables - streams 9, 11 and 29 will vary depending on power to refrigeration ratio. They can be controlled with flow control valves on stream 9 and 29, allowing stream 11 to float.

The simulation result indicate for a particular operating condition at the Heat Source temperature of 150°C, sink temperature 40°C, evaporator temperature of about -8°C a net power output of 29.3 kW and the cooling capacity of 214.5 kW could be attained with the first law thermal efficiency of the combined cycle is 32.16 %. This paper will describe the simulation results of the combined cycle for various power to refrigeration ratio and its impact on the thermal efficiency of the system.

ACKNOWLEDGMENT

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THERMAL EFFICIENCY COMPARISON OF CONVENTIONAL, SUPERHEAT, REHEAT AND RECUPERATIVE ORC CONFIGURATIONS USING AMMONIA, R134A AND R245FA

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ABSTRACT

A thermal efficiency comparison of wet, isentropic and dry working fluids for 4 different organic Rankine cycles (ORCs) configuration are given. The conventional (CNV), superheat (SP), superheat reheat (SPR) and recuperative (RV) are tested for ammonia, R134a and R245fa using IPSEPPO simulation software. Waste heat temperature and mass flow rate are assumed respectively 200°C and 1 kg/s. The evaporating is 95°C for CNV and RV cycles. While superheated temperature is set to 130°C for SP and SPR ORCs. The 30°C condensation temperature is used for all of the configurations. The simulation shows that CNV and SP ORCs are applicable for ammonia, R134a and R245fa. Whereas SPR configuration is able to be utilized well for both R134a and R245fa, however, RH ORC can only be employed effectively for R245fa. It is found that R245fa in SPR configuration provides highest the thermal efficiency (12.6%), followed by ammonia in SP configuration (12 %) and R134a in SPR (11.8%). This result shows that the types of working fluids in different ORCs configuration have different implication on thermal efficiency.

INTRODUCTION

In recent years, there has been an increasing interest in low grade waste heat recovery using organic Rankine cycles (ORCs). ORC is a system that converts low grade waste heat to electrical power. Low grade waste heat has temperature range 90 to 230°C US department of energy (2008). Generally, ORC system consists of an evaporator, a turbine, a condenser and a pump. A common challenge is to select a working fluid which capable to produce high thermal efficiency ($\eta_{th}$). Currently, there are several common type of ORC, namely conventional (CNV), superheated (SP), superheat reheat (SPR) and recuperative (RV). While working fluids can be classified into wet, isentropic and dry (Hung et al. 1997). Wet fluid is characterized by negative saturation vapour slope, as can be seen in Figure 2. Isentropic fluid has near vertical saturation vapour slope, as shown in Figure 4. Dry fluid is has positive saturation vapour slope, shown in Figure 6. The aim of this paper will give an account of wet (ammonia), isentropic (R134a) and dry (R245fa) working fluids in CNV, SP, SPR and RV ORC configuration.

A numerous studies have attempted to compare ORC configurations. For example, (Chen et al. 2011) compared conventional and superheat ORCs, (Aneke et al. 2012) investigated two sources of waste heat using conventional, dual source and two turbine ORCs configurations, in recovering low grade waste heat in food industry, (Saleh et al. 2007) evaluated conventional, superheat, conventional with internal heat exchanger and supercritical ORCs. However, these studies did not clearly mention what is the required condition to apply ORCs configurations in certain type of working fluids. Therefore, the main question address in this paper is what type of working fluid which provides optimum efficiency for each configuration and what are the requirements to select each ORCs configuration. The contribution of this paper is a decision tree to choose the ORC configuration which has the optimum thermal efficiency for ammonia, R134a and R245fa.

RESULT

In order to compare different configurations with different working fluids, several assumptions are made. It is shown in Table 1. It is been decided to set the evaporating temperature to 95°C and condensing temperature to 30°C for all of the configurations.

\[\text{References}\]

43
### Table 1: Assumption for ORC calculation

<table>
<thead>
<tr>
<th>Assumption for all ORC configurations</th>
<th>95°C</th>
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<tbody>
<tr>
<td>Evaporating temperature</td>
<td>95°C</td>
</tr>
<tr>
<td>Waste heat temperature source</td>
<td>200°C</td>
</tr>
<tr>
<td>Waste heat energy content</td>
<td>121 kW</td>
</tr>
<tr>
<td>Minimum temperature approach in evaporator, condenser and internal heat exchanger</td>
<td>10°C</td>
</tr>
<tr>
<td>Condensing temperature</td>
<td>30°C</td>
</tr>
<tr>
<td>Waste heat mass flow rate</td>
<td>1 kg/s</td>
</tr>
<tr>
<td>Cooling water temperature source</td>
<td>10°C</td>
</tr>
<tr>
<td>Pump and turbine mechanical efficiency</td>
<td>80%</td>
</tr>
<tr>
<td>Pump and turbine isentropic efficiency</td>
<td>100%</td>
</tr>
<tr>
<td>For SP and SPR ORC configurations</td>
<td></td>
</tr>
<tr>
<td>Superheating temperature</td>
<td>130°C</td>
</tr>
</tbody>
</table>

Figure 1 shows the thermal efficiency comparison for 4 different configurations using ammonia, R134a and R245fa

![Figure 1: Thermal efficiency comparison](image)

**REFERENCES**


NEW COMPOSITE "MG(OH)$_2$-VERMICULITE": A PROMISING CANDIDATE FOR STORAGE OF MIDDLE TEMPERATURE HEAT

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INTRODUCTION

Heat storage is a necessary stage to harmonize heat production and heat consumption processes (Dincer et al, 2001). The PCM heat storing typically offers the storage density of 100-500 kJ/kg (Cabeza et al, 2011). Thermochemical energy storage (TES) is based on reversible chemical reactions which allow absorption of heat in the course of decomposition process, like Mg(OH)$_2$ $\leftrightarrow$ MgO + H$_2$O (Kato, 2007). This reaction was suggested for storage of heat with $T > 350^\circ$C (Kato et al, 2005). To improve TES dynamics we confined Mg(OH)$_2$ in the pores of expanded vermiculite and studied it for TES application.

MATERIALS AND METHODS

The composite preparation included two steps. The first step was an impregnation of the vermiculite pores with a magnesium nitrate aqueous solution; the second step was a deposition of magnesium hydroxide inside the pores. The hydroxide content in the composites was up to 67 mas. % (sample HC). This composite was decomposed at 300$^\circ$C for 3 hs and then rehydrated by saturated vapor at 60$^\circ$C and P(H$_2$O) = 0.2 bar for 1 h (sample RHC). The synthesized materials were investigated by DSC, TG, BET, SEM, X-ray diffraction and chemical analysis.

RESULTS AND DISCUSSION

The hydroxide is stabilized inside the pores as irregular sticks of app. 1 μm in size across flats of the base and 10-100 μm in length (Figure 1a). Despite of this macroscopic texture, the zone of coherent scattering is much smaller, just 5-20 nm, as estimated by XRD data (Figure 1b).

Figure 1: SEM photographs of composite HC (a) and weight loss TG curves for samples H, HC and RHC at $P$(H$_2$O) = 22.2 mbar and the heating rate 5 K/min.

The temperature of decomposition of the confined hydroxide is found to be some 50$^\circ$C lower than that of the bulk one as revealed by both TG (Figure 1b) and DSC data. Significant acceleration of decomposition of the confined hydroxide as compared with an authentic Mg(OH)$_2$ has been revealed (Figure 2b). The maximal heat storage capacity measured by DCS is 700 kJ per 1 kg of the composite that is just twice lower than that for a bulk magnesium hydroxide.
Figure 2: a - TG curves for samples H, HC and RHC at $P(\text{H}_2\text{O}) = 22.2$ mbar and the heating rate 5 K/min, b - comparison of the isothermal curves of Mg(OH)$_2$ decomposition in bulk (H) and in the pores (HC) at $T = 270^\circ$C.

CONCLUSIONS

New composite material for storage and transformation of middle temperature (260-350$^\circ$C) heat has been synthesized by precipitation of magnesium hydroxide in the pores of expanded vermiculite. The composite can store heat with the temperature potential 40-60$^\circ$C lower than an authentic Mg(OH)$_2$. Possible reasons of this reduction are a) acceleration of the decomposition of the confined hydroxide, and b) shifting the decomposition equilibrium to lower $T$ due to chemical modification or pore-size effects. The heat storage density of 700 kJ per 1 kg could be of practical interest. Further study of the new composite material for storage of middle temperature heat is of high interest for further increase of the storage density and harmonizing the reaction temperature with the temperature level of heat to be stored.

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ACKNOWLEDGEMENTS

The authors thank Drs. A.N. Salanov and T.A. Kriger for recording SEM and XRD data.
A HIGH-EFFICIENCY SOLAR RANKINE ENGINE WITH ISOTHERMAL EXPANSION FOR USE IN DESALINATION

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ABSTRACT

The solar-Rankine cycle has been considered as an alternative to photovoltaic devices. It is particularly interesting for special applications such combined heat and power (CHP), or desalination (García-Rodríguez and Delgado-Torres 2007). Though the principle of the solar Rankine cycle is well known with several examples reported in the literature, today relatively few systems are operational aside from the large (>1 MW) concentrating solar power plants situated mostly in the US and Spain. The solar-Rankine cycle suffers from relative complexity and sometimes uses toxic fluids (e.g. toluene).

Here we explore the feasibility of a variant of the Rankine cycle which, like the idealized Stirling cycle, uses an isothermal expansion to achieve a theoretical efficiency close to the Carnot limit. Generation of steam inside the power cylinder obviates the need for an external boiler and high-pressure steam piping. The device is suitable for slow moving applications and is of particular interest for driving a batch-desalination process giving high recovery of fresh water.

DESCRIPTION OF THE CYCLE

Figure 1 shows a schematic layout of the proposed ‘boiler-less’ Rankine engine and Figure 2 shows the $T$-$s$ diagram for the cycle. Operation may be described as follows with reference to the numbered states (1−6). The power cylinder is heated externally by a heat transfer fluid (e.g. oil) which is supplied from solar collectors. Water (1) injected into the cylinder evaporates on contact with the walls, thus expanding at constant temperature and performing work against the piston. Due to further heat transfer from the cylinder walls, the water vapour continues to expand at constant temperature $T_h$ until it reaches state (3). On the return stroke of the piston, the steam exits via a valve, and passes through a heat exchanger whereby its temperature is reduced to $T_c$ which is the temperature inside the condenser. Before being returned to the injector, the condensed water (5) is pressurized by the feed pump and preheated in the heat exchanger.

Figure 1: System schematic  
Figure 2: $T$-$s$ diagram

ANALYSIS

The table below shows the ideal cycle efficiency $\eta$ in comparison to that of the Carnot efficiency $\eta_c$ for $T_h = 200–350^\circ C$ and $T_c = 50^\circ C$. Note that the work theoretical work ratio is high (>0.99).
In real implementations of the cycle, due to the finite rate of heat transfer, the upper temperature will be lower than $T_h$ and isothermal expansion may not be fully achieved. To assess the rate of heat transfer achievable in practice, we carried out an analysis of radiant heat transfer assuming that the cylinder wall is a grey surface with emissivity of 0.57 at a temperature of 250°C. The optical absorptivity of water vapour in the infra-red region was modeled using SpectralCalc® software. For example operating parameters of: piston area 0.035 m$^2$, stroke length 0.228 m, cylinder volume 0.008 m$^3$, and expanding steam pressure decreasing from 9 bar to 1.8 bar, it was found that the absorptivity and thus the absorptance (the absorptive power per unit area) decrease as the piston advances; but the increase in the surface area of heat transfer results in a steep increase in the energy emission rate from the cylinder wall, and consequently results in a net increase in the energy absorption rate of the steam (Figure 3). For this example, it was concluded that radiant heat transfer alone could supply the required heat in 37 s.

**Conclusion and Further Work**

The isothermal Rankine cycle provides a slow moving machine with high thermodynamic efficiency and work ratio. Because it is slow moving the capital cost will be high; however, the high efficiency will lead to economy in the area of solar collector needed – so the overall economics need to be studied. Currently we are exploring application to a batch desalination process whereby the power piston will be coupled to a pump piston to pressure saline water for reverse osmosis (Davies 2011).

**References**


NEW WORKING FLUIDS FOR A COMBINED POWER AND COOLING CYCLE

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EXTENDED ABSTRACT

A thermodynamic cycle was developed for the simultaneous production of power and cooling from low-grade heat sources such as solar thermal, geothermal and industrial waste heat. This cycle, first proposed by Goswami, combines the Rankine and absorption refrigeration cycles to provide the dual outputs (Demirkaya et al, 2011). Power is the primary goal of this cycle. To the best of the authors knowledge, so far the Goswami cycle analysis was performed with ammonia/water (NH$_3$/H$_2$O) mixture as a working fluid except in one case where organic fluid mixtures were used (Vijayaraghavan et al, 2005).

Lithium nitrate (LiNO$_3$) and Sodium Thiocyanate (NaSCN) have been used successfully in absorption refrigeration systems to replace H$_2$O as absorbent for NH$_3$ to overcome the drawbacks associated with NH$_3$/H$_2$O working fluid. Currently, Ionic liquids (ILs) are also suggested as suitable absorbent for NH$_3$ in absorption refrigeration systems. The main advantages of these absorbents (LiNO$_3$, NaSCN and ILs), is no rectification of the vapour generated in the desorber. This work extends the application of the Goswami cycle to working fluids consisting of different absorbents (LiNO$_3$ and NaSCN) for NH$_3$ to replace the conventional absorbent (H$_2$O). The thermophysical properties of NH$_3$/LiNO$_3$ and NH$_3$/NaSCN mixtures have been reported by our research group previous studies (Libotean et al, 2007, 2008 and Chaudhari et al, 2011) and used here in the cycle analyses.

CYCLE DESCRIPTION

Figure 1 shows a schematic layout of the Goswami cycle with NH$_3$/H$_2$O as a working fluid. In this cycle a mixture of NH$_3$ and H$_2$O, basic solution, pumped to a high pressure (state 2) and partially boiled in the desorber. The desorbed vapour (state 4) consists of mostly NH$_3$. A rectifier is used to increase the concentration of NH$_3$ in the vapour, by partially condensing H$_2$O out of the vapour from the desorber. The resulted purified vapour (state 6) may be superheated (state 7) and expanded in an expander to low pressure and temperature (state 8). This low-temperature vapour can be used to obtain cooling (states 8-9). The weak solution, in NH$_3$ from the desorber (state 10) is passed through a solution heat exchanger to recover heat from it, and throttled back to the absorber (states 10-12). The vapour and weak solution are used to regenerate the basic solution in the absorber (state 1) along with the rejection of heat from the cycle. The condensed liquid in the rectifier is also re-mixed with basic liquid solution coming from the absorber after recovering heat from the weak solution coming from the desorber in this schematic.

Figure 1: Combined power and cooling cycle with NH$_3$/H$_2$O as a working fluid
FINDINGS AND SUMMARY CONCLUSIONS

The application of the new absorbents (LiNO$_3$ and NaSCN) to a combined power and cooling cycle has been studied using several performance indicators. The results show that NH$_3$/LiNO$_3$ and NH$_3$/NaSCN can be used as a suitable alternative for NH$_3$/H$_2$O in a combined cycle. The cycle with NH$_3$/LiNO$_3$ gives better performance than NH$_3$/H$_2$O cycle (Figure 2), not only because of higher efficiency values, but also because of no need of rectifier. The NH$_3$/LiNO$_3$ cycle operates at a lower heat source temperature than the conventional NH$_3$/H$_2$O cycle (Figure 2). The power and cooling outputs of the cycles per 1kg/s of basic solution at a heat source temperature of 125°C are 24.7kW and 4.7kW for NH$_3$/LiNO$_3$ cycle, 14.3kW and 2.9kW for NH$_3$/NaSCN cycle, 22.6kW and 3.4kW for NH$_3$/H$_2$O cycle for solution cooled rectifier and 22.6kW and 3.4kW for NH$_3$/H$_2$O cycle with water cooled rectifier. The corresponding driving heat inputs to the cycles are 168.8kW, 133.1kW, 192.2kW and 228.0kW, respectively.

![Figure 2: Simulation results for the effect of heat source temperature on effective first law efficiency for NH$_3$/LiNO$_3$, NH$_3$/NaSCN and NH$_3$/H$_2$O cycles](image)

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DESIGN AND TEST OF A SMALL SOLAR ADSORPTION COOLING SYSTEM IN MEDITERRANEAN CLIMATE

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ABSTRACT

During last decades the energy consumption needed for air conditioning in residential and tertiary sectors is growing up. This behaviour could be related to the increasing of living standards and comfort demands as well as the innovative building architectural characteristics [Henning, 2007].

In this scenario one of the most promising technology is the solar cooling capable to reduce the electricity consumption due to the air conditioning demand. Actually solar cooling systems are driven by solar thermal energy, instead of electricity [Cacciola and Restuccia, 1999] and they can be considered environmentally friendly because the refrigerants used have no Global Warming Potential (GWP).

Nowadays main existing applications of solar cooling systems are oriented towards high power cooling systems (P > 70 kW), for example in industrial applications, due to the limited number of commercially available thermally driven chillers with nominal cooling power in the range between 3 and 20 kW.

Within the thermally driven chiller technology, adsorption one seems to have the highest potential for small size applications. Indeed such devices can be thermally driven by low temperature heat source (60-90°C), utilize safe and not polluting materials and do not present moving parts. Moreover few examples of adsorption chillers having nominal cooling power lower than 10 kW are already available on the market.

In this work the design, building and results of test of an adsorption solar cooling plant for air conditioning of a small office located at the CNR ITAE in Messina are presented.

The system realized consists of a solar thermal collector field, a backup gas boiler, a hot water storage as thermal buffer and a commercial small adsorption chiller. In Figure 1 three different views of the main components of the system are shown.

Figure 1: Three views of the main components of the cooling system: backup gas boiler, vertical heat storage, adsorption chiller and solar thermal collector field are shown.
The presented solar cooling system has been designed by means of a numerical model implemented in TRNSYS [Vasta et al, 2009]. The results of the several simulation performed have been used to fit system performance to the cooling load of small office in Mediterranean climate. To this aim and to enhance the adsorption chiller performance, high temperature radiant panels have been installed into the small office. Finally, in order to monitor the main functional parameters and to evaluate the performance of the system, several sensors have been installed (e.g. temperature, humidity, solar radiation). The system has been monitored during the hottest summer days in Messina and performance have been calculated. The results demonstrated that the system is able to maintain excellent indoor comfort conditions, as shown in Figure 2. Picture shows the inlet and outlet temperature of the radiant panel, the inlet air temperature and the external ambient temperature. As shown a temperature of 25 °C is maintained also in severe summer ambient conditions (32°C) using chilled water at 18°C. Nevertheless the solar fraction calculated results to be lower than expected probably due to the high condensation temperature, typical of Sicilian summer days, which heavily affects the performance of the adsorption chiller.

Figure 2: Evolution of the outdoor temperature, indoor temperature, inlet and outlet temperature of the radiant panels.

REFERENCES
GRAVIMETRIC VERSION OF LARGE TEMPERATURE JUMP METHOD: EFFECT OF THE FAM-Z02 GRAIN SIZE ON ADSORPTION DYNAMICS

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INTRODUCTION

Technologies and systems based on adsorption heat transformations represent nowadays a fascinating option to meet the growing worldwide demand of space heating and air conditioning. Nevertheless still considerable efforts must be made in order to enhance the performance, aiming at competing with common used electrical systems as well as with absorption machines. For this purpose, an intelligent design of an adsorption machine should be firstly focused on a convenient choice of the adsorbent material by a comprehensive analysis that takes into account both thermodynamic and dynamic aspects. Thermodynamic requirements for an optimal adsorbent under specific operating conditions were extensively studied while the study of adsorption kinetics is still an open topic. Recent studies have been carrying out by a “Large Temperature Jump” (LTJ) method (Aristov et al, 2008) since it closely imitates the isobaric stages of an adsorption machine. This memory presents the preliminary results of kinetic measurements carried out by a new gravimetric experimental setup developed at ITAE-CNR that is a useful extension of the common LTJ method.

DESCRIPTION OF EXPERIMENTAL SET UP

Figure 1 shows the new experimental measurement system mainly composed of four components: a gravimetric weighing unit, the vacuum chamber (chamber 1) where the weighing unit is placed, the evaporator/condenser chamber (chamber 2) and the hydraulic heating/cooling circuit.

Figure 1: Schematic and view of the experimental setup for gravimetric LTJ measurements

The weighing unit is the core component and consists of an aluminum plate working as a support for the sorbent material as well as heat exchanger. The amount of water desorbed/adsorbed is directly weighted by two load cells on which the plate is placed. The bottom of the metal plate is hydraulically connected to two thermo-cryostats (TCR1 and TCR2) simulating the thermal source and sink needed to reproduce the isobaric heating/cooling stages. The weighing unit is able to test samples with the mass ranging between 4 g and 250 g with the accuracy of ±0.1 g. This large sample mass is the main advantage of the new TG version as compared with the standard LTJ scheme. Surface temperature of adsorbent sample and the metal plate were measured respectively by a remote IR-sensor and a T-type thermocouple.
EXPERIMENTAL ACTIVITY

The activity aimed to evaluate the effect of the size of the FAM Z02 adsorbent on the adsorption dynamics. Two configurations have been tested: a) monolayer of loose grains, and b) constant sample mass equal to 26.1 g. The grain size was ranged as 0.15 - 0.2, 0.3 - 0.35, 0.425 - 0.6, 0.7 - 0.8, 1.0 - 1.18, and 2 mm. For all tests adsorption runs were conducted for temperature drops of 65°C - 30°C and 65°C - 40°C while the pressure was maintained at 11.5 mbar. Figure 2 showing the dimensionless uptake \( \chi(t) = w(t)/w_\infty \) where \( w(t) \) is the current uptake and \( w_\infty \) is the final uptake, points out that for the monolayer configuration, the adsorption kinetics becomes faster for smaller grains.

![Figure 2: Adsorption kinetic curves as dimensionless uptake \( \chi(t) \) measured at 65°C - 30°C.](image)

All curves were satisfactorily described by an exponential equation \( \chi(t) = 1 - \Delta \chi \exp(-t/\tau) \) where \( \tau \) is a characteristic adsorption time. The initial adsorption rate (at \( t < 30 \) s or \( \chi < 0.2 \)) does not depend on the grain size while at longer time, the rate is gradually reducing, and the power corresponding to \( \chi = 0.8 \) depends on the total amount of water adsorbed. Table 1 reports the characteristic times \( \tau \) and \( \tau_{0.8} \) as well as the specific cooling powers \( W_{\text{max}} \) and \( W_{0.8} \) measured for the monolayer configuration.

Table 1: Characteristic times \( \tau \) and specific cooling powers for the monolayer configuration.

<table>
<thead>
<tr>
<th>( d, \text{mm} )</th>
<th>( m, \text{g} )</th>
<th>( \Delta m, \text{g} )</th>
<th>( \tau, \text{s} )</th>
<th>( \tau_{0.8}, \text{s} )</th>
<th>( W_{\text{max}}, \text{W/g} )</th>
<th>( W_{0.8}, \text{W/g} )</th>
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<tr>
<td>0.300-0.350</td>
<td>9.0</td>
<td>1.8</td>
<td>43.5</td>
<td>70</td>
<td>12.8</td>
<td>8.7</td>
</tr>
<tr>
<td>0.425-0.600</td>
<td>13.0</td>
<td>2.3</td>
<td>47.6</td>
<td>78</td>
<td>8.8</td>
<td>6.9</td>
</tr>
<tr>
<td>0.700-0.800</td>
<td>26.1</td>
<td>5.0</td>
<td>125</td>
<td>197</td>
<td>4.4</td>
<td>3.0</td>
</tr>
<tr>
<td>1.0-1.180</td>
<td>32.0</td>
<td>5.7</td>
<td>143</td>
<td>244</td>
<td>3.6</td>
<td>2.2</td>
</tr>
<tr>
<td>2.0</td>
<td>42.1</td>
<td>8.5</td>
<td>333</td>
<td>480</td>
<td>2.7</td>
<td>1.3</td>
</tr>
</tbody>
</table>

REFERENCES

DESIGN OF A SMALL-SCALE ORGANIC RANKINE CYCLE ENGINE USED IN A SOLAR POWER PLANT

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Under the economical and political pressure due to the depletion of fossil fuel and global warming potential, it is necessary to develop more sustainable techniques to provide electrical power. In this context, medium and large scale Organic Rankine Cycle solar power plants appear to be a promising solution because of their good efficiency, robustness and acceptable economical probability. However, only a few ORC solar power plants are in operation today, but technical literature indicate that several demonstration projects are under development. The spreading out of that technology requires, among others, a good knowledge of control strategies of the entire plant comprising the field of parabolic trough collectors, the ORC engine and thermal storage systems.

The present project aims at designing, building and testing a small scale ORC solar power plant (a few kWe) in order to define and optimize control strategies that could be applied to larger systems.

The paper presents the design step of the solar power plant, and more specifically the ORC engine. This design is defined based on simulation models of the ORC engine and on the experience gained in the Thermodynamics laboratory in the field of testing small-scale ORC systems. The design accounts for technical limitations such as allowed operating ranges and technical maturity of components.

The paper first presents the architecture of the envisioned solar plant. The choice of the different technologies of components is justified. Simulation models of components (scroll expander, plate and fin-and-tubes heat exchangers, etc.) and of the whole system are then presented. Based on those models, parametric studies are conducted in order to optimize the sizing and the operating conditions of the system and to select the most appropriate working fluid. Also, the relevance of using two expanders in series is discussed and the control of the ORC engine is investigated. Simulations indicate that ORC engine efficiency close to 12% can be reached for evaporating and condensing temperatures of 140°C and 35°C.
INTRODUCTION

In the Netherlands, the main part of the residential energy demand consists of space heating and domestic water heating. During the summer period the available solar energy is sufficient to cover the tap water heating demand for houses equipped with solar thermal collectors. However, during winter, the heating demand exceeds the solar supply (Figure 1). Solar energy can be harvested in summer to fulfill the heat demand in winter by means of a seasonal heat storage system. However, for a passive house of 110 m² with a winter heat demand estimated at 6 GJ, seasonal heat storage in a traditional water tank would require a tank volume above 40 m³ which is way too large to be integrated in individual houses. Therefore, more compact heat storage technologies have to be found.

Figure 1: Solar energy consumption related to the solar irradiation all over a year in Netherlands

Figure 2: Design of the 6m³ packed bed reactor optimized for seasonal heat storage
SEASONAL SORPTION HEAT STORAGE DEVELOPMENT AT ECN

ECN proposes a compact heat storage system based on the reversible reaction of water vapor with a ThermoChemical Material (TCM). TCMs present a high energy density in packed bed (1 GJ/m³) which result for the abovementioned 6 GJ heat demand in a much more compact storage of around 6 m³. This is suitable for integration in individual houses. Interesting TCM materials for seasonal heat storage are salt hydrates, which are available in large quantity at low cost, are environmentally friendly and can take up and release heat under the conditions of seasonal heat storage. The heat storage design focuses on a packed bed through which air is flown (Figure 2). During summer, the salt hydrate materials can be dehydrated by means of hot air provided by solar thermal collectors with a temperature below 150°C. During winter, the dehydrated salt is rehydrated with moist air from the ambient. To improve the reactor performance, the vapor pressure of the incoming air is increased to 13 mbar by means of the evaporation of water with 10°C heat taken from a borehole.

SORPTION MATERIALS DEVELOPMENT

In an screening of TCMs carried out at ECN, several salt hydrates such as MgSO₄·7H₂O and MgCl₂·6H₂O showed promising performances. However, tests in a lab scale reactor indicated that these materials need further optimization in terms of stability and water uptake. MgSO₄·7H₂O has a low water uptake as shown in Table 1 but a rather good material stability during the cycling process. On the other hand, MgCl₂·6H₂O shows adequate heat release for domestic applications but overhydration of the material reduces its efficiency over the storage lifetime (Table 1).

Table 1: Performances of MgSO₄·7H₂O and MgCl₂·6H₂O as TCM in packed bed reactor

<table>
<thead>
<tr>
<th>Materials</th>
<th>MgSO₄·7H₂O</th>
<th>MgCl₂·6H₂O</th>
</tr>
</thead>
<tbody>
<tr>
<td>Theoretical bed energy</td>
<td>7 → 1 (150°C)</td>
<td>6 → 2: (130°C to avoid HCl vapor)</td>
</tr>
<tr>
<td>density (50% porosity)</td>
<td>1.10 GJ/m³</td>
<td>0.97 GJ/m³</td>
</tr>
<tr>
<td>Hydration @ 13 mbar</td>
<td>1 → 2</td>
<td>2 → 6</td>
</tr>
<tr>
<td>Experimental energy</td>
<td>0.4 GJ/m³</td>
<td>≈ 1 GJ/m³</td>
</tr>
<tr>
<td>density (50% porosity)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature rise</td>
<td>ΔT = + 6°C</td>
<td>ΔT = + 60°C</td>
</tr>
</tbody>
</table>

Optimization of material properties is required in order to develop an adequate sorption material. With this objective, ECN started in November 2011 a research project, financed by the Dutch ADEM program, which aims at improving the system by designing a new optimized material which remains efficient and stable over the lifetime of the heat storage. The objective of the first phase of this project is to characterize the water vapor sorption process (heat- and vapor transport) and the changes in these TCM storage materials (structure, composition) at the levels of a single crystals and grains.

STORAGE OPTIMIZATION

The storage design is very important for the performance. A prototype lab scale thermochemical open sorption heat storage system with a storage capacity of 20 liters was built and tested at ECN. As TCM material, MgCl₂·6H₂O was used. On discharging, the prototype sorption storage was found to be able to deliver heat at 60°C for over 20 hours.

At the conference, results will be presented both on the performance of the lab-scale TC storage system, as well as on the detailed TC materials characterization.
APPLICATION OF THE DATA RECONCILIATION METHODOLOGY IN A DOUBLE-STAGE ABSORPTION CHILLER DRIVEN AT TWO INPUT TEMPERATURES

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In this paper is analysed experimentally the performance of a double/single-effect absorption chiller driven simultaneously by heat at two temperatures levels. When heat from a low temperature heat source such as that from low/medium temperature thermal solar collectors is available, the chiller can work as a single-effect chiller and as double-effect chiller when additional heat at high temperature needs to be provided to complement the low temperature heat. Also a data reconciliation and gross error detection method is applied to the raw measurements to calculate the chiller performance.

The Insitut für Festkörperphysik und Technische Physik from the Technische Universität München developed a water/lithium bromide heat pump prototype with 500 kW of heating power. This prototype operated as heat pump from 1993 until 2001 at the German Aviation Museum in Munich. Later it was moved to the CREVER-URV facilities in Tarragona where it was repaired and adapted to be used as double-effect chiller driven at two temperature levels: one level driven by solar thermal energy, and the other level driven by natural gas. This machine has the particularity that the intermediate pressure generator-condenser group can reverse its function and work as evaporator-absorber. Because of this, the chiller can be operated at different modes: double-effect chiller, single-effect chiller, single-double effect chiller, and double-lift heat pump. The main components of the chiller are presented in Figure 1. The separation of water vapour from the lithium bromide solution at generator G2 takes place by means of a heat input at 160ºC, and condensation of this water vapour is done in condenser C2. Similar processes occur in a intermediate pressure casing where vapour generated in GA1 is produced by the heat of condensation rejected in C2 plus the heat provided by the solar collectors at around 90ºC. The cooling effect is produced in the low pressure casing where water condensed in CE1 and C2 is evaporated in E0, and then absorbed in A0 by the solution that comes from G1. Heat is rejected at temperatures around 35ºC in the absorber A0 and condenser C1 by means of a cooling tower.

Measurements can be redundant if the amount of measurements available when monitoring a system is greater than the degrees of freedom of the system. In this case we have more information than the necessary to estimate the performance of the system. Data reconciliation methodologies are used in order to overcome these problems. Through data reconciliation we adjust the measurements as close as possible to the measured value, and at the same time we obtain calculations that are in agreement with the laws of conservation. In this study we used the Lagrange multipliers method to reconcile the flow rates, temperatures, pressure, and concentration measured at the internal and external circuits of the machine. The mass and energy balances of the main components of the chiller are used as restrictions for the data reconciliation problem. When gross errors are present the data reconciliation process tends to spread these errors over all the measurements. Therefore, measurements with gross errors need to be detected and eliminated from the data reconciliation problem. In this study through a Modified Iterative Measurement Test, variables m₂₁, Vₐₐ and Tₐₐ were identified as gross errors. Those measurements were eliminated and data reconciliation was applied again. Table 1 presents the measured values, the reconciled values before and after gross error detection (GED) eliminating the measurements with gross errors, during one of the test when the machine was working as double effect chiller. It can be seen that most of the measurements were adjusted as close as possible from the original measurement in order to satisfy the mass and energy balances. In the case of m₂₁ and m₁ this adjustment is large with respect to the measurement errors of the sensors and these measurements are considered to contain gross errors. In this Table are given also the heat duties calculated with the reconciled
values without gross errors. In these conditions the achieved COP was of 1.28. In conclusion the applied Lagrange multipliers method for data reconciliation can be a valid methodology for complex non-linear systems as is the case of absorption chillers if a gross error detection procedure is also applied.

Figure 1: Schematic of the double-stage heat pump/absorption chiller

Table 1: Comparison of raw and reconciled data and final reconciled heat duties

<table>
<thead>
<tr>
<th>Variable</th>
<th>Measured Value</th>
<th>Value (Before GED)</th>
<th>Value (After GED)</th>
<th>Variable</th>
<th>Measured Value</th>
<th>Value (Before GED)</th>
<th>Value (After GED)</th>
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<tbody>
<tr>
<td>$t_1$ (°C)</td>
<td>30.5</td>
<td>30.6</td>
<td>30.6</td>
<td>$t_{28}$ (°C)</td>
<td>9.1</td>
<td>8.6</td>
<td>9.1</td>
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<tr>
<td>$t_2$ (°C)</td>
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<td>60.7</td>
<td>$t_{30}$ (°C)</td>
<td>80.6</td>
<td>80.8</td>
<td>81.2</td>
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<tr>
<td>$t_3$ (°C)</td>
<td>76.8</td>
<td>76.9</td>
<td>76.9</td>
<td>$t_{33}$ (°C)</td>
<td>83.2</td>
<td>83.5</td>
<td>83.3</td>
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<tr>
<td>$t_5$ (°C)</td>
<td>40.2</td>
<td>40.1</td>
<td>40.1</td>
<td>$m_1$ (kg/s)</td>
<td>1.62</td>
<td>1.03</td>
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<td>$t_{13}$ (°C)</td>
<td>112.6</td>
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<td>112.8</td>
<td>$m_{23}$ (kg/s)</td>
<td>10.9</td>
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<td>6.1</td>
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<tr>
<td>$t_{14}$ (°C)</td>
<td>129.7</td>
<td>129.7</td>
<td>129.7</td>
<td>$m_{23}$ (kg/s)</td>
<td>16.7</td>
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<td>$t_{15}$ (°C)</td>
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<td>75.8</td>
<td>$m_{23}$ (kg/s)</td>
<td>10.7</td>
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<td>$t_{21}$ (°C)</td>
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<td>159.8</td>
<td>160.2</td>
<td>$m_{20}$ (kg/s)</td>
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<td>8.6</td>
<td>9.2</td>
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<td>$t_{22}$ (°C)</td>
<td>155.7</td>
<td>156.1</td>
<td>155.7</td>
<td>$HP$ (mbar)</td>
<td>558</td>
<td>557</td>
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<td>$t_{23}$ (°C)</td>
<td>25.1</td>
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<td>25.0</td>
<td>$IP$ (mbar)</td>
<td>48</td>
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<td>$t_{24}$ (°C)</td>
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<td>$LP$ (mbar)</td>
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<td>$X$ (%)</td>
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<td>$t_{27}$ (°C)</td>
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Heat Duties (kW)

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<th>$Q_{C2}$</th>
<th>80</th>
<th>$Q_{GA1}$</th>
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<tr>
<td>$Q_{G12}$</td>
<td>115</td>
<td>$Q_{HEX1}$</td>
<td>55</td>
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<tr>
<td>$Q_{HEX2}$</td>
<td>86</td>
<td>$Q_{EO}$</td>
<td>147</td>
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<tr>
<td>$Q_{CE1}$</td>
<td>79</td>
<td>$Q_{AO}$</td>
<td>182</td>
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ACKNOWLEDGMENTS

The authors would like to acknowledge the funding for this project provided by the Ministry of Science and Innovation of Spain, Plan Nacional de I+D+i, ref. ENE2009-14182.
THE EFFECT OF ENVIRONMENTAL FRIENDLY REFRIGERANTS ON THE PERFORMANCE OF AN EJECTOR COOLING SYSTEM

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INTRODUCTION

The search of new refrigerants, environmentally favorable and appropriate for ejector cooling systems, is still a matter of study. In this work, the effect of refrigerants 410a, 507 and 245fa on the system operation is studied.

The system studied is shown in Figure 1, whose ejector is of the gas-gas type and its main parts: the main nozzle, the suction and mixing chambers and the diffuser, are shown in Figure 2. Also, the diameters for the throat and exit of the main nozzle -d\textsubscript{t} and d\textsubscript{e}; the mixing chamber -d\textsubscript{m}; the diffuser exit -d\textsubscript{d}, as well as the main nozzle separation distance -L\textsubscript{ln} and the mixing chamber length -L\textsubscript{nm}, are indicated.

Figure 1: Ejector cooling system configuration.

The ejector operation is given by a one-dimensional mathematical model with secondary fluid choking, which was developed by Lu, 1986, making the following assumptions:

- Steady state performance;
- Fluids behave as perfect gases;
- Flow along the main nozzle is isentropic;
- Friction losses in mixing chamber are considered as pipeline minor losses;
- Null main nozzle separation distance;
- Complete fluid mixture at mixing chamber

On the other hand, this model was validated with the experimental results of a multi-geometry ejector which uses R123 and is applied for cooling, Yapici \textit{et al} 2008. The main nozzles employed had different separation distances. As result, the calculation of the actual Mach number at the main nozzle’s exit and the friction factor, were carried out. With the ideal and actual Mach numbers of the main nozzle’s exit, its efficiency was evaluated and its dependency with \( \psi \), the main nozzle ratio of discharge to throat areas, was found. As well, the friction factor, corresponding to the mixing chamber losses, has a dependency with L\textsubscript{ln}, the dimensionless main nozzle separation distance given by the ratio of this distance to the mixing chamber length. These two ejector parameters are plotted against L\textsubscript{ln} and \( \psi \) as Figure 3 shows.
ANALYSIS OF RESULTS

A parametric study is carried out for a unitary capacity of the cooling system, in which the generation, condensation and evaporation temperatures are varied. The found ejector parameters are: $\phi$, the area ratio of the mixing chamber to the main nozzle throat; $U$, the entrainment ratio given by the mass flow ratio for the fluids secondary to primary; and $\eta_E$, the ejector efficiency. For the system its coefficient of performance COPs, is calculated, whose plots against the temperatures of generation, condensation and evaporator are shown.

The obtained results represent potential design points of an ejector cooling system which operates under secondary flow choking. As well, one and only one $\phi$ value corresponds to each combination of the of generation, condensation and evaporation temperatures. Therefore, this $\phi$ value corresponds to a specific ejector. Finally for the three refrigerants 410a, 507 and 245fa, a comparison of the results for the ejector and cooling system is carried out.

REFERENCES


DYNAMIC OPTIMISATION OF ADSORPTIVE HEAT TRANSFORMERS: EXPERIMENT AND MODELING

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corresponding author, e-mail: aristov@catalysis.ru

INTRODUCTION

Heat driven adsorption machines with small- and medium-scale capacity have recently been developed in the world, and many of them have now passed over from the prototype stage to small serial production (Jakob and Kohlenbach, 2010). Despite the significant progress achieved, still there is a big room for improvement of the AHT technology, first of all, the Coefficient Of Performance (COP) and the specific power (SP). The main lines of this amelioration are

- Thermodynamic harmonization of the adsorbent used with a particular AHT cycle (Aristov, 2007). This would allow reduction of the thermal coupling between the adsorbent and the external heat source, decrease of appropriate irreversibility, and the COP increase.
- Dynamic optimization of an integrated unit "Adsorbent – Heat Exchanger" (Ad-HEx) (Aristov, 2009) to enhance heat and mass transfer processes and to increase the SP-value.
- Enhancing cycling stability of adsorbents under real AHT conditions (Henninger et al, 2011).

Here we summarize recent progress in dynamic optimization of AHT and give appropriate recommendations. Two main concepts of Ad-HEx configuration are considered: a) loose adsorbent grains contacted with HEx fins, and b) the fins coated with a consolidated adsorbent layer. Here we focus on the former one which has been realized in several prototype (Tamainot-Telito et al, 2009) and commercial (Saha et al, 2001) AHT units. Dynamic experiments have been performed by a novel Large Temperature Jump (LTJ) method (Aristov et al, 2008) which imitates conditions of isobaric stages of AHT. A new TG version of this method is presented in HPC-12 by Sapienza et al. For a monolayer configuration, a mathematical model (Okunev et al, 2008) has been used to study transient and steady-state AHT modes and effect of the adsorption isobar shape on AHT dynamics. For the multi-layer case, a model based on a COMSOL Multiphysics® has been developed by Freni et al, 2012. Several recommendations followed from the LTJ study were checked with AHT prototypes in ITAE-CNR (Aristov et al, 2012) and the Warwick University (Veselovskaya et al, 2010).

MAIN FINDINGS AND CONCLUSIONS

The main experimental findings on isobaric ad-/desorption dynamics initiated by a fast temperature drop/jump of HE fins are as follows (for water, methanol and ammonia as working fluids):

- the initial stage of ad-/desorption process follows exponential kinetics (Figure 1);
- in the monolayer case, the adsorbent grain size has a dominant effect on adsorption dynamics;
- in the multi-layer case, the sorption dynamics is invariant with respect to the ratio $S/m$;
- a concave adsorption isobar is profitable for desorption, while a convex one - for adsorption;
- the isobaric adsorption stage of AHT cycle is usually slower than the isobaric desorption stage;
- the driving force for adsorption is the temperature difference between the plate and the grains.

![Figure 1: a - Uptake curves of water adsorption on loose Fuji silica grains: a - monolayer of grains of different size; b – n layers of 0.8-0.9 mm grains. Lines – exponential approximation.](image-url)
Each of these findings is clearly illustrated and all-round discussed in the paper. E.g., the characteristic sorption time \( \tau = \frac{\Delta H \Delta m \Delta q}{\alpha S \Delta T_{\text{max}}} = \left( \frac{\Delta H \Delta q}{\Delta T_{\text{max}}} \right)^\prime \frac{m}{\alpha S} \) depends on the thermodynamic parameters \((\Delta H, \Delta q, \Delta T_{\text{max}})\) which can be taken from the cycle diagram, and the parameters of Ad-HEx, namely, \( \alpha \) – the heat transfer coefficient, \( S \) – the heat transfer surface, \( m \) – the adsorbent mass. Indeed, it is found that the maximal specific power \( W_{\text{max}} \sim 1/\tau \) is a linear function of the ratio \( S/m \) (Figure 2a) that is valid at \( S/m < 5-6 \) for grains of 0.2-0.7 mm size. The typical \( \alpha \)-values are found to be 100 W/(m K). As the Ad-HEx configuration in the LTJ experiments is favorable for both heat and mass transfer, the absolute SP-value obtained was much larger than that for known AHT prototypes. Therefore, one can expect significant increase in the SP-value due to improvements in engineering of the Ad-HEx and AHT unit in whole, rather than as a result of optimizing the adsorbent itself.

Figure 2: a - Maximal specific power vs. the ratio \( S/m \) for silica Fuji RD grains of various size between 0.2 and 0.8 mm; b – temporal evolution of radial distribution of the vapor pressure inside the grain during water desorption. \( r = 0 \) - the grain center, \( r = 1 \) - the grain surface.

Mathematical modeling was used to investigate a coupling between the heat and mass transfer for the monolayer case. At \( t = 0 \), the driving force for heat transfer is maximal while the driving force for mass transfer equals zero. Then, the temperature gradient generates the pressure gradient inside the grain which is the driving force for mass transfer. It is proportional to the derivative \( dP(r)/dr \) near the grain external surface (at \( r = 1 \), Figure 2b). After a short transient period (app. 10 s), this slope gradually decreases with time (Figure 2b). Under this steady-state mode, the driving forces for heat and mass transfer are interactively reducing while the desorption proceeds.

ACKNOWLEDGEMENTS


REFERENCES

HYDRATION AND DEHYDRATION OF SALT HYDRATES AND HYDROXIDES FOR THERMAL ENERGY STORAGE - KINETICS, ENERGY RELEASE AND CYCLABILITY

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INTRODUCTION

Thermal energy storage drives the change towards the use of renewable energy resources and energy efficiency. Thermal storage can be used for waste heat and solar thermal usage. Solar thermal systems work at temperatures between 120°C (space heating and hot tap water) and up to 600°C (CSP plants), where thermal energy storage is a key component [Gil 2010]. In comparison to most commonly used water or molten salt storage, thermochemical storage materials provide ten times or higher storage capacities per mass or volume and the ability to store heat for longer times without the need of insulation [Fisch 2005]. Recently published results state MgCl$_2$ * 6 H$_2$O and CaCl$_2$ * 6 H$_2$O to be useful for space heating and hot tap water [van Essen 2009] and Ca(OH)$_2$ for higher temperature levels [Schaube 2009]. We reported earlier about our studies in MgCl$_2$ * 6 H$_2$O as pure and composite materials [Opel 2011]. In this paper we discuss detailed TGA/DSC investigations of hydration and dehydration reaction of salt hydrates and hydroxides. The main questions/issues addressed in this paper are the kinetics of water release and uptake as well as the released power in dependence on water vapor pressure and temperature. For the most promising materials we also studied the cyclability.

METHODS AND MATERIAL

We used a TGA/DSC1 (Mettler) to analyze heat fluxes and changes in sample mass during hydration and dehydration experiments. The Mettler TGA/DSC1 provides the opportunity to introduce two different reaction gases into the TGA/DSC oven in addition to the purge gas (figure 1). CaCl$_2$ and silica gel-dried nitrogen has been used as purge gas at a flow rate of 50 mL/min. The gas flow was controlled by a gas box providing two independent mass flow controllers connected to the TGA/DSC control. In addition to the purge gas flow, we used a separate nitrogen flow humidified via a thermostatised gas bubbler flask as the reaction gas. The purge gas flow was merged with reaction gas in oven to get a specified atmosphere.

The humidity of the mixture gas was measured at the oven outlet and used for calculation of the water vapour partial pressure inside the TGA/DSC oven. This setup allowed an investigation of hydration characteristics in addition to the dehydration of the thermochemical storage materials. Investigation on the cycling stability of the thermochemical storage materials were done by multiple measurements of the integrated hydration heat fluxes of one sample during consecutive dehydration/hydration cycles. The temperature program used for dehydration of the samples prior to the hydration was set to ramp up from 35°C to dehydration temperature of thermochemical storage materials at 5°C / min, and then to cool down at -5°C / min to 35°C. The hydration process was conducted at 35°C and switched on after a settling time of 5 min to start the hydration of the sample. The moist nitrogen gas flow was 125 mL/min, corresponding to a water vapour pressure from 20 hPa during hydration at 35°C. The amount of sample used was around 10 mg. Integration of the hydration heat fluxes over time was done manually using the Mettler Toledo STAR’ Software 9.30b.
RESULTS AND DISCUSSION

During the Hydration and Dehydration, the enthalpies of MgCl$_2$ * 6 H$_2$O, CaCl$_2$ * 6H$_2$O and Ca(OH)$_2$ were measured over several cycles. We report on cycle stability in depending of hydration completeness, water vapour pressure and temperature. Dehydration behavior of thermochemical storage materials was determined in temperature of dehydration steps.

CONCLUSION AND OUTLOOK

We will outline the cycle stability and heat storage density of thermochemical storage materials. Herefrom the ability for heat storage applications is given.

REFERENCES:

SYNTHESIS OF SAPO-34 ON GRAPHITE FOAMS FOR ADSORBER HEAT EXCHANGERS

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INTRODUCTION

The feasibility of direct synthesis of zeolites on carbon materials was already proved (Garcia-Martinez et al., 2001) and for SAPO-34 on carbon supports by the authors (Bonaccorsi et al., 2011). In this paper, such a possibility was further investigated by the preparation of an innovative adsorber bed obtained by growing the zeolite on cellular graphitic structures. Coatings of SAPO-34 were directly deposited by in-situ synthesis over on-purpose prepared graphite foams. It is known that the direct growth of zeolite layers on the heat exchanger surface allows an ideal contact at the zeolite/support interface that improves heat transfer between the adsorbent material and the adsorber heat exchanger (Miltkau and Dawoud, 2002). In-situ synthesis can be a realistic technology for adsorption heat pumps provided that proper supports with high surface area are employed (Freni et al., 2009). However, the mass ratio between heat exchanger and adsorbent phase must be as low as possible (Cacciola and Restuccia, 1999). Cellular aluminum has been suggested as ideal support for zeolite direct synthesis (Bonaccorsi et al., 2007) although graphite, which shows intrinsic low density and high thermal conductivity, could be even a better material for SAPO-34 deposition, once that an open-cell structure is given to the support. For this reason, different samples of graphite with cellular structure have been prepared and used for direct synthesis of SAPO-34 and their properties evaluated for a possible application in low-temperature driven adsorption heat pumps.

EXPERIMENTAL

Samples of graphite foam with different porosities were prepared by removable-template technique: a mixture of granular sodium chlorite and granular phenolic resin was pressed in a cylindrical mold and heated, then immersed in distilled water to remove the template and pyrolysed in order to graphitize the open-cell phenolic structure. Finally, the so-prepared graphite supports were pre-treated by oxidation in air and then used as support for the direct growth of SAPO-34 zeolite by hydrothermal synthesis. For deposition, it was adopted the same SAPO-34 formulation used in (Bonaccorsi et al., 2011) and the support was directly immersed in the synthesis mixture in an autoclave where the reaction was conducted for 72 hours. The coated supports were characterized by SEM, EDS, XRD analysis and measurement of water adsorption isobars by thermo-gravimetric methodology.

RESULTS

The oxidation pre-treatment significantly increased the support surface hydrophilicity and, consequently, wettability of graphite surfaces by the hydrothermal synthesis environment. The growth of coatings of pure zeolite SAPO-34 covering completely the foam surface was observed (figure 1). The final porosity of graphite foams influenced the quantity of zeolite deposited being directly correlated to the support surface area. Some samples of graphite foam reached a weight increase of 20% after one in-situ synthesis treatment. Measurements of water uptake in the typical range of pressures (P = 10 – 50 mbar) and temperatures (30 °C ≤ T ≤ 150 °C) of a “low temperature” adsorption cycle, confirmed the known behaviour of a regular SAPO-34, which means a large capability to adsorb/desorb water vapour in a narrow operating temperature range and the possibility of a complete adsorbent regeneration even at (relatively) low temperatures.
Figure 1: SEM of SAPO-34 coating on graphite foam and related X-Ray diffraction pattern.

REFERENCES


To study the adsorption kinetics of the granular activated carbon (GAC)/R134a pair, a test rig is designed and built as shown in figure 1. The experimental results were introduced in a comparison with the predicted results of three theoretical models (linear driving force LDF, Fickian FD and modified linear driving force MLDF). The results for the experimental runs and the models were introduced at four different temperatures (25°C, 35°C, 45°C and 65°C).

**Figure 1: Schematic diagram of the experimental test rig**

**THEORETICAL MODELS**

The Linear Driving Force (LDF) model for gas adsorption kinetics is frequently and successfully used for analysis of adsorption column dynamic data and for adsorptive process designs because it is simple, analytic, and physically consistent. The rate of adsorption in the LDF model for the spherical shape estimates using equation 1 [Ruthven, 1984].

\[
\frac{w - w_{in}}{W_0 - w_{in}} = \left(1 - \exp \left(-k_a \alpha \gamma \frac{w}{W_0} t\right)\right)
\]

Where, \(k_a\) is the overall mass transfer coefficient, \(w\) is the instantaneous uptake, \(w_{in}\) is the initial uptake and \(W_0\) is the maximum uptake in kg/kg adsorbent.

The exact value of adsorption uptake can be estimated using FD which is in equation 2 [Khairul Habib et al., 2011, 2011 and Crenk J. 1975].

\[
\frac{w - w_{in}}{W_0 - w_{in}} = 1 - \frac{6}{\pi^2} \sum_{n=1}^{\infty} \frac{1}{n^2} \exp \left(-n^2 \frac{\pi^2}{\rho} k_a \alpha \gamma t\right)
\]

Where \(n\) is an integer value and varies from 1 to infinity and \(t\) is the time in seconds.

El-Sharkawy, 2011, introduced a modification for the classical linear driving force. The MLDF appears in equation 3, below.
\[ \frac{w - w_{in}}{W_0 - w_{in}} = 1 - \exp \left( -15 \left( \frac{T}{T_c} \right)^n \left( \frac{D_s t}{R_p^2} \right)^m \right) \]  

[3]

Where \( n \) and \( m \) are constants, \( T \) is the adsorption temperature, \( T_c \) is the condenser temperature, \( D_s \) is the surface diffusion and \( R_p \) is the radius of the adsorption particle.

RESULTS

Figure 2 shows an example of the results of adsorption uptake at 25°C.

![Figure 2: Adsorption uptake at 25°C](image)

The results showed that all models achieved a good approach with the experimental results and the FD model captures the experimental results more accurately. The MLDF model drew better fitting with the experimental results than the classical LDF model. The MLDF model was did not agree well with the experimentally measured adsorption uptake data from the first 100 to 500 seconds. The MLDF requires more modifications.

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El-Sharkawy I I, On the linear driving force approximation for adsorption cooling applications, Int. J. Refrig. 34, 3, 2011, pp. 667-673


INTRODUCTION

The presented work aims at the experimental evaluation of the kinetic performance achievable by a binder based coating of a SAPO with chabazite framework type for the use in adsorption chillers or heat pumps. The samples have been prepared at the CNR-ITAE laboratories and the adsorption kinetics performance has been measured comparatively by means of two different setups: the first one available at the Fraunhofer ISE Institute and the second one at the CNR-ITAE labs. The samples generally show faster sorption rate compared to a loose grains configuration.

EXPERIMENTAL ACTIVITY

The SAPO-34 coatings have been prepared at CNR-ITAE by a dip-coating method. Two different compositions have been chosen and deposited on two identical aluminum substrates. Afterwards, measurements of adsorption kinetics have been carried out both using the Large Pressure Jump - LPJ method and the Large Temperature Jump - LTJ method. The LPJ measurements have been executed by means of a volumetric setup available at the Fraunhofer ISE Institute [Schnabel et al. 2010], while the LTJ measurements have been carried out by means of a gravimetric apparatus recently developed at the CNR-ITAE Institute [Sapienza et al. 2012]. Numerous tests have been performed changing several parameters in order to evaluate their influence on the global dynamic performance of the samples. In figure 1 the experimental evolutions obtained for a coated sample, during an LPJ measurement, have been represented. As a first step to judge the possible performance of such a coating in a defined heat exchanger configuration and application, performance figures such as the adsorption halftime or the rise up time [Dawoud 2010] have been calculated.

Afterwards, the same experimental activity was carried out using a configuration of loose adsorbent grains and the results were compared to the kinetic performance of the previously tested coated samples.

Finally, both for the coated samples and the loose grains configuration, the experimental results have been fitted by using two different models of the non-isothermal adsorption kinetics previously developed [Füldner et al. 2011, Freni et al. 2011], such obtaining heat and mass transfer parameters, which in a second step will be used to simulate the adsorption cycle with realistic boundary conditions to predict possible COP and volumetric specific powers or to e.g. optimize the layer thickness or the size of adsorbent grains.

CONCLUSIONS

The presented work has been focused on the analysis of the dynamic performance of two binder based coatings for adsorption chillers or heat pumps applications.

The kinetic tests were conducted using both the LPJ and the LTJ methods, by means of two different apparatus, and the evolutions obtained have been compared to the loose grain configuration.

Finally two models of the non-isothermal adsorption kinetics were used to fit the test results obtained both on the coated and loose grains configurations, so that the heat and mass transfer
parameters for the different configurations have been identified, and in a second phase they will be implemented to simulate the whole adsorption cycle using the realistic boundary conditions.

Figure 1: Relative water uptake and surface temperature of the sample after large pressure jump for different desorption condition and coupling to the coldplate with or without thermal grease (TG).

REFERENCES


SIMULATION OF A DYNAMIC ADSORPTION MODEL IN REAL WEATHER CONDITIONS

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ABSTRACT

Adsorption cycles use a low-grade heat source and an environmentally benign working pair to provide the compression traditionally produced by an electric compressor in a refrigeration cycle. The elimination of conventional refrigerants and the need to reduce carbon dioxide emissions have increased interest in sorption technologies, with much research being published on how to improve the low COPs and poor heat transfer characteristics of the adsorbent bed (Anyanwu 2003).

Generally, one of two approaches is taken to simulate an adsorption cycle: either a detailed analysis of the heat and mass transfer in the adsorbent bed is carried out to develop a more accurate generator model; or a lumped parameter generator model is used as a basis for modelling the whole system (refrigeration cycle, heat source, cooling load). Most research focuses on developing and optimising a single system for a specific climate.

TRNSYS (Transient Energy System Simulation) is a piece of software that is growing in popularity for modelling HVAC systems by setting the characteristics and connecting together a range of pre-existing components from the TRNSYS library. It also allows real weather data to be used to classify different climates and to input to the system (Aristov et al. 2007). These more detailed models can produce more reliable performance results than simplified models (Weber et al. 2009).

This paper describes a model which has been set up using TRNSYS and Matlab, enabling a range of working pairs to be tested under different conditions. The TRNSYS model includes a heat source (hot water tower), a cooling water source (cooling tower), a cooling load (simple building adapted for food or human cooling), real weather data, and a link to the adsorption system (see Figure 1). The generator is modelled in Matlab and assumes the kinetics of sorption follow a linear driving force model with Arrhenius type isotherms. A detailed temperature gradient between the heating/cooling water and the adsorbent is modelled for a flat plate heat exchanger and for a coated tube heat exchanger.

Figure 1: System layout in TRNSYS
Adsorption cooling systems have two main advantages: they can be powered using solar thermal energy and they are very low maintenance. This makes them suitable for use in remote locations where there are limited funds for continuing maintenance and fuel supply. Important applications of solar cooling include storage of medicines and preservation of food stuffs. Bearing this in mind, potential climates were assessed against a range of criteria reflecting the need for a suitability of solar cooling systems. These criteria were: climate – specifically the average daily temperature and solar radiation, reliance on agriculture, risk from infectious diseases, and poverty levels.

This analysis identified an ordered list of countries of which the top 5 were chosen: Niger, Burundi, Somalia, Zimbabwe and Nepal. They are spread over latitudes from -17.8 to 27.7, with average daily temperatures ranging from 18°C to 30°C. An example of the system performance for the activated carbon-methanol working pair using a flat plate heat exchanger to cool a food storage unit in Bujumbura, Burundi is shown in Figure 2. The temperature inside the building is maintained below 10°C while the ambient temperature is around 24°C.

![Figure 2: Building temperature vs. ambient temperature in Burundi during one year](image)

REFERENCES


INTRODUCTION

Two basic configurations of combined heat and power (CHP) systems using various low-temperature (up to 200 °C) heat sources have been built and analyzed (see for example Schuster et al, 2009). Practically all of them include a power generating unit operating according to the subcritical Rankine cycle. The first configuration comprises three heat exchangers. In the vapor generator the heat source transfers a fraction of its energy to the power generating unit. It is then directed to a second heat exchanger which reduces further its temperature and supplies a heating load (district heating is a common application). The third heat exchanger is the condenser of the power generating unit. The second configuration comprises only two heat exchangers since the heating load is supplied by the condenser of the power generating unit. A variant of this second configuration incorporating a heat pump was recently analyzed by Guo et al (2011).

The choice of working fluid for such power generating units is the subject of many studies. Water is not suitable at these low temperatures. Therefore organic fluids or mixtures such as H₂O/NH₃ have been used in prototypes and commercial systems (Di Pippo 2004, Hjartarson et al 2005). Theoretical studies have also considered trans-critical cycles with fluids such as CO₂ (Cayer et al, 2010). Most recent studies of such units fix the source and sink temperatures and compare the performance of different working fluids (Saleh et al, 2007) or seek to optimize a particular performance indicator (maximize the thermal efficiency, minimize the exergy losses, etc.) by varying the evaporation pressure of the working fluid (Lakew and Bolland, 2010). However these studies have not addressed the optimization problem when the power generating unit is part of a CHP system which must satisfy a prescribed heating load.

The present study analyzes and compares the performance of the two basic CHP configurations for fixed temperatures of the heat source, the heat sink and the fluid supplying the heating load. The power generating unit is an Organic Rankine Cycle (ORC) using R134a. The evaporation pressure which maximizes the net power output is determined for different fixed values of the heating load. Corresponding values of other significant system variables are also determined and compared.

MODELING OF THE CHP SYSTEM

The system is supposed to operate under steady state conditions. Heat and pressure losses as well as kinetic and potential energies are neglected. The working fluid is assumed to be saturated liquid at the exit from the condenser. The heat source is an industrial gas which is modeled as air. The isentropic efficiency of the pump and the expander are fixed (80%). We also fix the temperature of the heat source entering the vapor generator (Tₛ,in=100 °C), of the cooling water entering the condenser (Tₚ,in=10 °C) and of the water which supplies the heating load at the inlet of the heat exchanger (Tₕ,in=55 °C). The difference DT between the temperature of the working and external fluids at the inlet of the latter into the vapor generator and the condenser takes one of three chosen values (5, 10 or 15 °C). Two constraints are also imposed on the operating variables. Firstly, we require that at the expander exit the liquid content of the working fluid should be less than 5%. Secondly, we specify that the temperature pinch in each heat exchanger must be no less than DT/2.

The model comprises the equations expressing mass and energy conservation for each component of the system, the expressions of the expander and pump isentropic efficiencies in terms of the appropriate thermodynamic properties as well as the expressions of the heat and power transfers in terms of the appropriate mass flowrates and enthalpies. It also includes the expression of each heat transfer rate in terms of the thermal conductance and logarithmic mean temperature difference of each heat exchanger (vapor generator, load heat exchanger and/or condenser). These equations
have been implemented in EES (Klein 2009) which also includes relations between the thermodynamic properties of R134a and many other pure fluids. The resulting system of non-linear algebraic equations has been validated (Khennich and Galanis, 2011) in the case of a simple ORC generating unit (i.e. which is not part of a CHP system).

In the present case this system of equations involves more variables than equations. We have therefore addressed an optimization problem for different specified values of the heating load $Q_h$. Its objective is to determine the evaporation pressure of the R134a which maximizes the net power output of the ORC and satisfies the specified heating load. The investigated range of evaporation pressures is limited by the saturation pressures of R134a corresponding to the condensation temperature ($T_c = T_{p,in} + DT$) and its temperature at the exit from the vapor generator ($T = T_{s,in} - DT$).

RESULTS

Typical results for DT = 5 ºC and the two studied configurations are presented in Tables 1 and 2 respectively. The extensive quantities (heating load $Q_h$, maximum net power output $\dot{W}_{n,max}$, total exergy destruction $\dot{E}_d$ and total thermal conductance of the heat exchangers $UA$) are given per unit mass flowrate of the heat source. The intensive quantities are the optimum evaporation pressure (determined with the Golden Section Search method available in EES) and the corresponding evaporator pinch $\Delta T$. Additional results in graphical form will be included in the full paper.

### Table 1: Typical results for the first configuration (heating load supplied by heat source)

<table>
<thead>
<tr>
<th>$Q_h$ (kJ/kg)</th>
<th>4.59</th>
<th>9.18</th>
<th>13.8</th>
<th>18.4</th>
<th>23.0</th>
<th>27.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>$W_{n,max}$ (kJ/kg)</td>
<td>3.14</td>
<td>2.62</td>
<td>2.07</td>
<td>1.51</td>
<td>0.94</td>
<td>0.38</td>
</tr>
<tr>
<td>$P_{ev,opt}$ (kPa)</td>
<td>2880</td>
<td>3078</td>
<td>3266</td>
<td>3374</td>
<td>3374</td>
<td>3374</td>
</tr>
<tr>
<td>$\Delta T$ (ºC)</td>
<td>2.50</td>
<td>2.50</td>
<td>2.50</td>
<td>3.21</td>
<td>5.00</td>
<td>5.00</td>
</tr>
<tr>
<td>$\dot{E}_d$ (kJ/kg)</td>
<td>2.38</td>
<td>2.20</td>
<td>2.05</td>
<td>1.91</td>
<td>1.78</td>
<td>1.64</td>
</tr>
<tr>
<td>$UA$ (kJ/kgK)</td>
<td>9.40</td>
<td>8.27</td>
<td>6.97</td>
<td>5.39</td>
<td>3.77</td>
<td>2.32</td>
</tr>
</tbody>
</table>

### Table 2: Typical results for the second configuration (heating load supplied by condenser)

<table>
<thead>
<tr>
<th>$Q_h$ (kJ/kg)</th>
<th>4.59</th>
<th>9.18</th>
<th>13.8</th>
<th>18.4</th>
<th>23.0</th>
<th>27.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>$W_{n,max}$ (kJ/kg)</td>
<td>0.262</td>
<td>0.517</td>
<td>0.723</td>
<td>0.852</td>
<td>0.879</td>
<td>0.784</td>
</tr>
<tr>
<td>$P_{ev,opt}$ (kPa)</td>
<td>3455</td>
<td>3368</td>
<td>3106</td>
<td>2830</td>
<td>2551</td>
<td>2277</td>
</tr>
<tr>
<td>$\Delta T$ (ºC)</td>
<td>4.25</td>
<td>2.50</td>
<td>2.50</td>
<td>2.50</td>
<td>2.50</td>
<td>2.50</td>
</tr>
<tr>
<td>$\dot{E}_d$ (kJ/kg)</td>
<td>0.232</td>
<td>0.422</td>
<td>0.598</td>
<td>0.784</td>
<td>1.022</td>
<td>1.268</td>
</tr>
<tr>
<td>$UA$ (kJ/kgK)</td>
<td>1.93</td>
<td>4.14</td>
<td>5.96</td>
<td>7.52</td>
<td>8.89</td>
<td>10.1</td>
</tr>
</tbody>
</table>

These results show that for the first configuration $W_{n,max}$ decreases as $Q_h$ increases while for the second one it increases, reaches a maximum and then decreases. In the case of the first configuration $P_{ev,opt}$ and the corresponding $\Delta T$ increase with $Q_h$ while in the case of the second one they both decrease when $Q_h$ increases. $P_{ev,opt}$ for the first configuration is higher when $Q_h$ is small and lower when $Q_h$ is big. The exergy destruction is always lower for the second configuration while $UA$ is lower for the second configuration only for small values of $Q_h$.

REFERENCES

EXPERIMENTAL INVESTIGATION OF SILICA GEL GRANULAR PACKED RECTANGULAR FINNED TUBE ADSORBENT-BED

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Recently interest in adsorption cooling systems has increased due to their capability to utilise low grade heat sources and environmentally friendly refrigerants. They are applied in combined cooling, heating and power systems (CCHP) employed in many industrial and commercial applications. Currently, most of the commercially available adsorption cooling systems utilise granular packed adsorbent beds, where their advantages of high permeability level and ease of manufacture. Several adsorption pairs were evaluated and silica gel/water has been shown to have significant advantages in terms of thermal performance and environmental impact. Although water refrigerant is considered in chilling applications only, it has excellent thermo-physical properties of high latent heat of evaporation, high thermal conductivity, low viscosity, thermally stable in wide range of operating temperature and the compatibility with wide range of materials. Silica gel as water vapour adsorbent has the advantages of high adsorption/desorption rate, low generation heat and low generation temperature.

This paper investigates the effects of operating conditions on the performance of scaled down silica gel / water adsorbent bed. The effects of the adsorption time and cooling water flow rate on the bed water vapour uptake were studied at fixed generation temperatures. Figure 1 is a schematic diagram and pictorial view of the test rig and the adsorbent bed used. Figure 2 presents the effect of adsorption time on the amount of adsorbed water vapour at cooling water flow rate of 30 LPM, heating water flow rate during regeneration period 23 LPM, regeneration temperature of 65˚C and cooling water inlet temperature of 25˚C. Modelling of the adsorption bed based on the empirical lumped analytical simulation technique was carried out and results were comparable to the experimental results. This experimental and modelling approach can be used as effective tool to investigate and optimise such bed design.
Figure 1: Test rig (a) test rig schematic diagram (b) scaled down adsorbent bed (c) pictorial view of the test rig

Figure 2: The adsorbed water vapour versus adsorption time
THE DESIGN AND MANUFACTURE OF A 50 KW JET-PUMP CHILLER FOR AIR CONDITIONING APPLICATIONS

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INTRODUCTION

A prototype ejector chiller with a nominal cooling capacity of 50 kW has been built at Frigel s.p.a. in Florence. The plant is intended as a test bench for future ejector chillers to be used in the industrial temperature control market.

The working fluid is R245fa, which is thought to be a likely candidate for future ejector refrigeration, due to its high critical temperature, moderate pressure at generator, reverse slope vapour curve and reasonable GWP. At present R245fa has a rather high cost, but further diffusion as a working fluid for ORCs (Organic Rankine Cycles) should increase availability.

PROTOTYPE FEATURES

Plate heat exchangers produced by GEA GmbH have been used for the generator, evaporator and condenser. For experiments, the generator will be heated by water at 115°C produced in an electric heater. The evaporator is designed to produce cold water at 7°C. The condenser is designed to work on cooling water up to 35°C.

A vertical layout with condenser on top, evaporator in the middle, generator and pump at the bottom has been adopted, in order to mitigate cavitation problems at pump inlet. The pump is a multistage centrifugal unit with magnetic drive and inverter controlled variable speed produced by Speck Pumpen. The primary nozzle position can be moved by a screw actuated slid.

The mixing chamber and diffuser have been manufactured in a single piece of carbon fibre composite. Seven static pressure ports have been set on the piece. Other pressure probes are set on the generator, evaporator and condenser. Thermocouples are installed at all relevant cycle points. Primary and secondary flow rate are measured. A data acquisition system computes in real time the entrainment ratio and the COP.

DESIGN PROCEDURE

A detailed description of the design procedure, based on a monodimensional calculation, has been presented by Grazzini et al., 2012. Real gas properties have been used throughout the modelling, incorporating the NIST REFPROP subroutines within the simulation code.

The ejector mixing chamber and diffuser (figure 1) are designed according to Constant Rate of Momentum Change (CRMC) principle introduced by Eames, 2002.

The CRMC part has been kept short, in order to limit the length of the high speed zone and hence reduce friction losses. Downstream of the throat, the diffuser has a conical profile with 5° half angle. However, the first results of a CFD analysis have shown that the length available for complete mixing may be insufficient. Therefore, a second diffuser, with a longer throat, has been manufactured. The two diffusers are interchangeable. A detailed CFD analysis of these two diffusers is in process.

According to one-dimensional simulation, the prototype should have an entrainment ratio 0.55 and a COP of 0.39. Cooling capacity should be slightly less than 47 kW.

STATE OF THE PROJECT

The prototype is now complete (Figure 2) and has been pressure-tested. The first running tests are planned for the second half of February.
Figure 1: Mixing chamber and diffuser

Figure 2: View of the prototype and calculated thermodynamic cycle

REFERENCES


DEVELOPMENT OF A SOLAR DRIVEN DOUBLE/SINGLE EFFECT LiBr-H2O COOLING UNIT USING INNOVATIVE MEDIUM TEMPERATURE SOLAR COLLECTOR

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The aim of this study is to develop a solar driven double/single effect LiBr-H2O cooling unit using an innovative medium temperature solar collector. The studied LiBr-H2O absorption cooling unit can be altered between the double effect and single effect modes automatically, depending on the driving temperature provided by the solar collectors. In this way, the solar thermal energy can be used more efficiently for cooling, i.e. the average COP of the unit is higher than that of the single effect absorption unit and the time for producing cooling is longer than that of the double effect unit. Moreover, by using the specially developed medium temperature evacuated tube solar collector, the solar collecting section can provide thermal energy at higher temperature level compared to the normal solar collectors. Results show that this kind of solar collector can work steadily at temperature over 130°C, and the thermal conversion efficiency is about 46%. It is expected that the daily average COP of the studied solar absorption cooling system can be about 0.8, and can produce cooling for 6-8 hours relying purely on solar energy during a typical sunny day. In this work, the performance of such unit will be investigated extensively.

SYSTEM DESCRIPTION

Figure 1: Solar driven absorption cooling system

In summer, the solar energy is gained through the collector array and is accumulated in the storage tank. Then, the hot water in the storage tank is supplied to the absorption chiller, thereby providing cooling by fan coil for the building. An auxiliary energy source is provided, so that the hot water is supplied to the generator when solar energy is not sufficient to heat the water to the required temperature level needed by the generator of the absorption chiller. Figure 1 shows both the P-T diagram and the schematic of the solar cooling system. It is seen that depending on the temperature level of the solar collector, the solar heat can be input to low pressure generator or high pressure generator. In this case, the solar heat can be used more efficiently and the chiller can work for a longer time.

Figure 2 shows the photo of the M-T solar collector and the collector array, as well as the efficiency curve. It can be found that the solar collector with improved coating and the CPC reflector, the thermal efficiency is improved at high operation temperature level. The efficiency can reach 50% when the temperature is 130°C. Thus the collector is feasible to drive the double effect solar cooling cycle.
It is believed that M-T solar collector will have a perspective to adopt in solar power air condition system. Two generators working at different temperatures are operated in series, whereby the condenser heat of the refrigerant desorbed from the first generator is used to heat the second generator. Thereby a higher COP in the range of 1.1–1.2 is achieved. When the hot water temperature value from collectors can achieve more than 140°C, the chiller runs in double effect mode. And when it is between 130°C and 88°C, the chiller will automatically be switched to single effect mode. So it is expected that the COP of the system could reach to 1.2 at the highest value in double effect mode and the daily average COP of the system can be about 0.8. It can produce cooling for 6-8 hours relying purely on solar energy during a typical sunny day.

ACKNOWLEDGMENT

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A HOT AIR DRIVEN THERMOACOUSTIC-STIRLING ENGINE

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INTRODUCTION

Significant energy savings can be obtained by implementing a thermally driven heat pump into industrial or domestic applications. Such a thermally driven heat pump uses heat from a high-temperature source to drive the system which upgrades an abundantly available heat source (industrial waste heat, air, water, geothermal). A way to do this is by coupling a thermoacoustic engine with a thermoacoustic heat pump. The engine is driven by a burner and produces acoustic power and heat at the required temperature. The acoustic power is used to pump heat in the heat pump to the required temperature. This system is attractive since it uses a noble gas as working medium and has no moving mechanical parts or sliding seals. This paper deals with the first part of this system: the engine. In this study, hot air is used to simulate the flue gases originating from a gas burner. This is in contrast with a lot of other studies of thermoacoustic engines that use an electrical heater as heat source. Using hot air resembles to a larger extent the real world applications.

DESCRIPTION OF THE ENGINE

A schematic illustration of the thermoacoustic Stirling-engine is shown in Figure 1. The engine consists mainly of a torus-shaped section attached to a quarter-wavelength acoustic resonator. The torus-shaped section contains the regenerator (REG) sandwiched between a ambient heat exchanger (AHX) to remove the rejected heat from the engine and a hot heat exchanger (HHX) to supply heat to the engine. The torus contains a feedback tube (L) to create the travelling-wave phasing necessary to operate in a Stirling cycle (Ceperly,1979; Backhauss,1999; Tijani, 2011). The gas column in the thermal buffer tube (TBT) provides thermal insulation for HHX. An acoustic load consisting of consists of a 1-liter tank and an adjustable valve is used to control the power output of the engine.

![Figure 1: Schematic illustration of the thermoacoustic Stirling-engine.](image-url)

The thermoacoustic computer code DeltaEC (Ward, 1994) is used to design the engine. The engine uses helium at 40 bar and an operation frequency of 130 Hz. The regenerator consists of a 6 cm long stack of 180-mesh stainless steel screens with a diameter of 6.68 cm. The diameter of the screen wire is 40 μm. A volume porosity of 78% and a hydraulic radius of 35 μm. The AHX is of cross-flow type and consists of a cylindrical brass block with a length of 2 cm and containing copper fins (50 fins/in) at the helium-side and at the other side cooling water flows through groves. The HHX consist of fin-fin cross-flow heat exchanger with 50 fin/inch at the helium side and 10 fins/inch at the hot air side. The TBT consists of a cylindrical tube with an inside diameter of 7 cm, a wall thickness of 1.6 mm, and a length of 17 cm. The feedback tube (L) consists of a cylindrical tube with a length of 73 cm and diameter of 6.69 cm. The total length of the resonator is about 3.5 m consisting of a straight and a conical part. The regenerator holder, HHX, and the TBT are made from the same high temperature steel material. Groves are machined inside the steel block where fins are placed and brazed at both air and helium side.
RESULTS

Figure 2 shows the measured performance of the thermoacoustic engine as a function of the temperature of the hot air for two drive ratio’s. The drive ratio is the ratio of the acoustic pressure and the average pressure. The performance is defined as the ratio of the acoustic power output of the engine and the heat input. The performance increases as function of the hot air temperature. The engine produces about 300 W of acoustic power with a performance of 41 % of the Carnot performance at a drive ratio of 6 % and a hot air temperature of 620 °C. This relatively good performance is obtained in spite of a considerable heat leakage down the TBT and the not optimal operation of HHX. An improvement of the performance might be expected by suppressing this heat leak and improving the HHX.

CONCLUSION

A thermoacoustic-Stirling engine is designed, built, and tested. The engine produces about 300 W of acoustic power with a performance of 41 % of the Carnot performance at a drive ratio of 6 % and a hot air temperature of 620 °C. The performance of the engine can be improved by suppressing the heat leak down the thermal buffer tube and improving the HHX. A better solution might be to use a hot head instead of the HHX. The hot head with fins can be directly heated by the hot gases as usually done in Stirling-engines.

REFERENCES

INTRODUCTION

A gas-fired heat pump system intended to replace conventional condensing boilers is under development. The machine uses four sorption generators with heat recovery between all beds plus mass recovery and has a nominal heat output of 7kW. The sorption pair is active carbon – ammonia. Predicted annual average heating COP (Heat output / gas energy input based on gross calorific value) exceeds 1.35 in a UK application with low temperature radiators.

In the drive to develop products and systems that will result in major energy end use reductions the development of domestic sorption heat pumps has a long history and many variations have been tried. One of the earliest attempts was by Shelton (1986) and the Wave-Air company. More recently products are being developed by Vaillant, Viessmann and Robur, Albus, 2009. The work described here has been carried out by the University of Warwick as part of a project called CALEBRE (Consumer Appealing Low Energy Technology for Building Retrofit) funded by EPSRC and EoN and is the subject of further development by Sorption Energy Ltd. The concept is an affordable air source sorption heat pump that can replace a conventional gas-fired boiler and reduce gas consumption by over one third.

MODELLING AND DESIGN

A simulation model of a four bed system has been validated by Metcalf (2009), and used to generate the predicted performance data of Figure 1. The machine was designed around a specification for a typical UK house that had been retrofitted with good insulation to the point that the heat pump needed to deliver 7kW of heat. Assuming the UK climate and heat delivery at 50°C to low temperature radiators the calculated mean annual COP based on the higher calorific value of gas burnt would be 1.35.

Previous sorption generators have used plate shim designs but theoretical modelling suggests that a shell and micro-tube construction could have less thermal mass and equivalent heat transfer. The core of such a heat exchanger is shown in Figure 2 and fully assembled in Figure 3.

Figure 4 shows the completed prototype in its initial form but without insulation. It is designed as a single unit to be installed outside the house with water, gas and electrical connections being made through the wall.

Figure 1: Predicted COP
PROGRESS TO FEBRUARY 2011

The prototype is mechanically complete inside an environmental test chamber that will be used to evaluate the performance under a range of ambient conditions to EU standards. The load is simply a set of radiators in an adjoining environmental chamber. Initial testing was successful in that the unit pumped heat and delivered water at high temperature (up to 60°C). However, the efficiency was poor because of excessive heat leakage from pipework and internal leakage of water in the valve assemblies. Bespoke valves using ceramic discs to direct water flows have been designed and should be fitted during February 2012, after which testing will recommence.

ACKNOWLEDGEMENTS

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FIRST CYCLE SIMULATIONS OF THE HONIGMANN-PROCESS WITH LIBR/H\textsubscript{2}O AND NAOH/H\textsubscript{2}O AS WORKING FLUID PAIRS AS A THERMOCHEMICAL ENERGY STORAGE

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A thermochemical energy storage invented in the 19\textsuperscript{th} century, the so-called ‘Honigmann-process’, is being reconsidered for the storage of renewable energies and waste heat (Jahnke et al., 2009). In general, the storage can be charged with the input of heat or mechanical work and discharged with the release of heat, cold or mechanical work. This flexibility is the major advantage of the process compared to other storage technologies. Another advantage is the absence of self discharge.

In this work, one basic process option of charging with the input of low grade heat and discharging with the release of mechanical work is being studied. A fundamental model based on transient energy and mass balances has been implemented with the help of the modeling language Modelica. With this model, it is now possible to simulate the dynamic behavior of the process to investigate operating and control strategies and the choice of different working fluid pairs. At the conference the key features of the model and first simulation results are being presented.

In this section, the working principle will be described. The discharging process is based on two basic physical effects (Figure 1 and 3):

1. The first effect is the vapor pressure depression of an aqueous solution compared to the working fluid (water). Due to the difference in vapor pressure an expansion machine can be operated by steam flowing from a water vessel to a solution vessel at an even higher temperature.
2. The second essential effect is the occurrence of heat of absorption which can be recovered. The water vapor is absorbed in the solution and heat of absorption is set free. This heat is transferred to the water vessel to evaporate more water and the expansion continues.

At first glance this resembles a perpetual motion machine, which, of course, it is not. The solution becomes more and more diluted until the vapor pressure difference becomes too small to run the expansion machine. This is the time when the storage needs to be recharged. It can be done, for example, with the input of heat to evaporate (desorb) the water out of the aqueous solution (Figure 2). To recover the water and maintain a closed cycle process the water vapor will be condensed at a lower temperature.

Figure 1: Discharging

![Figure 1: Discharging](image1)

Figure 2: Charging

![Figure 2: Charging](image2)

The four main components have been modeled and the physical models have been translated into the object-oriented modeling language Modelica: one vessel with a heat exchanger containing aqueous solution, one vessel with a heat exchanger containing water, one valve, and one expansion machine. The vessels were assumed as completely stirred tank reactors with only one temperature, mass fraction, and the corresponding vapor pressure. Effects of superheating or subcooling were not considered. The full model and all assumptions are described in detail in Strenge (2011).

First simulations with two different working pairs have been carried out: aqueous lithium bromide solution and water (LiBr/H\textsubscript{2}O) and sodium hydroxide solution and water (NaOH/H\textsubscript{2}O). Results are
shown in the Raoult plot with isosteres for both working pairs (Figure 3). Thermodynamic property data are from Pátek et al. (2006) (LiBr/H$_2$O) and Olsson et al. (1997) (NaOH/H$_2$O), and the Modelica media library (IAWPS 97) was used for water. The initial temperature and pressure as well as all boundary conditions are identical for both simulations. In both cases 3.6 kg water was expanded. The resulting differences originate from the different media and initial concentrations. Reasons and consequences will be discussed in the full paper.

![Figure 3: Raoult plot with vapor pressure lines and process cycles with LiBr/H$_2$O and NaOH/H$_2$O](image)

This work is being carried out within a research project funded by the DFG (Deutsche Forschungsgemeinschaft). To validate the simulations an experimental test facility is being constructed and experiments will be done.

Thanks to Alexandra Mehlhase for help with Modelica specific problems and the usage of variable-structure models.

REFERENCES

Simulation of adsorption cycles is carried out in different levels of detail. For detailed optimization of components such as the adsorber heat exchanger (AdHex), usually detailed heat and mass transfer models are preferred, while for the understanding of the system level including its dynamics, usually so called lumped parameter models or linear driving force (LDF) models are in use.

In this contribution, simulations in a detailed PDE model with parameters such as diffusion coefficients, heat conductivities or transfer resistances have been performed with COMSOL Multiphysics. Then, a lumped parameter model has been parameterized by these detailed model results.

INTRODUCTION

Adsorption heat pumps or chillers can be used to make efficient use of available thermal energy such as solar or waste heat for chillers or natural gas for gas fired heat pumps [Meunier 1998]. Mathematical models in various levels of complexity are used to understand, design and optimize the system and its components [Yong 2002] such as the AdHex, evaporator/condenser or the heat rejection unit.

CYCLE SIMULATION WITH DETAILED HEAT AND MASS TRANSFER

In previous publications it has been shown that detailed heat and mass transfer models describing the Adhex geometry and solving the adequate coupled PDEs such as diffusion and heat conduction can be used to optimize e.g. the AdHex geometry with respect to power density at high COP values [Füldner 2011]. Values for heat and mass transfer are taken both from direct measurement (densities, porosities, pore size, heat conductivity) and from comparison of adsorption kinetic measurements with a volumetric uptake method after a large pressure jump [Schnabel 2010] to the PDE model of non-isothermal adsorption kinetics [Schnabel 2009]. In this publication, such cycle simulations with realistic boundary conditions (temperature switches between ad- and desorption) and evaporator/condenser pressure modeled with a single-node model making use of effective parameters fitted to experiments on evaporation and condensation have been performed. COP and volume specific cooling/heating power have been calculated. Due to the level of complexity, such simulations only allow to look at a few cycles with fixed boundary conditions.
Figure 1: Cyclic water uptake and mean temperature in composite AdHex consisting of aluminum fibres coated with a zeolite layer in detailed heat and mass transfer simulation.

CYCLE SIMULATION IN LDF-MODEL

To describe a sorption apparatus in a system context, e.g. coupled to a storage and serving a load, having non-stationary boundary conditions such as varying fluid inlet temperatures, simulations can be performed with lumped parameter models [Schwamberger 2011, Schicktanz 2009]. The parameters of these models usually have to be obtained by fitting to experimental data from measurements of a specific components in a given system. Extrapolation to different configuration is usually difficult. In this contribution, the lumped parameter model is calibrated using the results from detailed modeling, such obtaining empirical relations for the parameters dependencies on e.g. operating conditions but also geometrical configurations of the AdHex.

CONCLUSION

Detailed ad-/desorption cycle simulations with a heat and mass transfer model of the AdHex coupled to single-node models of evaporator and adsorber are used to calibrate and validate a simplified lumped parameter model of an adsorption chiller or heat pump.

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THERE’S (STILL) PLENTY OF ROOM AT THE BOTTOM

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APOLOGIES TO R. P. FEYNMAN

In 1959, the Nobel Prize winning physicist, Richard P. Feynman, presented a lecture at a meeting of the American Physical Society (held at the California Institute of Technology, Feynman’s home institution) with (nearly) the same title as we have used for this abstract. Feynman suggested the possibility of “construction” on the atomic scale: “In the year 2000, when they look back at this age, they will wonder why it was not until the year 1960 that anybody began seriously to move in this direction.” With the rise of nanotechnology, that talk is now referenced frequently.

The purpose of this presentation is to suggest a similar opportunity in small-scale energy production. Interest, research, and engineering effort in the fields related to energy conversion, and particularly the conversion of heat to electricity, has focused on large-scale power plants since Edison switched on the Holborn Viaduct plant in London and built the Pearl Street Station generating plant in Manhattan, both in 1882. To this day, 130 years after Edison’s successes, most research and development is still directed toward systems that produce large amounts of power - from 50 kW (e.g., automobile engines) to coal and nuclear plants with power generation capabilities in excess of 1 GW.

Two aspects of technology have changed over the past two decades that motivate a re-examination of the power generation infrastructure, especially in developing countries. First, the recent rise of useful low-power electronics and high-efficiency lighting (e.g., fluorescent tubes and LEDs) means that significant benefits can be realized with substantially less power. Secondly, the same time period has seen a substantially increased global concern about the waste products that accumulate in the atmosphere causing potentially cataclysmic global consequences (Campbell, 2009). These shifts lead to the idea that there now may be some important applications of very small-scale (<100 W) distributed electrical power generation to serve the 1.5 billion inhabitants of this planet currently without access to an electrical distribution grid (World Energy Outlook, 2009). Of course, that quarter of our world’s population which is without electricity is concentrated in developing countries. For example, in Africa, the countries of Burundi, Chad, Central African Republic, Liberia, Rwanda, and Sierra Leone have between 95% and 97% of their population without access to electrical service.

The Energy Impoverished, Biomass-Burning Cook Stoves, Health, and Climate Change

Even a small amount of electricity can be transformative. It can provide power for high-efficiency electrical lighting (e.g., LED or fluorescent) that let children study at night or provide parents with the opportunity to produce items at home that can be sold to generate additional income. Small amounts of electricity can provide connection to family, markets, and opportunity via radio, cellular telephones, and other small electrical appliances.

The same 1.5 billion people who lack electrical services are also using biomass burned in “three-stone fires” to cook their meals. These open fires are responsible for 75% of the black carbon which is the cause of the Asian Brown Cloud (Szikat, 2009) and constitute a larger contribution to global warming than the CO₂ produced in those developing countries (UNEP, 2011).

Reduction of black carbon (BC) and other products of incomplete combustion (PIC) is an attractive approach to mitigation of the effects of global warming in the short-term because BC has an atmospheric lifetime that is shorter than two weeks, whereas the atmospheric lifetime of CO₂ is centuries and has a long-time tail of approximately 30,000 years (Archer, 2004). It has been estimated that removal of BC could delay the predicted consequences of global warming by a decade or two thereby providing time to craft politically palatable solutions to limit CO₂ emission. Additionally, improved cook stoves are an attractive policy alternative because they also improve the health of women and children and reduce deforestation (Wallack and Ramanathan, 2009).
A biomass-burning cook stove that can convert a small amount of the thermal energy produced by the stove (typically 4 to 8 kW\textsubscript{Thermal}) into electricity makes the improved stove a stronger value proposition in many areas. Finally, a small amount of that electricity can be used by the stove itself to power a fan ($\lesssim 2$ W\textsubscript{el}). It has been shown many times that forced convection is the best way to reduce the products of incomplete combustion emitted by traditional stoves (Jetter and Kariher, 2009).

**Small-scale Cook Stove Co-generation Benchmarks and Technological Trade-Offs**

This talk will focus on the difficult questions that surround the options for using electrical power co-generation technologies at small-scales and will review progress to date using thermoelectric, steam, and thermoacoustic methods to produce electricity as a by-product of cooking (Garrett, et al., 2009). The thermoelectric approach provides a useful benchmark, since it has been investigated by sophisticated manufacturers (e.g., Philips, BioLite) and is currently being field-tested and nearing commercialization. Issues related to the advantages of individual stove co-generators ($< 10$ W\textsubscript{el}) and pot surrogates (i.e., co-generative kettles placed on a stove when it is used for space-heating while not cooking), institutional stove co-generators ($\lesssim 100$ W\textsubscript{el}), and larger dedicated biomass-burning village battery “charging stations” will be presented, but not resolved. Similarly, potential unintended consequences, such as the increased use of stoves solely to generate electricity at extremely low efficiency (i.e., not exploiting “waste heat” but burning biomass at a rate of $5$ kW\textsubscript{Thermal} to generate a mere $5$ W\textsubscript{el}), will be exposed, though not quantified.

This presentation will focus on the technical and engineering trade-offs, but it is equally important to realize that potential solutions for alleviation of rural energy poverty require input from stakeholders as well as sociological and marketing research that explores use patterns, fuel type and availability, and user preference. Also important is the development of quantitative, significant, and reproducible performance metrics and test protocols that lead to the development of cook stove pollution and efficiency standards. Creation of successful business models that result in the marketing, distribution, and sales of 250,000,000 improved stoves each year to customers that live far from conventional transportation arteries, on less than $2.50US/day, may provide an even greater challenge (Cordes, 2011).

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INTRODUCTION

Spoelstra et al., 2002 have shown that for the Dutch chemical and refinery industry over 100 PJ of waste heat per year is actively released in the temperature range between 50 and 150 °C. Ideally this heat is reused in the same process, thereby reducing release of waste heat in the environment and reducing primary energy use. For small temperature lifts (around 30 °C) compression heat pumps are a suitable option but they limited to a maximum output temperature of approximately 120°C. For larger temperature lifts (>50°C), a heat-driven heat pump type II would be suitable. Such a heat pump - based on ammonia-salt reactions - has been developed at ECN. The disadvantage of this concept is that the required waste heat temperature should be at least 100°C but preferably higher than 120°C. This reduces the amount of available waste heat considerably (see Figure 1). By introducing a compressor in this heat pump cycle, the required waste heat temperature can be reduced whilst still realizing net energy-savings. This paper will show this concept is feasible, both technical and economical.

Figure 1: Amount of waste heat actively disposed in the Dutch refinery and chemical industry as a function of waste heat temperature.

Figure 2: Thermodynamic cycle of the hybrid heat pump type II configuration.
HYBRID ADSORPTION COMPRESSION CONFIGURATION

Figure 2 shows the thermodynamic cycle in a PT-diagram with a compressor placed in the high pressure/temperature discharge phase. The discharge phase is preferred over the charge phase because the work provided by the compressor leads to additional useful heat production and the volume flow is relative small so a smaller and thus cheaper compressor can be used.

Using a database containing thermodynamic properties of various ammonia-salts, various salt combinations were found that would be suitable for use in the above shown cycle (van der Pal et al., 2011). Most promising is the CaCl₂(4-8)NH₃ reaction in combination with MnCl₂(6-2)NH₃. Part of the analysis is determining the power density and the coefficient of performance (COP) for heat and work (electricity). The COP_{heat} is mainly determined by the required temperature lift and the (assumed) heat losses to the environment and only partly affected by the chosen salts. The COP_{electric} is determined by the electricity used by the compressor, which is determined by the required pressure ratio. The pressure ratio is strongly affected by the chosen salts and therefore an important parameter for the salt selection. For the above mentioned salt-combination, a COP_{electric} (ratio heat out: electricity input) of more than 5 has been calculated for temperature lifts of >50°C for waste heat temperatures exceeding 70°C.

A total of four configurations have been assessed in the economical evaluation, resulting from two varied parameters. These parameters are:
1) the presence of a secondary circuit versus a direct steam feed ;
2) reactor consisting of a standard priced shell-and-tube heat exchanger based on the numbers form the DACE-handbook, 2009 versus a reactor consisting of a shell and tubing.

The final cost calculation includes the costs for integration, maintenance and operation. The benefits have been calculated using a range of operating times, energy prices and COP_{electric}. The results show that for current energy prices, the cheapest configuration has a payback time of less than 2 years whereas the most expensive configuration takes approximately 9 years to pay back. The payback times of two other configurations are 5 and 6 years. These periods will be shorter for higher energy prices.

DISCUSSION AND CONCLUSIONS

The hybrid adsorption compression heat pump concept is considered technical feasible in terms that sorbents have been found that meet the criteria regarding pressure-temperature behavior and regarding power density. The payback time of the hybrid concept has been calculated to be between 2 and 9 years at current energy prices. This is considerably better than payback time for PV-panels and comparable with wind turbines. With an outlook of increasing energy prices, the payback times will only be reduced. Therefore, it is concluded that the hybrid adsorption compression heat pump concept is feasible. Based on the available waste heat in the range from 70 to 150°C, about 80 PJ of waste heat could be used, resulting in a reduction of primary energy use of approximately 30 PJ per year for the Dutch refinery and chemical industry. Reusing waste heat from other streams such as hot flue gases could increase this amount considerably. Worldwide the energy-savings potential is even larger.

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ORGANIC RANKINE CYCLES IN WASTE HEAT RECOVERY: A COMPARATIVE STUDY

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INTRODUCTION

Waste heat is an inherent and abundant by-product of many industrial processes. Often, this waste heat is too low a temperature for any useful energy to be extracted from it by conventional cycles. Organic Rankine Cycles (ORCs) offer potential solutions to recover some waste heat and convert it into mechanical power. The overall efficiency of a wide range of thermal cycles can be improved by the addition of an ORC to reduce the amount of heat rejected from the system.

Many studies consider potential ORC applications individually (e.g. Roy et al., 2010 investigate power station flue gas). Here, three potential ORC applications are considered with a range of energy scales. This study investigates to what extent it is possible to design ORCs with common features with different energy scales. Matlab has been used to build an ORC model capable of simulating different organic fluids over a wide range of operating conditions. The properties of the organic fluids in the model were taken from FluidProp.

THE THREE POTENTIAL WHR APPLICATIONS EXAMINED IN THE STUDY ARE:

IC Engines

Several ORC WHR systems exploiting the waste heat available from internal combustion engines are currently under development e.g. the Cummins Super Truck Program (Koeberlein 2011). This program aims to improve the specific fuel consumption of class 8 trucks by 10% Heat rejected in the exhaust of a large diesel engine accounts for approximately 31% of the raw fuel energy (Butts 2006) and is rejected to atmosphere at a temperature of approximately 120°C (Tchanche et al, 2011). ORC systems operating under these conditions could make a significant contribution to the overall performance improvement goals of this and other similar programmes.

Hot Brines

Aging oil wells require water injection to maintain the pressure and sweep oil from pore spaces. North Sea petroleum reservoirs are between 2.5km - 5km deep so the oil and water mix produced from the well has a temperature in the range 70°C - 150°C. This is within the operating range of ORCs. The quantity of hot water produced can be very large (~1000kg/s), resulting in high potential for large power production from low grade heat energy recovery systems (Younger et al, 2011). Platforms generate power for water injection pumps by burning gas produced by the oil; however in wells with falling production the gas produced becomes insufficient and must be imported at vast expense. This creates a requirement for alternative methods to generate power. Typically the pumps used are rated at 4MW. The potential for ORCs to meet this demand is investigated in the study.

Industrial Plant

Many industrial processes produce waste heat, for example cement, textile and electricity production. The example used in this study is a typical plant supplying steam and electricity to surrounding process works. In this plant, steam from a gas and a biomass fueled boiler is used to produce thermal and electrical energy. WHR from exhaust steam and flue gases is explored for a number of available sources around the plant. The steam supplies considered are between 125°C - 326°C, with mass flow rates of 0.4 kg/s - 32 kg/s. Similarly, the flue gas sources are 130°C - 137°C at a flow rate of 145kg/s.
DESCRIPTION OF THE MATHEMATICAL MODEL

A Rankine cycle has been modeled using Matlab and FluidProp. The fluids modeled are dry, as recommended by Tchanche et al., 2011. The fluids selected are R245fa, n−Pentane, n−Octane, Toluene; they have a range of critical temperatures covering the various thermal conditions modeled. Pinch point heat exchange, condensing temperature and efficiencies of each component were set as constant values. Each heat source and fluid was simulated; in each case the Turbine Inlet Pressure (TIP) was optimised for the maximum work output.

Figure 2 shows a plot of Power output against TIP. The graph shows two cases; a source of steam in the industrial plant and a typical hot brine oil well. The graph shows potential mechanical power generation from both sources is in the order MW. Thermal efficiency is labeled (eff) on Figure 2; thermal efficiency increases with TIP. Power output calculated for the hot brines is larger than for the industrial steam despite having a lower temperature, this is because the mass flow rate of hot brine produced is much larger.

![Figure 1: Example plot from two simulations, using n-Pentane as the working fluid.](image)

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MEASUREMENTS OF TEMPERATURE FLUCTUATIONS IN THERMOACOUSTIC SYSTEMS

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INTRODUCTION

In order to maximise the performances of thermoacoustic systems, it is crucial to better understand how heat is transferred between the stack (the main component where the thermoacoustic effect takes place), and the heat exchangers. Heat transfer between the stack and the heat exchangers depends on the aerodynamic field. Our group was the first to demonstrate, both numerically and experimentally, the generation of vortices near the edges of the stack that can influence the performances of a thermoacoustic device (Blanc-Benon et al. (2003), Berson et Blanc-Benon (2007), Berson et al. (2008)) For instance, the instability of vortex shedding between the stack and the heat exchangers can be responsible for low-frequency oscillations of the cooling load.

The technique for measuring velocity fields in thermoacoustic systems is now well defined. However, in order to fully characterize heat transport, the instantaneous temperature field has to be characterized too. No existing technique has a sufficient resolution to measure temperature fluctuations at the low amplitude and high frequency found in thermoacoustic systems. In this paper, we will summarize our recent work on the development of a new methodology for measuring temperature fluctuations using a “cold wire” operated by a constant-voltage anemometer (CVA) (Berson et al. (2009, 2010, 2010b)). The new methodology is validated inside an acoustic standing-wave resonator. Measurements near the end of a thermoacoustic stack show that the temperature field is nonlinear, in good agreement with theoretical models.

A new method for measuring temperature fluctuations in oscillating flows

To our knowledge, “cold wires” are the only way to measure temperature fluctuations accurately over a frequency range (up to 1000Hz in our case) and for small amplitudes (a few kelvins at most) relevant to thermoacoustic systems. “Cold wires” are fine metallic wires whose resistance varies with the temperature of the surrounding fluid. Their frequency response is limited by the thermal inertia of the sensor and, in order to measure high-frequency temperature fluctuations, wires with very fine diameters are necessary, which makes them difficult to make and very fragile.

We have developed a new procedure for correcting the thermal inertia of “cold wires”. The procedure consists in operating the wire first in the heated mode in order to measure the “time constant” of the wire. In oscillating flows, the “time constant” is actually not constant with time as it fluctuates with velocity. We take advantage of the unique features of the CVA to measure this fluctuating “time constant” directly, without having to measure the velocity. Once the “time constant” is known, the wire is operated in the cold mode to record temperature fluctuations. Subsequently, during post-processing, the temperature measurements obtained from the “cold wire” are corrected for the thermal inertia of the wire by combining them with the measurements of the “time constant”. With this method, the frequency range of “cold wires” is significantly extended.

Measurements in an acoustic resonator and near the ends of a thermoacoustic stack

The technique is first validated in an acoustic standing-wave resonator. The resonance frequency is 464 Hz, and the acoustic level inside the resonator ranges between 1000 Pa and 3000 Pa. Measurements are performed with a 2.5 μm diameter cold-wire probe. Figure 1 shows the spatial distribution of acoustic pressure amplitudes for three different acoustic levels (1000 Pa, 2000 Pa and 3000 Pa). Temperature fluctuations are largely underestimated when the effect of thermal inertia is not corrected. With our correction, a good quantitative agreement is found with predicted
temperature fluctuations. We are able to measure accurately temperature fluctuations less than 0.2K at 928 Hz thanks to our new procedure.

Figure 1 Example simulation, using R245fa and TIP = 10.24 bar,

The new method is used to measure temperature fluctuations behind the edges of the stack. The temperature field behind the stack is nonlinear. Good agreement is found with a simple one-dimensional model describing the temperature field near the stack (Berson et al. (2011))

REFERENCES

STUDY ON HYBRID REFRIGERATION AND HEAT PUMP SYSTEMS

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ABSTRACT

There is a project undergoing at Warwick that is aimed to investigate various conceptual designs of novel hybrid refrigeration and heat pump systems, including building and testing a proof of concept prototype (up to 5 kW cooling capacity). The hybrid system uses an environmentally friendly refrigerant as R723 (azeotropic mixture: 40% DME and 60% Ammonia) that is compatible with conventional refrigeration copper alloy (CuNi10): it is a combined sorption-conventional vapour compression refrigeration machine driven by dual source (heat and/or electricity). The potential use of R723 in both adsorption process with out its decomposition (Tamainot-Telto, 2009) and conventional vapour compression machines (Final Report 2005, Palm 2007) is already established. The use of dual source makes the proposed systems when operating highly flexible and energy efficient. The use of R723 in an adsorption cycle and the dual energy source are key novelties of this project.

The main objective of this paper is to describe four configurations of hybrid system including associated challenges and potential applications.

Figure 1 is an illustration example of hybrid system layout. This system consists of three heat exchangers (cooler, condenser, and evaporator), two compressors (conventional and thermal), two water circulation pumps, a float valve/receiver and a 3-way valve. For this configuration only one compressor operates at one time on common refrigerant loop. It could either be the Heat Driven Compressor (HDC) or the Electrically Driven Compressor (EDC) that is operating. A 3-way valve diverts the refrigerant to the operating compressor. This could be used for waste/solar heat driven refrigeration machine backed up by electrical power as the waste/solar heat sources are usually intermittent. This flexibility offers the advantage of cost effective running cost of machine. Although R723 refrigerant has good miscibility with mineral and synthetic oils, the effect of oil on the sorption reactor performance will need to be investigated before considering a hybrid system with common refrigerant loop. If the presence of oil has a negative effect on sorption generator performance, the use of oil free type of Electrically Driven Compressor (EDC) such as reciprocating compressor from Aximaref (Quiri) will be a viable solution available for a practical operating machine.
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THEORETICAL INVESTIGATION OF AMMONIA–WATER KALINA CYCLE SYSTEM 11

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ABSTRACT

With the ever-increasing energy demand, exploiting low-temperature heat sources has seen significant interest recently. The conventional Organic Rankine Cycle (ORC) is a typical approach used to exploit low temperature heat sources, but suffers from low efficiency (Papadopoulos et al, 2010). The Kalina Cycle (Kalina, 1984) is a reversed absorption cooling system that uses ammonia/water binary mixture as the working fluid. The use of binary mixture that exhibit variable boiling temperature during the boiling process offer the advantage of maintaining the temperature difference between the heat source and the working fluid small thus allowing for a good thermal match between them with potential of higher efficiency compared to the conventional ORC at the same heat source and heat sink temperatures (Ibrahim, 1996).

This paper reviews the various Kalina cycle configurations used to harness low grade waste heat and particularly investigates the performance of Kalina Cycle System11 (KCS11) that utilises low temperature heat sources below 200°C compared to that of ORC using pure ammonia or pure R134a as the working fluid. Figure 1 shows a schematic of the KCS11 comprising of evaporator, separator, turbine, absorber, condenser, regenerator, throttling valves and a circulating pump. Thermal modelling of the KCS11 cycle was carried out using conservation of energy and mass laws applied to each component and the resulting set of equations were solved using the Engineering Equation Solver (EES) software.

The cycle performance was investigated at various operating conditions where the evaporator pressure ranges from 10-32 bars, heat source temperature (T_{source}) ranges from 333K-463K, and ammonia mass fraction of up to 90%. Figure 2 shows the performance of the KCS11 cycle at evaporator pressure of 15 bars and heat sink temperature of 283K at various evaporator temperatures. Results show that the cycle thermal efficiency increases steeply with the increase in the ammonia mass fraction to reach a point of maximum and then decreases gradually to reach a minimum value at relatively high ammonia mass fraction. The significant drop in efficiency at low ammonia mass fraction could be due to the phase of the ammonia/water mixture approaching saturated liquid state with limited or no vapour to be produced in the separator.

Figure 3 compares the performance of the Kalina cycle KCS11 with ammonia mass fraction of 0.55 and 0.66 to that of ORC using ammonia or R134a as the working fluid at various evaporator pressures while the heat source and heat sink temperatures fixed at 373K and 283K respectively. It can be seen that the thermal efficiency of the Kalina cycle with ammonia mass fraction of 0.55 is higher than those of the ORC using ammonia and R134a for evaporator pressure values below 20bars. For example, at evaporator pressure of 15 bars, the KCS11 thermal efficiency (11.38%) with ammonia/water concentration of 0.55 is approximately 40% higher than that of the ORC using pure ammonia (7%) and 20% higher than those of the ORC using pure R134a (9.2%). Therefore, it is clear from these figures that Kalina cycle can produce significant advantages in the efficiency compared to the conventional ORC cycle in terms of higher efficiency at lower operating pressure values.

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Figure 1: Schematic diagram of Kalina Cycle

Figure 2: Kalina (KSC11) cycle thermal efficiency at maximum pressure of 15 bars and sink temperature of 283K.

Figure 3: Comparison between Kalina cycle and ORC based on pure ammonia and pure R134a at evaporator temperature of 373 K and heat sink temperature of 283 K.
ABSTRACT

Waste or prime heat can be converted into electricity with thermoacoustic-Stirling engines coupled to piezoelectric alternators. An inline arrangement of engines and alternators allows a vibration balanced, multiphase power generator that is compact, lightweight and low cost. The engines convert heat into high amplitude ~400 Hz oscillations in pressurized helium gas. These pressure oscillations cause the thin steel diaphragm of the alternator to flex like a drumhead. The diaphragm is supported at its perimeter by a ring of piezoelectric elements. As the diaphragm flexes in either direction, it pulls inward on the piezoelectric elements causing a large, amplified ~800 Hz fluctuating compressive stress in the elements. The piezoelectric elements then convert the fluctuating stress into electricity with high efficiency. The flexible-diaphragm piezoelectric alternator overcomes the large acoustic impedance mismatch between the helium and piezoelectric elements without exceeding the limited fatigue strength of available materials. Acoustically, the chain of engines act as springs that generate power, and the alternators act as masses that extract power. Vibration balance is achieved with an integer number of oscillation wavelengths along the chain of springs and masses.

So far, a prototype generator has produced 37 W, and is being modified to produce 600 W. An application is the recovery of 7 kW peak electrical power (4 kW average over typical driving cycles) from the exhaust of an over-the-road heavy-duty diesel truck. The generator appears scalable up to megawatt power levels.

(Work supported by the US Department of Energy, US Office of Naval Research, Clean Power Resources, Innovation Works, and the Penn State Applied Research Laboratory.)
New and more efficient refrigeration methods are needed to reduce building energy consumption and environmental impact. Buildings currently account for 72% of the electricity use of the United States and 40% of carbon dioxide emissions each year, 1.5% of which comes directly from commercial refrigeration. In addition, the refrigerants used in refrigeration technologies are potent greenhouse gases (GHGs) that may contribute to global climate change. Because the majority of refrigeration systems run on electricity, and most U.S. electricity comes from sources that produce CO₂, there is a pressing need to support improvements that increase the efficiency of these technologies and reduce the use of GHG refrigerants.

This paper presents Trillium, a thermoacoustic food freezer with 3.5 kW of cooling capacity that uses helium as a working fluid. This inline design is based on the ideas of Backhaus and Keolian ("In-line Stirling Energy System," US Patent No. 7,908,856 B2, 2011). If there is a differentiation between a thermoacoustic machine and an alpha-Stirling machine, this design represents the closest approach between the two.

Shown in Figure 1, Trillium is comprised of three linear motors driven 120° out of phase, an arrangement which provides a vibration-free operation. In between each motor is the thermal core where the acoustic pressure and volume velocity are in-phase to pump heat from the load heat exchanger to the exhaust heat exchanger.

This talk will present the details of the design strategy including load matching the drivers, determination of the thermal core area, and dynamic variables such as frequency, pressure and volume velocity. Performance predictions will also be presented.
INTRODUCTION

Fundamental progress in the material science can give new opportunities for adsorption heat transformation applications. In fact, their performance and efficiency is foremost governed by the adsorbent properties and characteristics (Aristov 2009). However, a common and comparable thermophysical characterization of adsorbent materials for heat transformation applications, remains a critical point. To this aim, in a previous paper (Henninger et al. 2011), an unified procedure for water adsorption measurement on sorption materials was presented. Measurements on two different adsorption materials, namely silica gel 127 B and SAPO-34, showed a good agreement.

Aim of this paper is to investigate if the procedure for adsorption equilibria measurements of meso-microporous solid adsorbent materials, already defined presented and considered as a standard, is suitable for characterization of other class of materials (MOFs) that recently have been proposed for their use in adsorption heat pumps. MOFs are a new emerging class with unique features and properties. Compared to common adsorbents used in heat transformation like zeolites and zeo-like materials (AlPOs, SAPOs) or other porous adsorbents and composites (MCM, SBA, SWS,..) MOFs show a completely different building concept, therefore facing new problems on determination of the equilibrium characteristics like slow adsorption rate, extremely high uptake, large hysteresis and unsure stability.

The previous measurement procedure is critically reviewed and results obtained with different apparatus are compared.

WATER ADSORPTION CHARACTERISTICS

Water adsorption measurements on MOFs are still scarce, however recently several publications reporting tremendous water adsorption capacity are emerging (Küsgens et al. 2009; Akiyama et al. 2010; Henninger et al. 2009; Ehrenmann et al. 2011).

According to the previously proposed procedure, a commercially available MOF, namely Basolite™ C-300 supplied by Sigma-Aldrich and produced by BASF has been evaluated. This MOF is one of the first 3-dimensional porous MOFs, also called HKUST-1 or just Cu-BTC. The results of the measurements are shown in Figure 1. As can be seen, this material shows the highest water load within the reference materials with an uptake of nearly 40 wt. %. The shape of the isobar shows a steep increase between the temperature range of 80°C – 50°C, which is advantageous for the application within a sorption heat pump or chiller. However, there is a difference between the two measurements which may be attributed to the poor hydrothermal stability of this particular material. Independently performed cycle tests showed a degradation of 40% within 10 cycles. Therefore, the first desorption step may cause irreversible change of the material. In addition the measurements were performed in two different systems, first a Cahn-Balance operated under vacuum and pure water atmosphere (ITAE) versus a Setaram open-flow system under atmospheric pressure and carrier gas (ISE). Taking all these points into account, these results are showing a surprisingly good agreement.
Figure 1: 12 mbar water adsorption isobars measured for different samples at Fraunhofer ISE and CNR ITAE.

CONCLUSION

Although a hydrothermal unstable MOF is no good choice for a reference material in order to compare measurement results or to align different measurements, the measurement procedure itself shows a quite good reliability towards these effects ending with a good agreement between the adsorption data measured by the different apparatuses available at Fraunhofer ISE and CNR ITAE.

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MIXTURES AS WORKING FLUIDS IN ORGANIC RANKINE CYCLES

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ABSTRACT

We present a systematic and comparative study of the potential of zeotropic mixtures as working fluids in ORCs, considering most of the commonly used hydrocarbon and siloxane substances as components in various concentrations. We investigate the impact on the operation, the efficiency and power output of an ORC. The ORC cycle is realistically simulated in steady state conditions, taking into account all elements of an actual cycle (including an internal heat exchanger). By performing a pinch analysis, the power output is maximized, given the heat profiles of both the heat source and heat sink. The use of suitable zeotropic mixtures as working fluids has a positive effect on the ORC performance in all investigated cases. The potential increase in cycle efficiency and generated electricity is larger for lower temperature heat sources and when the temperature drop over the heat source exchanger is larger. When the ORC is optimized for operation with a low temperature heat source, the potential for increase in electricity production (maximum reported value approximately 23\%) by using mixtures is particularly remarkable.

BACKGROUND

The main advantage of mixtures as ORC working fluids stems from their non-isothermal phase transitions during vaporization and condensation. This effect becomes apparent when comparing the temperature-entropy diagrams of the cycle processes, for example for pure pentane and for the mixture pentane/hexane (0.5/0.5), see Figure 1.

Figure 1: Temperature-entropy diagrams of the ORC process for (a) the mixture pentane/hexane (0.5/0.5) and for (b) pure pentane as the working fluid.

It is clear that the temperature profile of the mixture [Figure 1(a)] follows the heat source profile better than that of the pure fluid [Figure 1(b)]. This leads to less exergy losses and, as apparent in Figure 1, to higher turbine inlet temperatures, due to the smaller temperature difference across the heat exchanger. This in turn results in a higher total efficiency. Similarly, at the condensation side, the non-isothermal behavior of the mixture results in heat rejection at a lower (average) temperature, again increasing the total efficiency.
METHOD

We developed a tool to simulate the described cycle. The FluidProp library, in combination with the REFPROP database, provided the thermophysical data of the considered mixtures and pure fluids. Central in our calculations is a pinch analysis of the heat flows. A nonlinear optimization algorithm enables the determination of the maximal power output and corresponding operating pressures with respect to the preset pinch temperatures. The same algorithm can be used to find the mixture composition that maximizes the cycle efficiency. We define the cycle efficiency as the ratio of the net output power (electricity produced by the generator at the turbine minus the consumed electricity by the pump) over the thermal input power (source heat transferred to the ORC medium in the evaporator). Furthermore, the tool offers the possibility to analyze the temperature profiles in the evaporator, condenser and recuperator.

RESULTS

Although the efficiency of the ORC decreases when the heat source is cooled down more, the electricity production increases up till an optimum temperature gradient $\Delta T_{opt}$ and then decreases again. For a heat source at 150 °C, the maximum net generated electricity versus the corresponding temperature gradient $\Delta T_{opt}$ for all considered pure fluids and mixtures is plotted in Figure 2. The mixture compositions are also optimal. The radius of a disk symbolizing a data point is proportional to the relative cycle efficiency. This figure clearly visualizes the increased electricity production potential of mixtures compared to pure fluids. The mixtures operate optimally at higher $\Delta T_{opt}$ compared to pure fluids, signifying an increase in recuperated power from the heat source. In addition, as already mentioned, the evaporation (condensation) process occurs at higher (lower) average temperature, resulting in a higher cycle efficiency. For the considered heat source and ORC, under optimal operation, an increase in electricity production of over 23% by using mixtures instead of pure fluids as medium is attainable. Again, 3-component mixtures only yield a minor increase in generated electricity compared to 2-component mixtures. In our survey, using R245fa/isopentane/isoheptane (0.64/0.17/0.19) leads to the highest net electricity production (291.8 kW).

CONCLUSIONS

The use of suitable zeotropic mixtures as working fluids has a positive effect on the ORC performance in all investigated cases. The potential increase in cycle efficiency and generated electricity is larger for lower temperature heat sources and when the temperature drop over the heat source exchanger is larger. When the ORC is optimized for operation with a low temperature heat source, the potential for increase in electricity production (maximum reported value approximately 23%) by using mixtures is particularly remarkable.

Figure 2: Maximum net generated electricity versus corresponding optimal heat source temperature gradient for the fluids and mixtures given in the table, with optimal mixture compositions.
An adsorption chiller typically comprises a “reactor concept”, or “AdHex”, and a set of engineering structures (cooling system, containment); the two are connected thermally. Our work concerns a particular isolated “AdHex” for which an activated carbon cloth (ACC) adsorbent is interleaved between aluminium fins (the heat exchange surfaces). Our ultimate twofold aim will be to (1) measure all net heat and mass transfer to/from an AdHex, in isolation from complicating engineering components, throughout a simulated adsorption cycle (2) screen AdHex concepts using a calorimetric method. This paper concerns a first step in this program, measurement of heat rejection following a large pressure jump (LJP).

The calorimeter was a variant of Ahamat and Tierney’s (2012). The AdHex comprised six aluminium fins and a base, manufactured from a single block of aluminium (40 mm x 40 mm, with a mass of 60 grams) and 9 grams of ACC (Chemviron FM50K). A thermoelectric module (TEM) was thermally bonded directly to the AdHex base, and the base temperature held constant by controlling the electrical current to the TEM. Measurement of the TEM voltage and current supply yielded heat rejection via a previously discussed calculation method (Ahamat & Tierney, 2012). The AdHex was exposed to ethanol vapour and a large pressure jump (LPJ), typically from 0 to 20 mbar vapour pressure.

Plots of heat rejection (Figure 2) showed an exponential recovery. This simple form was notwithstanding the complex (and coupled) heat and mass transfer. In particular there were large temperature gradients within the ACC layers; temperatures measured at the centre of the ACC layers were as much as 30 K hotter than the fin root. This simple functional form of heat rejection has been previously observed for the silica-gel water working pair and is undoubtedly empirical. Measured heat rejection would allow estimates, by heat balance of average cooling power over an adsorption period of ~1000 s, in which time the ACC had cooled to its initial temperature (and was the same temperature as fin root). For time specific cooling power we expect temperature gradients to add ~6% to levels of uncertainty to such estimates. It is fair to point out that real-world chiller operation corresponds more closely to a large temperature jump (LTJ) than to a large pressure jump (LPJ) and for this reason a further set of experiments is needed.
Figure 2: Plots of cumulative heat rejection versus time, showing fits to exponential recoveries - the evaporator was held at ~0°C and the fin root held constant at the indicated temperatures

The adsorptive capacity was deduced from the total rejected heat (asymptotes on Figure 2). This serves to check the method accuracy. At present validation information is mildly encouraging. Changing the vapour pressure from 5% to 20% of the saturation value had minimal measured effect on adsorptive capacity; it remained within the range of 39% to 42%. Changing the adsorbent temperature had a distinct effect, however. At present, some mildly encouraging indications are (1) using a different (BASF) activated carbon, Gales too found almost constant adsorption capacity in the same range of relative pressure (but the adsorption capacity was lower, at about 30%) (2) we have found a similar asymptote (~40%) gravimetrically for methanol on the same Chemviron ACC. However, a gravimetric assessment of adsorption capacity of EtOH on Chemviron carbon is needed for full validation.

In conclusion, an exponential recovery fits the heat rejection from a particular AdHex following a large pressure jump; notwithstanding several coupled physical processes. We have inferred adsorption capacities; as in previous work with silica-gel water these checks against gravimetric measurements are needed. Further tests will be applied to large temperature jumps, more indicative of chiller boundary conditions. Temperature gradients are expected to add ~6% uncertainty when time specific cooling power is inferred.

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One of the most effective ways to improve the efficiency of a turbine-based power plant is to increase the Turbine Inlet Temperature (TIT). TIT is limited by material thermal and mechanical resistance. The route to the high temperature is ever more complex and effective blade cooling by forced internal convection. A recent publication (Gallo et al., 2012) describes experimental results demonstrating that the most relevant heat transfer augmentation effect is due to impinging flows. The goal of this work is to evaluate a new high-potential concept for blade cooling: liquid metal flows through channel walls and magnetic fields promote impinging, analogously to the ribs (turbulence promoters). The dimensionless numbers governing a liquid metal flow through a magnetic field are the Reynolds ($Re$) and the Stuart ($St$) numbers. The latter provides the relative weight of the Lorentz forces with respect to the inertial forces. DNS has been used to study the effects of non-uniform magnetic fields on the flow field and on the heat transfer of a liquid metal (lithium) flowing into a two-dimensional channel. The numerical experiments have been carried out in laminar regime ($Re=2000$) and Stuart number equal to 0 (non-magnetic case), 750, 1000 and 2000 using a grid $16x16x32$ in a computational domain $10Dx10Dx1D$. The two-dimensional non-uniform magnetic field applied to the liquid metal flow is periodic and constant in time and it is generated by two ideal electric currents positioned underneath the two channel walls in the span-wise direction (Figure 1).

From the performed simulations it came out that for values of the Stuart number up to 1000 the flow field is two-dimensional whereas becomes three-dimensional for $St=2000$. Figure 2 shows an exemplary simulation, which puts into evidence the influence of the intensity of the magnetic field on the flow field. Indeed the magnetic field promotes secondary flow fields having a periodic behaviour as highlighted by normal to wall velocity distributions $w$ (Figure 2a, b). The intensity of such secondary flow fields increases as the $St$ increases. Figure 3 and 4 report respectively the fluid temperature distribution for $Re=2000$ and $St=1000$ and the trends of the Nusselt number ($Nu$) in streamwise direction calculated at the upper surface ($z=1$) for all the simulated values of the $St$. 

![Figure 1: Magnetic field distribution with the field lines superimposed](image-url)
Figure 2: Normal to wall velocity component distributions ($w$) with velocity vectors superimposed ($Re=2000$; $y/D = 5$): a) $St=750$; b) $St=1000$

The periodic behaviour of the flow field causes both the thinning and/or breaking up of the boundary layer and the convective transport of “cold fluid pockets” from the core of the flow to the channels walls (Figure 3). The combination of these two effects produces a periodic increase of the convective heat transfer (Figure 4).

Figure 3: Fluid temperature distributions ($Re = 2000$, $St=1000$, $y/D = 5$)

The $Nu$ profiles (Figure 4) exhibit an oscillating trend having amplitude decreasing in streamwise direction and the maximum local increase of the Nusselt number has been calculated at $x/D=2$ for $St=2000$ and is about 2.8 times higher than $Nu$ measured in the non-magnetic case ($St=0$). The outcomes of this work are very promising and show the great potential of the “magnetic ribs” (non-uniform magnetic fields) as tool for improving the convective heat transfer in the turbine blade cooling systems.

Figure 4: Nusselt number profiles ($Re = 2000$, $z/D=1$)

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Early developments of adsorption heat pumps started 35 years ago, after the 1973 energy crisis, yielded interesting results on solar refrigerators and solar ice makers as well as on advanced cycles and consolidated adsorbents. The early commercialization of low-grade heat adsorption chillers stayed confidential for a long time.

More recent R&D activities confirmed the early results on solar refrigerators and solar ice makers without technical or commercialization breakthrough.

On the contrary, breakthroughs were achieved on new families of compound adsorbents which open new stimulating perspectives. Lots of progresses have also been achieved on coated surfaces, consolidated materials and better knowledge on the dynamics of adsorbent beds. Surprisingly, few studies on advanced cycles were performed so that no real advances were observed in that field.

Commercial products of adsorption chillers and heat pumps are now well known and much better products are proposed in terms of performances, robustness and reliability. Therefore, the question is now: what future for adsorption heat pumps in the present context of energy transition and global warming?

Heat recovery and low driving-temperature (including solar) chillers present great perspectives, even with moderate COP, if the recent and ongoing breakthroughs are introduced in the commercialized products so as to improve their performances. The applications could be enlarged, towards geothermal energy, if advanced cycles yielding a higher heat source temperature spread was achieved so as to increase the cooling rate for a given flow rate from the heat source.

Gas-fired heat pumps will have a chance to play an important role only if high performances are achieved so as to save CO$_2$ emissions as compared to other systems. For reversible gas-fired heat pumps, this will be possible only if R&D on advanced cycles is re-activated.
ABSORPTION REFRIGERATION AND JET PUMPS

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ABSTRACT

This paper provides an overview of heat powered refrigeration cycles and argues that whenever practicable, low-grade heat should be used to drive them. By way of example descriptions absorption and jet pump cycle refrigerators and their some innovative combinations of these cycles are described.

Unlike vapour compression refrigeration machines, which are driven entirely by mechanical work, both absorption and jet pump refrigeration cycles are driven by a combination of work and heat. The necessary work input to these machines, however, is used to compress the working fluid in its liquid state and because of this it normally a small fraction of the total. For this reason absorption and jet pump refrigerators are therefore described as either ‘heat-powered’ or ‘thermally activated’ cycle machines.

Fundamentally all heat powered refrigerators are based on the 3-temperature cycle, shown schematically in Figure 1. In the system shown, heat (Qg) at a high temperature, (Tg), is absorbed by a fully reversible heat engine which produces an amount of work (W) whilst rejecting heat to a sink at an intermediate temperature, (Tc). This work is then absorbed by a fully reversible refrigerator, which absorbs heat (Qe) at a low temperature (Te) and rejects heat to a sink at an intermediate temperature (Tc).

The overall coefficient of performance of this fully reversible machine is equal to the product of the Carnot cycle thermal efficiency of the heat engine and the coefficient of Performance of the Carnot cycle refrigerator (CoPc):

$$\text{COP}_{\text{overall}} = \eta_c \text{CoP}_c = \frac{Q_e}{Q_g} = \frac{(T_g - T_c)}{(T_c - T_e)} \frac{T_e}{T_g} \quad [1]$$

Figure 1: A 3-Temperature Heat Powered Cycle Refrigerator

Figure 2: Variation in $\text{COP}_{\text{overall}}$ with temperature

Figure 2 shows how $\text{COP}_{\text{overall}}$ varies with the heat source temperature, Tg. In this case Tc and Te are assumed to be 5°C and 40°C respectively.
When comparing the energy efficiencies of heat power cycles with those of work powered cycles, such as a conventional vapour compression machine, it is useful to compare their respective Carnot efficiencies. In this case Carnot efficiency is defined as the ratio between actual CoP_{overall} of a particular machine with the maximum theoretical value given by Equation 1. For the single-effect absorption refrigeration cycle operating with a vapour generator temperature of 80°C and an actual CoP of 0.7, its Carnot efficiency, according to the results given in Figure 2, will be in the order of 78% of the theoretical maximum. The double-effect cycle (also described later) produces an almost equal result. The thermal efficiency of the best gas engine is probably not more than 65% whilst for air conditioning refrigerators the best CoP is probably not greater than 7. Therefore, the CoP_{overall} of a gas-engine driven vapour compression refrigerator will probably not greater than about 4.6, (0.65 x 7), and according to Equation 1, this is about 68% of the theoretical maximum. In this case an average gas combustion temperature (Tg) of 1800°C was assumed, and as for all other cases Tc and Te were assumed to be 40°C and 5°C respectively.

This result indicates that whenever possible it is more energy efficient to utilise low-grade waste heat or that derived from other sources, such as solar, to create a required cooling effect, than to burn a fossil fuel to do so. However, there will almost always be a capital cost penalty because of smaller approach temperature differences found in low temperature heat powered refrigerators invariably lead to larger heat exchangers. Also, condensers must be larger to cope with the additional heat rejection.

The paper begins with a brief review of the vapour absorption cycle and the operation of the jet pump before going on to describe some of the research carried out by the author and various co-workers into combined absorption-jet-pump refrigerator cycle machines. Figure 3 and 4 show schematics of two examples that will be covered in the presentation.

**Figure 3:** Combined absorption – jet pump refrigerator cycle machine (circa 1989)

**Figure 4:** Combined absorption jet-pump cycle refrigerator cycle machine (circa 1998)