Technical and Economical Feasibility of the Hybrid Adsorption Compression Heat Pump Concept for Industrial Applications

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Abstract

Heat pump technologies offer a significant potential for primary energy savings in industrial processes. Thermally driven heat pumps can use waste heat as driving energy source to provide either heating or cooling. A chemi-sorption heat transformer can upgrade a waste heat source to temperatures of 150-200°C. The specific heat transformer process however requires waste heat temperatures in the range of 120°C, whereas waste heat sources of lower temperatures are more abundant. Using this lower temperature waste heat, and still reach the desired higher output temperatures can be achieved by the integration of a chemi-sorption and mechanical compression step in a single hybrid heat pump concept. This concept can offer an increased flexibility in temperatures, both for the waste heat source as for the heat delivery.
The technical and economical feasibility of the proposed hybrid heat pump concept is evaluated. The range of operating temperatures of different chemi-sorption working pairs for as heat driven and as hybrid systems are defined, as well as their energy efficiencies. Investment costs for the hybrid systems are derived and payback times are calculated. The range of payback times is from 2-9 years and are strongly influenced by the number of operating hours, the electrical COP of the compression stage, and the energy prices.

Introduction

Spoelstra et al. (Spoelstra, S. 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 °C and 160 °C (See Figure 1). Ideally this heat is reused in the same process, thereby reducing release of waste heat in the environment and reducing the primary energy use. The main reason this heat is released is that the temperature level of the heat is too low or the heat is not required at that place and/or at that time. In this paper we will focus on options for upgrading the heat to a higher temperature.

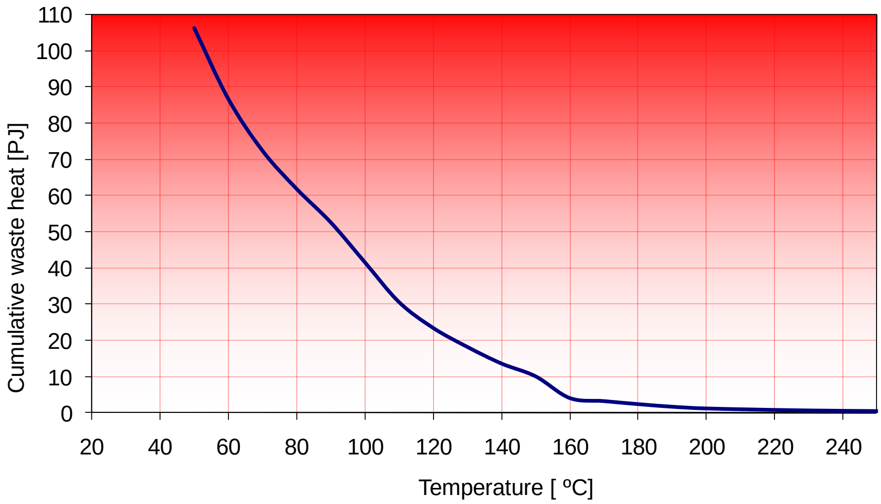

Figure 1: Amount of waste heat actively disposed in the Dutch refinery and chemical industry as a function of waste heat temperature.

Upgrading heat to higher temperatures can be achieved by using a heat pump. The common option is a (work-driven) compression heat pump. This, however, is only a suitable option for small temperature lifts (around 30 °C) and for a maximum output temperature of approximately 120 °C. Higher temperature lifts will reduce the efficiency of the compressor considerably and thereby the primary energy savings whereas the number of (accepted/approved) refrigerants for higher output temperatures is very limited (Narodoslawsky, M. et al, 1988)

A heat-driven heat pump type II (also known as heat transformer) could be used for upgrading heat with 50 °C or more. Such a heat pump is being developed at ECN (van der Pal, M. et al, 2009) and consists of a low temperature sorbent (LTS) and a high temperature sorbent (HTS). The working principle is as follows: by applying waste heat to the high temperature sorbent ($MgCl_2$) whilst keeping the low temperature sorbent (LiCl) at ambient temperature, the sorbate (NH_3) is released from the HTS ($MgCl_2(6\text{-}2)NH_3$) and adsorbed by the LTS ($LiCl(1\text{-}3)NH_3$). This is called the regeneration phase. Subsequently the LTS is heated using the low temperature waste heat. This results in a rise in (equilibrium) pressure. The HTS can now adsorb the sorbate and produce heat at a higher temperature, resulting in a net production of heat at a temperature higher than the waste heat temperature. This is called the discharge phase. The thermodynamic cycle is schematically shown in Figure 2. For these sorbents it has been shown that this cycle can yield temperature lifts of 70 ^{o}C for 130 ^{o}C waste heat in a lab-scale prototype heat pump (Haije, W. G. et al, 2007). The disadvantage of this concept is that the cycle requires relative high waste heat temperature for achieving both regeneration and temperature lifts of 50 ^{o}C. The waste heat temperature has to be at least 100 °C but preferably more than 120 °C. This greatly affects the amount of waste heat available for upgrading: at temperatures above 120 ^{o}C only 20 PJ/year is available. This is just 20% of the total amount of available waste heat.
A considerable larger amount of waste heat is available at lower temperatures. For example, if a heat pump cycle would work at temperature of 70 ^{o}C, 70 PJ/year of waste heat would be available, almost four times the amount at 120 ^{o}C. This cannot be done – assuming a temperature lift of 50 ^{o}C is required to outperform a standard compression system - using a single-stage heat-driven heat pump. Although multi-stage is in theory possible, the efficiency of such a system is poor, resulting in long payback times and low energy savings potentials.

To circumvent these problems, ECN is working on a new, hybrid technology using the heat-driven heat pump type II in combination with a standard mechanical compressor. By introducing a compressor in the heat pump type II cycle during the discharge phase, the pressure of the sorbate is increased before it is adsorbed by the HTS. This higher pressure will result in a higher equilibrium temperature and therefore it allows for higher output temperatures (see Figure 3). Alternatively, the compressor can be used in the regeneration phase of the cycle. Assuming the same LTS is used, it will cause the regeneration pressure of the HTS to drop (in this example the pressure on the LTS remains the same), and therefore reduce the required waste heat temperature for regeneration. The overall effect of the compressor is that hybrid heat-driven compression cycles can operate at lower waste heat temperatures (van der Pal, M. et al, 2011).

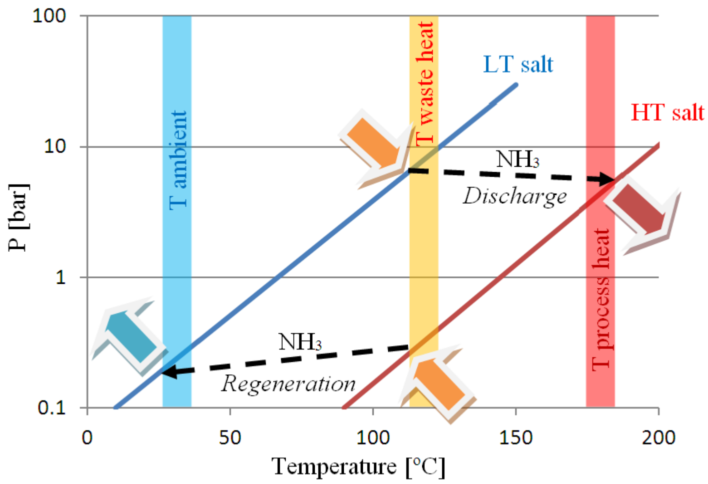

Figure 2: Thermodynamic cycle of the heat-driven heat pump type II configuration.

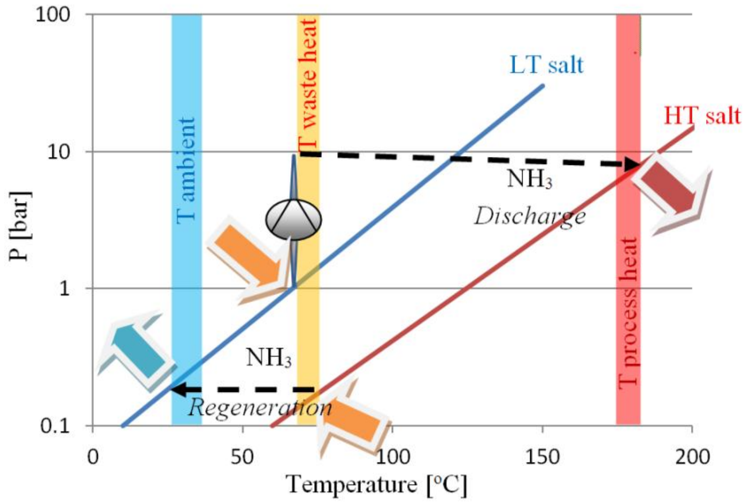

Figure 3: Thermodynamic cycle of the hybrid heat pump type II configuration.

The question remains whether this hybrid system is both technically and economically feasible. This paper looks into both aspects. The technical feasibility focuses on the availability of suitable sorbents for this cycle and into the primary energy savings potential. The latter is important as the main reason for applying such a heat pump must lie in reducing the primary energy usage and - provided the heat pump costs are low enough compared to energy prices - costs. The economic feasibility study looks into the costs related to the construction, integration and operating the hybrid heat pump. This should answer if and/or under what conditions a hybrid heat pump is a cost-effective solution.

Technical feasibility

The starting point for determining the optimal pair of sorbents for the hybrid cycle is a set of 50 ammonia salt reactions that included the entropy and enthalpy values for the reaction. This means that using the Clausius-Clapeyron equation, for each sorbent the relation between pressure and temperature can be calculated. By choosing a range for the ambient temperature (30 – 50 ^{o}C), the waste heat temperature (60 - 110 ^{o}C) and the output temperature (130 – 210 ^{o}C), the range of pressures for each sorbent can be calculated. By applying a minimum pressure at 0.1 bar and a maximum pressure at 60 bar, the number of sorbents suitable as LTS or HTS is reduced. The minimal pressure is required to avoid requiring very large and thus expensive compressors as their size correlates to volume flow rather than mass flow. The maximum pressure is chosen so no condensation of ammonia occurs during the discharge phase. Higher pressures would also further limit the compressor options.

Furthermore, a maximum pressure ratio for the mechanical compressor of 3 is applied. This value was chosen because the required work, in terms of required primary energy, for pressure ratios above 3, starts nearing the amount of primary energy saved by the hybrid heat pump. A pressure ratio of 3 is also considered the upper limit for dynamic compressors, which are in general more efficient than positive displacement compressors. An isentropic efficiency of 0.75 was used to determine the estimated work (*w*) of the compressor for given pressures.

The next step in the calculation involves determining the heat losses and the subsequent net heat output and COP values. The heat losses consist of losses due to heating up the ammonia and heat losses due to heating up the sorbent reactor. The heat losses of the reactor are related to heating of the salt and of the inert reactor-material. The contribution of the inert reactor mass (tubes, fins, thermal oil/steam) has been set to 4 J K^{-1} per g of salt. This is comparable with the inert mass-salts ratios used in an ECN prototype heat transformer (van der Pal et al., 2009). The required amount of salt depends on its molar weight, stoichiometry (ratio salt/ammonia in reaction) and the fraction of salt reacting per cycle. The latter was set to 0.7 which means that each cycle 70% of the salt changes from an adsorbed to a desorbed state and vice versa. Altogether this results in the following heat loss of the HTS-reactor (subscript hts) in the discharge cycle (J per mol NH_3):

$$Q_{\mathrm{loss,hts,dis}} = C_{\mathrm{p,NH_3}}(T_{\mathrm{h}} - T_{\mathrm{w}}) + \frac{n_{\mathrm{hts}} \cdot m_{\mathrm{hts}}}{\eta_{\mathrm{hts}}}\left(C_{\mathrm{p,hts}} + hc\right) \cdot (T_{\mathrm{h}} - T_{\mathrm{m}}) \qquad [1]$$

Similar equations were derived for the losses of the low temperature sorbent (subscript lts) and the regeneration phase (subscript reg). Together with the sorption enthalpies (ΔH), the heat input and output per phase can now be calculated:

$$Q_{\mathrm{out,dis}}(T_{\mathrm{h}}) = \Delta H_{\mathrm{hts}} - Q_{\mathrm{loss,hts,dis}} \quad [2]$$
$$Q_{\mathrm{in,dis}}(T_{\mathrm{m}}) = \Delta H_{\mathrm{lts}} + Q_{\mathrm{loss,lts,dis}} \quad [3]$$
$$Q_{\mathrm{in,reg}}(T_{\mathrm{m}}) = \Delta H_{\mathrm{hts}} - Q_{\mathrm{loss,hts,reg}} \quad [4]$$
$$Q_{\mathrm{out,reg}}(T_{\mathrm{l}}) = \Delta H_{\mathrm{lts}} + Q_{\mathrm{loss,lts,reg}} \quad [5]$$

The COP values can be calculated for both heat and work input from the work and the net heat input and output:

$$COP_{\mathrm{h}} = \frac{Q_{\mathrm{out,dis}}}{Q_{\mathrm{in,dis}}+Q_{\mathrm{in,reg}}} \quad [6]$$
$$COP_{\mathrm{w}} = \frac{Q_{\mathrm{out,dis}}}{w_{\mathrm{dis}}+w_{\mathrm{reg}}} \quad [7]$$

Table 1 shows the most favorable ammonia-salt reactions for heat-driven and hybrid systems based on efficiency and temperature range. Furthermore a minimum specific power output of 50 W kg^{-1} is used to restrict the number of potential salts to those that can provide sufficient power density. For the single compressor hybrid heat pump the highest performance is achieved when the compressor is placed in the discharge phase. The work applied to the compressor is then added as useful heat rather than reducing the required amount of waste heat for regeneration. Because the discharge phase always has a considerable higher pressure compared to the regeneration phase, it also requires a smaller and thus cheaper compressor. For hybrid operation the ammonia-salt reactions $CaCl_2(4\text{-}8)NH_3$ with $MnCl_2(2\text{-}6)NH_3$ and $CaCl_2(4\text{-}8)NH_3$ with $MnSO_4(2\text{-}6)NH_3$ are most favored. The $MnCl_2$ combinations can be used for a large temperature range, both in terms of waste heat temperature as well as useful heat output temperature, whereas the combination with $MnSO_4$ achieves high(er) efficiencies in a smaller temperature range.

Table 1: Most favorable ammonia-salt reactions for heat-driven and hybrid systems.

ammonia salt reaction		T_l	T_m	T_h	COP_h
lts	hts	°C	°C	°C	-
heat-driven					
$MnCl_2(2\text{-}6)NH_3$	$NiCl_2(2\text{-}6)NH_3$	30	110	180-200	0.22
$MnSO_4(2\text{-}6)NH_3$		40	100-110	170-210	0.19
$ZnCl_2(2\text{-}4)NH_3$		40-50	100-110	170-180	0.25
$ZnCl_2(4\text{-}6)NH_3$	$MgCl_2(2\text{-}6)NH_3$	30	110	180	0.25
single compressor					
$CaCl_2(4\text{-}8)NH_3$	$MnCl_2(2\text{-}6)NH_3$	30-50	80-110	150-210	0.18
	$MnSO_4(2\text{-}6)NH_3$	30-40	80-90	150-160	0.32
double compressor					
$CaCl_2(4\text{-}8)NH_3$	$MnCl_2(2\text{-}6)NH_3$	30-50	60-100	130-200	0.20
	$MnSO_4(2\text{-}6)NH_3$	40-50	80-90	150-160	0.33

Economic feasibility

The economic feasibility is determined by estimating the cash flow in comparison with the total capital investment (TCI). The total capital investment consists of fixed capital investment and additional costs. The latter is the sum of start-up costs, working capital costs, licensing and R&D funds and costs for capital during construction. The fixed investment costs are the sum of direct costs and indirect costs. Purchased equipment costs, installation costs, piping costs, instrumentation and controls costs, electrical equipment and materials costs, land costs, civil construction cost and service facilities costs are the components of the direct costs. Indirect costs consist of engineering and supervision costs, construction and supply costs and contingencies costs.
The cash flow consists of the sales minus the operating costs. The amount of steam produced by the heat pump amounts to sales. The operating costs consist of directs expenses that include utilities (electricity costs), operating labor, supervision, maintenance and insurance, and indirect expenses such as depreciation and taxes.

For the analysis of the economic feasibility a hybrid sorption-compression heat pump of the size of 10 MW heat output is considered.

Because it is common practice to calculate many of the TCI costs as a fraction of the purchased equipment costs (PEC), the PEC has a large effect on the value of TCI. Because the PEC strongly depends on the chosen configuration, it was decided to determine the PEC for a total of four different configurations, resulting from varying two parameters. These parameters are:

1) the presence of a secondary (oil)circuit versus a direct steam feed;
2) a reactor consisting of a standard priced shell-and-tube heat exchanger based on the numbers from the DACE-handbook (Webci and Wubo, 2006) versus a reactor consisting of a shell and tubing with pricing from a Dutch manufacturer.

The first parameter is chosen because an additional (oil) circuit requires additional heat exchangers that add considerably to the costs. The advantage of the secondary circuit compared to a direct steam feed is more freedom in reactor design, better temperature control and more effective use of the sorbents. The second parameter was chosen because significant differences were found between the cost estimates for a shell-and-tube heat exchanger based on the DACE-handbook and the (lower) costs estimates from a Dutch manufacturer for a reactor consisting of a shell and tubing. All configurations also include valves, compressor and tubing costs. Table 2 shows the TCI for these four different configurations for a hybrid heat pump with a size of 10 MW heat output.

Table 2: TCI for four configurations (in M€)

	Direct steam feed	Oil-circulation
Heat exchanger (DACE)	14.9	21.5
Heat exchanger (Manufacturer)	6.1	12.2

In order to determine the cash flow, a reference situation needs to be chosen in which prices for energy and heat pump performance should be decided. Table 3 shows the reference values. Table 4 shows the resulting cash flow, which equals about 2.6 M€ per year. Payback times, defined as ratio total costs investment/cash flow, range from 2 to 9 years. For configurations with less operating hours or lower COP electric, the payback time will increase considerably. On the other hand, (expected) higher energy prices will reduce payback times.

It is important to realize that these numbers only include the avoided energy costs as a benefit whilst there could be many other drivers for applying heat pumps, such as lower carbon taxes, receiving 'green' subsidies, complying with legal obligations toward higher efficiencies or reduced dependency on (variation in prices of) primary energy carriers.

Table 3: Reference values for the standard situation

Variable	Reference value
Electricity price	65 €/MWh
Steam price	25 €/ton of steam
COP electric	10 J of heat/J of work
Operating hours	8000 hours per year

Table 4: Cash flow (M€ per year)

	Direct steam feed	Oil-circulation
Heat exchanger (DACE)	2.51	2.51
Heat exchanger (Manufacturer)	2.74	2.67

Discussion and Conclusions

The hybrid adsorption compression heat pump concept can extend the range of applicable temperatures for a thermally activated sorption heat pump towards lower waste heat temperatures. This increases the amount of useable waste heat sources and creates more operational flexibility for the heat pump. From the analysis of various chemisorptions reactions, multiple sorbent working pairs have been identified that meet the pressure and temperature requirements as well as the power density. The integration of a mechanical compressor stage into a thermally driven sorption heat transformer is considered technically feasible.

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 further reduced.
Therefore, it is concluded that the hybrid adsorption compression heat pump concept is feasible from technical and economical point of view.
Based on the available amounts of waste heat for the Dutch refinery and chemical industry in the temperature range from 70 to 150 °C, about 70 PJ of waste heat could be used, resulting in a reduction of primary energy use of approximately 30 PJ per year. Reusing waste heat from other streams such as hot flue gases and other industries could even further increase this amount considerably.

Future work will address technical issues of the hybrid concept in more detail. This will deal with a.o. the combination of quasi-continuous operation of a chemisorption heat pump connected with a continuous mechanical compressor, the compressor requirements when dealing with hot-suction gasses and transient conditions. Also the selection of the type of compressor that fits best to the operational characteristics will be further studied. An important aspect for the chemisorptions reactors is to obtain a sorbent heat exchanger design that combines the requirements for high power density with low cost.

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Nomenclature list

$Q_{\mathrm{loss,hts,dis}}$	heat lost in discharge phase per mol of ammonia (J mol^{-1})
$C_{\mathrm{p,NH3}}$	heat capacity of ammonia (J mol^{-1} K^{-1})
T_{h}	temperature of useful heat (K)
T_{w}	temperature of the ammonia after compression (K)
n_{hts}	mol ratio hts/ammonia (mol mol^{-1})
m_{hts}	molar mass high temperature sorbent (g mol^{-1})
η_{hts}	fraction of salt reacting per cycle (0.7)
$C_{\mathrm{p,hts}}$	heat capacity of high temperature sorbent (J g^{-1} K^{-1})
hc	additional inert thermal mass per gram of high temperature sorbent (4 J g^{-1} K^{-1})
T_{m}	temperature of waste heat (K)
T_{l}	ambient temperature (K)

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