#2/17

DUTCH HEAT PUMPING
TECHNOLOGIES JOURNAL

Industrial Applications

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Industrial Heat Pumps in the Netherlands
No more cooling towers!

Industrial heat pumps are defined as active heat-recovery apparatus that increases the temperature of an excess heat stream in an industrial process to a higher temperature to be used in the same process or another adjacent process or heat demand. The heat pump is used to increase the energy efficiency of the process and in a lot of cases also increases the process efficiency, quality and yield of the product. Industrial heat pumps are in most cases not considered as renewable but as energy conservation rechnologies.

To increase the temperature with an industrial heat pump an external source of energy is used as driving force. This can be electricity, gas or another fossil fuel or thermal energy (i.e. heat). In Industry the most common applications are: Mechanical Compression; Mechanical Vapour Recompression, Thermal Vapour Recompression and Absorption. Each of these heat pumps have specific characteristics and application areas.

In the Netherlands heat pumps of different types can be applied in all levels of industry ranging from bulk distillation in chemical industry to the level of milk processing at the farm or growing tomatoes in greenhouses and steamproduction in paper and pulp. In the application the approach should be based upon the Trias Energetica.

The potential with industrial heat pumps for energy conservation and reduction of CO2-emissions in the Netherlands is enormous and at this moment not naturally a part of policy papers nor for a large part under the attention of industry itself. On the other hand there is a lot of attention for the use of industrial waste heat (i.e. excess heat from a production process) to be used in heat distribution systems. Real residual heat consists at least at four places in the Netherlands where the energy-intensive industry is strongly concentrated. However, according to Tjeerd Jongsma, Director of the Institute for Sustainable Process Technology (ISPT), it is more obvious to use the residual heat first in the industry itself, then only to use it for heating homes. "If industry is to use heat pumps on a large scale, this will achieve our goal faster in reducing natural gas consumption and CO2 emissions, without having to invest in fine distribution grids and expensive heating systems." A first, most logical, therefore to this challenge of excess heat is to reuse the heat within the same process through process integration or at the same site. In an ideal process that will be within the process unit, otherwise technology will have to be applied to transform the heat coming out of the process to a common carrier. This being high pressure steam by a high temperature heat pump or or electricity generated by an ORC.

In the Netherlands many projects on these topics are running, some still in the phase of R&D supported by ISPT, others already demonstrated and state of the art. This issue of the Dutch Heat Pumping Technologies Journal will show you the various innovative developments in the Dutch market showing that the potential is enormous and that excessheat does not have to be destroyed in cooling towers.

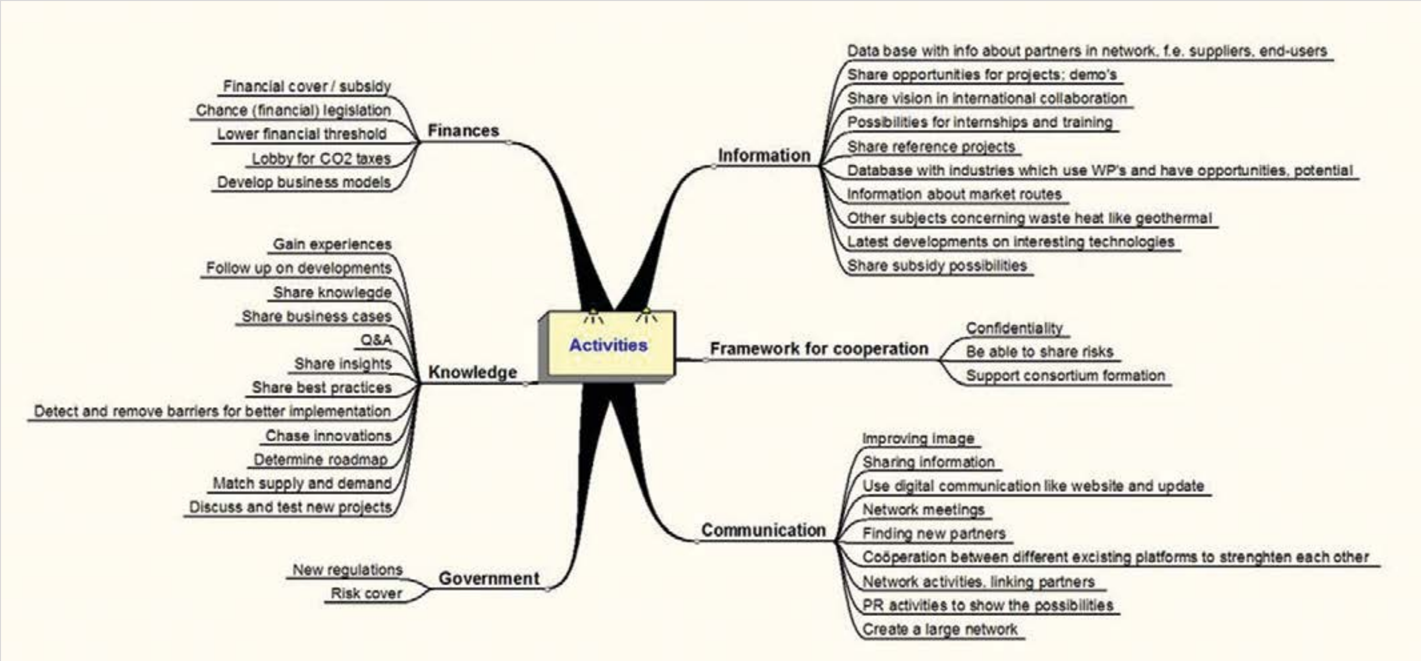
Platform for the accelerated uptake of Industrial Heat Pumps in the process industries in The Netherlands

Unlocking the Industrial Heat Pump value chain

Author: Andreas ten Cate – Director International Business Development,
Klaartje Rietkerken – Program Manager Heat and Utilities

A major part of the energy used in industry is applied as industrial heat at high temperatures (from 80 °C to more than 1000 °C). In a traditional operation the heat is generated by fossil fuels and cascaded down to high volume low quality heat which is usually discarded into the environment.

There is an urgent need to increase the efficiency of the use of heat in industry on the road to the Paris targets of 80 to 95% reduction in CO2 in 2050 There is a pivotal role for heat pump technology but at this moment the industrial implementation of these technologies is lacking. In the first half of 2017 ISPT is conducting a study among the key parties active in the value chain of the technology to assess and re-confirm the presence of persistent barriers in adoption of the technology and to evaluate how a platform of all stakeholders along the value chain can support an accelerated adoption. ISPT together with its subcontractor Industrial Energy Experts, have interviewed 27 parties from end users, to technology providers and heat pump manufacturers, to engineering firms and ECP contractors, to governmental bodies.



From the interviews several observations were made:

- There is a substantial industrial need for systems above the 1 MW scale.
- Steam generating closed cycle heat pumps up to 160 °C will fill a major market need for water based processes e.g. in food and paper & pulp sectors, for the chemical industries there is a need for technology that can exceed 200 °C.
- Up to 100 °C closed cycle heat pump technologies are readily available but lack industrial demonstration at large scale. Steam generating heat pumps that deliver low to mid-pressure steam over 100 °C are developed in several industry and research organizations and are at the brink of introduction.
- For higher temperature applications combinations with mechanical vapor recompression will need to be developed further.
- The business case in brown-field applications is difficult as payback times currently are at 5 to 7 years. Capex reduction, standardization and demonstration cases are needed to build a track record and kick-start the learning curve.

In follow-up to the interviews a group workshop was held with more than 50 participants representing all actors in the value chain, who were challenged to discuss and explore how the platform can contribute to acceleration of the technologies. The participants were asked to think about technological projects as well as non-technological activities that can support the adoption of the technology. In group discussions a large set of project ideas were generated for development and demonstration of heat-pump embedded systems in various applications. Some examples are:

- Heat pump integration in drying systems
- A system for recovery of heat from waste water for flexible generation of steam in various qualities
- Waste heat recovery systems for cleaning processes
- Recovery of latent heat from fractionation systems and from condensation/evaporation systems for drying or distillation

Another important finding of the workshop was that there is a clear need for a central place where overview of the information is found and where also the peers in the field are able to meet and share insights across the different technological areas and industrial sectors of origin. Examples of ideas for non-technological activities are summarized in the mind-map.

This year further actions will be taken to carry on building the network to support the platform. All important actors currently active in the field will be invited to contribute to the shared goal of a strong platform where essential information can be found, peers meet and ideas are shared. Join us if you are interested to contribute or want to know more about the opportunities and contributions that heat pump technologies present to create a sustainable industry on the road towards 2050. Check our website (www.ISPT.eu) or reach out to us at klaartje.rietkerken@ispt.eu.

Author: Edited by Onno Kleefkens - Phetradico, photo: www.shutterstock.com

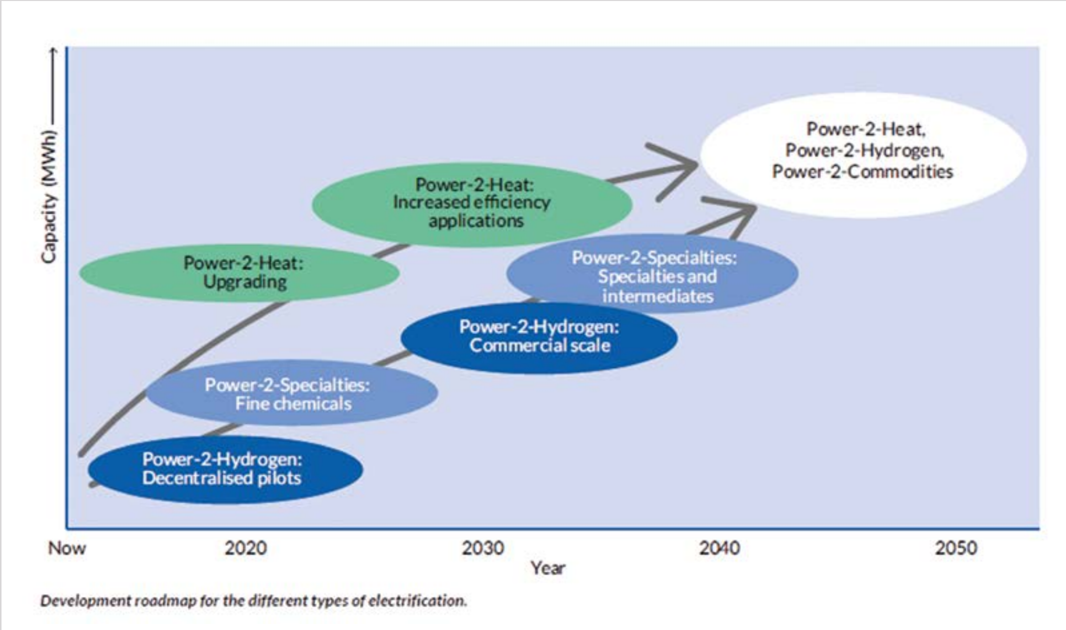
Empowering Chemical Industry,
opportunities for electrification

Societal awareness to become more sustainable is growing. Consumers are asking for more sustainable products, and regulations are being devised to ensure the implementation of measures fostering more sustainable production. This puts further pressure on the industry. In 2011, EU leaders endorsed the long-term objective of reducing Europe's greenhouse gas emissions by 80-95% by 2050, as compared to 1990 levels, to mitigate climate change to acceptable levels. This has since been followed up at COP21 in December 2015, with the collective aim to limit global warming to well below 2°C, and supporting initiatives, such as Action 2020 from the World Business Council for Sustainable Development. As a result, there is a need to rapidly decarbonise the EU's energy and feedstock supply. With this prospect, sustainability has become part of the mission statement and strategy of many chemical companies. CO2 reduction, more efficient use of energy and feedstocks and reducing waste are key to turning the need for sustainability into practice.

Recent progress in increasing the sustainability

Between 1990 and 2013, the amount of energy consumed by all end-use sectors in the EU increased by 2.2 % annually. This has offset the positive environmental impacts of improvements in the energy production mix and of other technological developments that were achieved in the same period. The share of final energy consumption of the industry in the total final energy consumption in Europe has gradually decreased from 34% in 1990 to about 25% in recent years.

While the total production of chemicals has increased by 59% since 1990, the chemical industry in Europe now uses just half of the energy to produce a unit of chemicals, as compared to 22 years ago. To a large extent, this



Source: Shared Innovation Program VoltaChem

reduction can be attributed to the use of combined heat and power units, the shift to higher value-added and lower energy intensive products, and continuous process improvements. The largest change has been in the use of natural gas that was reduced by 30% between 1990 and 2016. In the same timeframe, electricity demand has also been reduced by 12%, while oil use has increased by 5%.
Though much has been achieved so far, substantial efforts are still needed to reach the short and long term EU GHG objectives (e.g. 20% reduction by 2020).

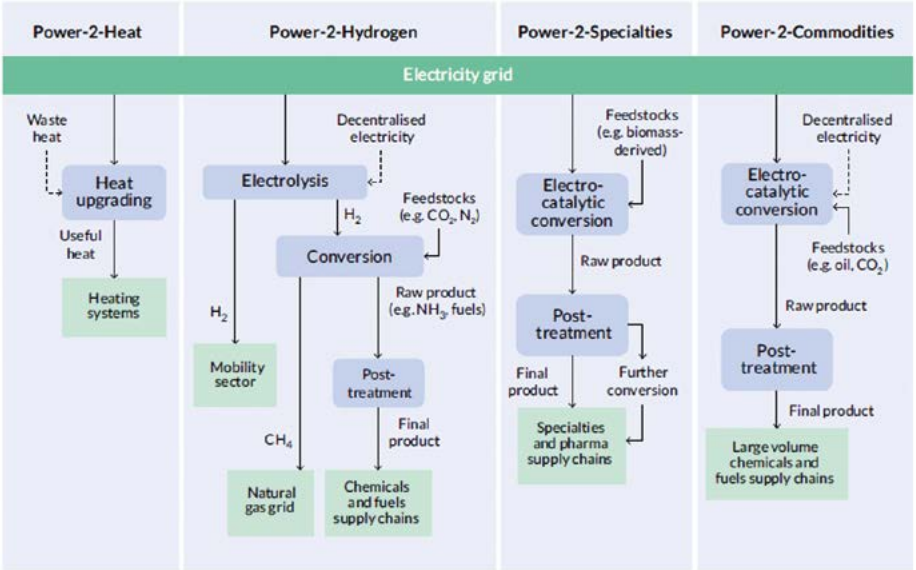
The landscape of power generation is changing drastically in the next decades

The key challenge in achieving a carbon-neutral energy and production system is to decarbonise the sectors that are currently heavily dependent on fossil fuel resources, such as oil and natural gas. The most promising option for future decarbonisation of final energy and feedstock use in the chemical industry is to convert the relatively abundant potential of wind and solar energy – produced in the form of electricity – into heat, chemicals and fuels. Electrification has the potential to realise major progress on sustainability and reduction in fossil energy and feedstock use.
The shift from fossil to renewable sources will be accompanied by a shift from centralised generation of electricity in high capacity power plants to decentralised generation with more low capacity locations. Wind and solar energy are mainly harvested as electricity, by means of wind turbines and photovoltaic (PV) solar cells. The supply of these renewables fluctuates continuously and is non-controllable. With increasing shares of renewables in the total electricity production, the magnitude of fluctuations is increasing.
Both effects result in a growing need for flexibility in the electricity system.
As the need for flexibility increases, other options will also be required. These include demand side response measures, storage, further expansion and upgrading of electricity grids and curtailment of renewables generation.

The following types of electrification can bedistinguished:
Power-2-Heat: the efficient generation and upgrading of heat and steam with electricity for use in chemical processes;

- Power-2-Hydrogen: the use of electricity to produce hydrogen through the electrolysis of water, which is subsequently used for a number of applications;
- Power-2-Specialties: the direct electrochemical synthesis of high value fine and specialty chemical intermediates and products using conventional and biomass-derived feedstocks;
- Power-2-Commodities: the direct electrochemical synthesis (both centralised and decentralised) of large volume commodity chemicals using conventional and sustainable feedstocks, such as CO2.

To retain a solid competitive position of the EU chemical industry and to fully utilise the potential of electrification, innovation and implementation should be accelerated. To achieve this, a number of barriers should be overcome. Industry representatives named as a main barrier the fact that many applications of electrification currently seem economically unfeasible. To solve this, low cost technolo-



Schematic representation of the four different types of electrification.

Source: Shared Innovation Program VoltaChem

gies should become available at industrial scales. Since downtime should be prevented, the availability of proven technology is also vital. Moreover, using fossil fuels is currently very inexpensive, which prevents companies from choosing more costly sustainable alternatives.

To overcome these and other barriers, stakeholders will have to cooperate and balance their interests in order to get the flywheel up to speed. An ecosystem should evolve in which stakeholders work together on topics like technology development, innovative business models and assessment of economic feasibility. All stakeholders will benefit when innovation in the field of electrification accelerates, because electrification fits with their individual interests.

A roadmap has been developed in which the various technologies are expected to be realised at different timescales, based on maturity levels and wide-ranging business cases. The first applications will be those with more positive economics, such as waste heat upgrading and electrochemical production of fine chemicals. With more technical developments, higher volume products and more advanced applications of Power-2-Heat, such as technologies which not only utilise waste heat but reduce the heat demand altogether, will become feasible. At the same time, Power-2-Hydrogen will expand from the pilot scales that are being developed today to commercially relevant scales. Though it will take time, maybe even decades, the long-term perspective of a highly electrified chemical industry where renewable feedstocks and energy are the basic ingredients of chemicals, including specialties, commodities, and fuels, is very appealing to many stakeholders.

Power-2-Heat is expected to be the first type of electrification that the chemical industry will implement on a large scale

With Power-2-Heat, electricity is used to either generate heat directly or to upgrade steam and waste heat for efficient (re-)use in chemical processes. Traditionally, heat is generated by burning natural gas, but new technologies are making replacement by heat generation with electricity more attractive. With both efficiency gains and the re-use of waste heat as process heat, the overall energy requirements of a process can be reduced.

The main drivers for realising Power-2-Heat are cost efficiency and sustainability. Low electricity prices compared to natural gas prices make electricity a cost efficient alternative to natural gas, especially when combined with options to flexibly apply Power-2-Heat at moments when electricity prices are low. Electrically driven heat pumps can produce heat in a very cost effective way by upgrading waste heat to useable temperature levels. This recycling of waste heat fits into the circular economy concept, leading to energy savings and CO2 emissions reduction, and substitutes a fossil energy carrier by electricity. Power-2-Heat technologies have a fast response time and are able to cope with fluctuations in electricity availability and prices. Investments in technology and making the infrastructure suitable for delivering large quantities of electricity should be compensated by low electricity prices and a higher energy efficiency.

Electrically driven heat pumps in industry will become an important means for re-using waste heat as process heat. Significant savings potential can be reached within the coming years when current technology is further developed to achieve higher temperature levels and temperature lifts to make heat pumps more broadly applicable. Business cases are now being evaluated for heat pumps and other technologies. For example Dow Benelux completed an economic feasibility study of the use of mechanical damp recompression, where steam is upgraded by using electricity7. The business case appears to be attractive and Dow is now considering piloting the technology and testing the technical reliability of the concept. As more such cases become known and are proven in industry, Power-2-Heat is expected to be the first type of electrification that the chemical industry will implement on a large scale.

WHAT ARE THE MAIN DRIVERS FOR ELECTRIFICATION?

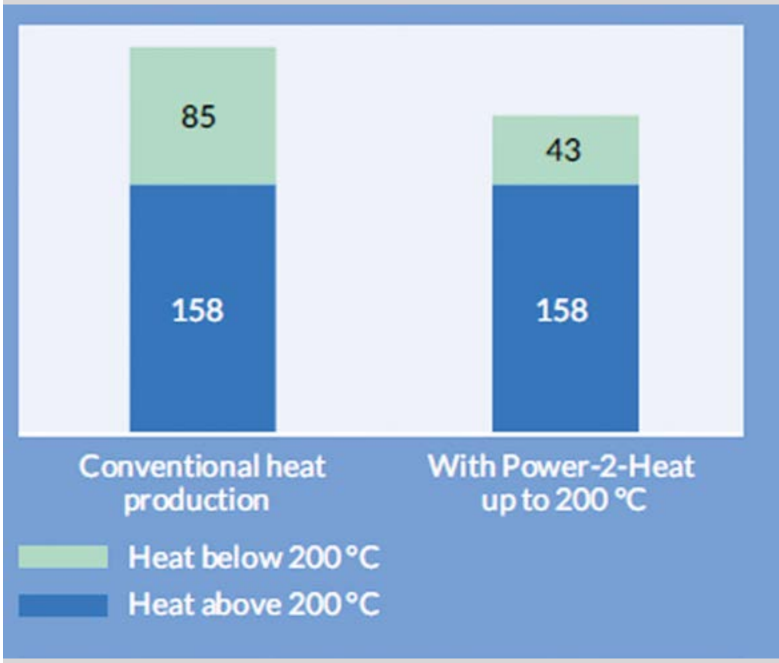
An industry consultation was carried out to obtain insights into the drivers and barriers for electrification and into the impact electrification may have on the industry.
During the industry consultation, chemical companies were asked for their drivers for electrification. The main drivers that were named are the following:

- Economic benefits, because of cost reductions by using cheap energy and by making processes more efficient and selective, thus leading to higher margins. This driver was named by each interviewee.
- Improved sustainability through the reduction of feedstocks, by-products, waste, energy use, solvents and CO2 (named by 70% of interviewees).
- Development of new products by applying innovative electrochemical technologies (named by 20% of interviewees, especially those who are interested in Power-2-Specialties).

Other interviewees (mainly from the energy sector) also named the following drivers for electrification:

- Easier realisation of flexibility with electricity (e.g. local production near electricity sources);
- Optimisation of investments in the electricity grid e.g. by matching local supply and demand;
- Increased stability in the electricity grid;
- Reduced dependence on (foreign) natural gas;

(this article is based with direct citations upon the whitepaper by the Shared Innovation Program VoltaChem: Empowering the Chemical Industry, opportunities for electrification - TNO Sustainable Chemical Industry — Karin van Kranenburg, Erin Schols, Hugo Gelevert, Robert de Kler; ECN Energy research Centre of the Netherlands — Yvonne van Delft, Marcel Weeda; May 2016;



Source: Shared Innovation Program VoltaChem

Power-2-Heat in the chemical industry could result in about 15-20% energy savings for heat production. In 2013, the final energy use for heat in the chemical industry amounted to 243 PJ. About 35% of the required energy is for heat at temperatures up to 200 °C. These temperatures can be obtained by heat pumps and upgrading of residual steam by mechanical vapor recompression. It is estimated that a 50% energy savings is possible through application of these technologies, resulting in 15-20% savings on energy for heat demand, as shown in the figure below.

Full deployment of Power-2-Heat technologies for generating heat up to 200 °C could lead to a reduction in CO2 emissions in the chemical industry of about 6 Mtonne. The maximum effect is achieved if all of the electricity is produced by additional generation capacity based on wind and solar photovoltaics.

Power-2-Heat could offer an important source of adjustable load for demand response schemes at local scales When operating at full load, a Power-2-Heat capacity of more than 1 GW is required. This represents more than 25% of the currently installed 3.9 GW of wind turbines and PV-panels, and about 4% of the expected 30 GW of installed wind and solar-based generation capacity in 2030. Clearly, this presents a significant source of adjustable load to help compensate for fluctuations in intermittent supply on a national level, and in particular on local or regional levels. It could thus contribute to minimising investments in expansion of the electricity grid associated with the deployment of intermittent renewables.

THE ENERGY CONSUMPTION OF REFRIGERATION INSTALLATIONS

AND HEAT PUMP POTENTIAL IN 40 DUTCH INDUSTRIAL SECTORS

ABSTRACT

A study by KWA Business Consultants has been carried out into the annual energy consumption of refrigeration installations in the Netherlands. The study shows the energy consumption of detailed sectors that use refrigeration. A large number of industrial sectors (such as chemical industry, dairy, beer, cold stores) has been distinguished and also various non industrial sectors (such as buildings air conditioning, supermarkets, agricultural sectors). With knowledge of these figures innovate companies may put focus on energy savings by improvement of refrigeration components, systems and controls in particular sectors.

Refrigeration installations deliver residual heat. This heat amount can be supplied to heat processes and buildings at moderate temperatures. By use of heat pumps this residual heat can be raised in temperature and reused more effectively. The use of fossil fuels for heating purposes is reduced significantly. This saving potential is identified in this study.

The food and drug industry sectors have an electrical energy consumption of 23 PJ in total. Refrigeration consumes 6 PJ, which is a share of 26%. The other, heavier industry sectors have an electrical energy consumption of 116 PJ in total. Refrigeration consumes 13 PJ, which is a share of 11%.

The non-industrial sectors have an electrical energy consumption of 42 PJ in total. Refrigeration consumes 10 PJ, which is a share of 24%.

1. INTRODUCTION

KWA Business Consultants has carried out hundreds of energy studies in the Dutch industry over the last 15 years. These studies analyze the energy distribution of electricity and heat into the consumers such as boilers, production processes, refrigeration plants, compressed air, HVAC. Many energy conservation studies of companies are representing industrial branches, therefore it is possible to distinguish the refrigeration consumption of each industrial and various non industrial sectors. The government in the Netherlands has set up programs for energy conservation in the industry by Long Term Agreements with industrial and non-industrial sectors. They are summarized for the sectors that use significant

Author: PENNARTZ, VAN DEN BOVENKAMP, photo: www.shutterstock.com

Sector	Total energy consumption	Total electricity consumption	Electricity in % of total	Average % refrigeration of electricity use	% Refrigeration of total energy consumption	Electricity use for refrigeration per year
	PJ	PJ	%	%	%	PJ/year
Food and drug industry						
Breweries	2,4	0,8	33%	28%	9%	0,2
Potato processing industry	6,6	2,0	30%	55%	17%	1,1
Cacoa industry	1,7	0,5	28%	14%	4%	0,1
Animal food industry	2,7	1,6	60%	5%	3%	0,1
Soft drinks	0,8	0,3	41%	5%	2%	0,0
Fruit and Vegetables	1,7	0,5	27%	25%	7%	0,1
Coffee production	0,8	0,2	28%	15%	4%	0,0
Margarine, fats and oil	7,0	2,0	29%	25%	7%	0,5
Flour production	0,6	0,3	58%	2%	1%	0,0
Meat processing	2,2	1,2	52%	45%	24%	0,5
Dairy	13,5	4,8	36%	35%	12%	1,7
Backeries	3,7	0,7	19%	45%	9%	0,3
Fish processing	0,5	0,3	56%	50%	28%	0,1
Confectionary, ice cream	1,5	0,3	21%	43%	9%	0,1
Sugar	5,5	0,4	7%	2%	0,1%	0,01
Starch production	6,4	1,7	27%	3%	1%	0,1
Other food and drug	24,2	5,1	21%	20%	4%	1,0
Total food and drug	**81,8**	**22,8**	**28%**	**27%**	**7%**	**6**
Industry						
Chemical industry	281,1	54,6	19%	12%	2%	6,6
Refineries	142,8	4,4	3%	15%	0%	0,7
Oil and gas production	37,5	13,3	35%	18%	6%	2,4
Rubber- en kunststofindustrie	5,2	3,2	61%	12%	7%	0,4
Other industry in coventant	22,6	7,9	35%	8%	3%	0,6
Refrigerated warehouses	1,2	1,1	92%	90%	83%	1,0
Industry without refrigeration	100,4	18,8	19%	1%	0%	0,2
Other industry non covenant	21,1	13,0	62%	8%	5%	1,0
Total industry	**612,0**	**116,2**	**19%**	**11%**	**2%**	**13**
Services, agricultural sectors, non industrial						
Super markets	6,3	3,1	48%	88%	42%	2,7
Universities	1,1	0,6	50%	9%	9%	0,1
Medical centres academic	3,4	1,5	43%	9%	9%	0,1
Universities academic	3,4	2,0	61%	16%	16%	0,3
Hospitals	13,1	6,6	51%	16%	8%	1,1
ICT-sector	7,0	6,6	95%	22%	96%	1,5
Hotels, restaurants	26,6	7,4	28%	15%	4%	1,1
Smaller commercial cooling		2,8		12%	0%	0,3
Ice rinks, ski rinks	0,3	0,2	62%	80%	49%	0,1
Offices with HVAC	34,5	10,8	31%	17%	5%	1,8
Mushroom production	0,4				27%	0,1
Flower bulbs						0,1
Cooled green houses						0,4
Farmers cooling tanks						0,4
Total	**96,1**	**41,5**		**24%**	**11%**	**10**
Total all sectors	**789,9**	**180,5**				**29**

Table 1: Annual energy consumption of refrigeration installations per sector

Sector	Electricity use by refrigeration per year	Electricity use for refrigeration per year	Average COP of refrigeration	Released condensing heat (T=40 °C)	Heat Pump on 40% condensing heat at 80°C
	MWh/year	PJ/year	-	PJth/year	PJth/year
Food and drug industry					
Breweries	62.657	0,2	3,3	1,0	0,5
Potato processing industry	308.117	1,1	2,2	3,5	1,8
Cacoa industry	18.057	0,1	3,5	0,3	0,1
Animal food industry	22.270	0,1	4,0	0,4	0,2
Soft drinks	4.421	0,0	3,5	0,1	0,0
Fruit and Vegetables	33.360	0,1	2,5	0,4	0,2
Coffee production	9.625	0,0	3,0	0,1	0,1
Margarine, fats and oil	141.988	0,5	2,5	1,8	0,9
Flour production	1.844	0,0	4,0	0,0	0,0
Meat processing	147.485	0,5	3,0	2,1	1,1
Dairy	467.500	1,7	3,5	7,6	3,8
Backeries	90.000	0,3	2,5	1,1	0,6
Fish processing	34.950	0,1	2,5	0,4	0,2
Confectionary, ice cream	38.222	0,1	2,5	0,5	0,2
Sugar	2.222	0,01	4,0	0,04	0,0
Starch production	14.351	0,1	4,0	0,3	0,1
Other food and drug	283.405	1,0	3,0	4,1	2,0
Total food and drug	**1.680.477**	**6**		**24**	**12**
Industry					
Chemical industry	1.820.577	6,6	3,5	29,5	14,7
Refineries	181.361	0,7	3,0	2,6	1,3
Oil and gas production	663.703	2,4	3,0	9,6	4,8
Rubber- en kunststofindustrie	106.137	0,4	4,0	1,9	1,0
Other industry in coventant	175.289	0,6	3,5	2,8	1,4
Refrigerated warehouses	287.359	1,0	2,0	3,1	1,6
Industry without refrigeration	52.148	0,2	3,5	0,8	0,4
Other industry non covenant	288.024	1,0	4,0	5,2	2,6
Total industry	**3.574.598**	**13**		**56**	**28**
Services, agricultural sectors, non industrial					
Super markets	746.044	2,7	3,0	10,7	5,4
Universities	13.912	0,1	4,0	0,3	0,1
Medical centres academic	37.109	0,1	4,0	0,7	0,3
Universities academic	90.385	0,3	4,0	1,6	0,8
Hospitals	293.795	1,1	4,0	5,3	2,6
ICT-sector	405.810	1,5	4,0	7,3	3,7
Hotels, restaurants	308.333	1,1	4,0	5,6	2,8
Smaller commercial cooling	93.333	0,3	3,0	1,3	0,7
Ice rinks, ski rinks	35.556	0,1	2,5	0,4	0,2
Offices with HVAC	500.000	1,8	4,0	9,0	4,5
Mushroom production	33.333	0,1	3,0	0,5	0,2
Flower bulbs	22.222	0,1	3,0	0,3	0,2
Cooled green houses	122.222	0,4	3,5	2,0	1,0
Farmers cooling tanks	121.111	0,4	2,5	1,5	0,8
Total	**2.823.166**	**10**		**47**	**23**
Total all sectors	**8.078.241**	**29**		**126**	**63**

Table 2: Condensing heat for reuse for process and buildings

Sectors with refrigeration	Heat consumption	Electricity consumption	Electricity by refrigeration	Condensing heat 40° C	Usefull share	Savings natural gas
	PJ/y	PJ/y	PJ/y	PJ/y	Usefull share	nm3/y
Food and drug industry	59,0	22,8	6,0	23,8	40%	300.700.000
Industry	495,8	116,2	12,9	55,5	40%	702.000.000
Services, agricultural sectors, non industrial	54,6	41,5	10,2	46,5	40%	588.000.000

Table 3: Available low temperature condensing heat

amounts of refrigeration in following groups:

Industrial sectors: chemical industry, cold stores, rubber industry, oil- and gas producers, potatoes processing, cocoa, fruit & vegetables. oil and fats, meat processing, dairy, beer and soft drinks, bakeries, confectionary.

Non industrial sectors: buildings air conditioning, ice rinks, data centers, hospitals, large offices, restaurants, supermarkets, agricultural sectors, green houses, flower bulb production, transport sector, household refrigerators, farmers cooling tanks.

The energy consumption of the sectors mentioned are published in PJ yearly final energy use for fuel and electricity. This means that data for electricity are presented by a conversion factor of 1 kWh = 3,6 MJ final energy use. Notice that in other studies electricity is often presented in primary energy use, in which the conversion and distribution losses are included. In the Netherlands predominantly a conversion factor of 1 kWh = 9 MJ primary energy use is used.

The key goals of the study are to help the refrigeration and air conditioning industry to take practical steps to reduce energy consumption by increasing efficiency of refrigeration and reuse heat by heat exchangers and heat pumps by:

- Raising awareness and understanding of refrigeration energy consumption in various sectors
- Providing information on condensing heat available and heat usage

2. The Annual energy consumption of refrigerastion installations for various sectors

As a result of the study, the share of refrigeration energy consumption for the various sectors is given in table 1.

The overview of table 1 shows the important sectors that use energy for refrigeration. One can focus on these sectors in order to stimulate energy conservation with the largest impact.

The value of the study is that refrigeration contractors and manufacturers of refrigeration components can focus on innovation of system design, control systems and apparatus, dedicated for the specific refrigeration installations in that sector. New designs will improve the energy efficiency of the various sectors. Each sector has a different refrigeration application that requires a specific installation design. The impact of improvements of energy efficiency can thus be related to this study. From table 1 can be concluded that energy saving programs in supermarkets will have a high effect.

A large consumer of electricity by refrigeration, not mentioned in table 1, are the household refrigerators with an annual electrical energy consumption of 14 PJ final use.

3. The REuse of condensing heat directly of by heat pumps in relation to The heat demand

Refrigeration installations can also deliver heat from their condensers to heat up processes and buildings. The potential of these heat pumps in various sectors has been calculated from table 1 by similar analysis, see table 2.
The long term goal is to replace a large number of heat boilers by heat pumps in the future to increase the energy efficiency for heating and reduce the consumption of fossil fuels. Electrically driven heat pumps are able to reduce the CO2 emission for heating (fossil fuel consumption) by more than 50%.
The data in the tables are based on absorbed electricity by compressors. The amount of electricity use of refrigeration is generally larger by the absorbed power of auxiliary pumps and fans. On a yearly basis, auxiliary equipment may consume 15% to 30% of the total energy consumption by refrigeration.
Table 2 shows the available low temperature condensing heat that may replace the heat consumption by buildings and processes mentioned in the second column of the table. The 40°C can be raised by high temperature heat pumps up to 85°C, still more than 50% reduction of primary energy use by heating. For instance, if 40% of the 23,8 PJ condensing heat in the food and drug industry is able to reduce the required heat of 59 PJ, the heat demand will be reduced by 9,5 PJ, equivalent to a saving of 300 million m3 of natural gas (see table 3).
High Temperature (HT) heat pumps that bring condensing heat at a temperature of 80°C as a second compression stage on an refrigeration plant, contribute even more to reuse of heat, since more applications are possible. Table 2 shows the PJ produced at 80°C by the additional heat pump ("add on") in case 40% of the regular condensing heat is reused. These PJ replace the PJ for heat consumption in table 3, but now at 80°C.

4. TRENDS AND OPPORTUNITIES

Optimisation of refrigeration installations and reuse the residual heat contributes significantly to the sustainability goals of many companies. The key to successful implementation for heat reuse by heat exchanger and heat pumps is the knowledge about the existing heating processes. The reused heat should fit perfectly in the heat demand side as far as timing, power and temperature are concerned. The heating processes are generally not known in practice nor by contractors, nor by the end users. They are not properly monitored. Pinch studies may contribute to this knowledge, but require actual data. Process designers should find solutions to lower the maximum heating temperature of the production process. This allows not only more heat recovery but also increases the recovery efficiency. To match demand and supply of heat in time and space, heat networks inside production areas, between productions sites and in turn the surroundings (buildings) allow more availability of reuse and increase the feasibility of heat recovery. In the long term the need for fossil fuels will decrease, providing all the benefits herewith.

Challenges for various parties in this respect are:

- Contractors: specialize on an industrial sector, develop knowledge on dedicated heat applications, apply innovation and control strategies.
- Manufacturers of equipment: develop more efficient heat pumps for larger operating ranges, heat exchangers.
- End users: integrate processes by matching heat demand and supply (refrigeration), increase understanding of own heating processes apply the Total Cost of Ownership principle.
- Policy makers: stimulate knowledge development, provide financial triggers of the use of control equipment and measuring & sensor technology.

5. RESULTS and CONCLUSIONS

The food and drug industry sectors have an electrical energy consumption of 23 PJ in total. Refrigeration consumes 6 PJ, which is a share of 26%. The other, heavier industry sectors have an electrical energy consumption of 116 PJ in total. Refrigeration consumes 13 PJ, which is a share of 11%.
The non-industrial sectors have an electrical energy consumption of 42 PJ in total. Refrigeration consumes 10 PJ, which is a share of 24%.

The use of condensing heat from refrigeration installations provides an good opportunity to reduce the CO2 emission by the heating of buildings and processes. High temperature heat pumps (up to 90°C) as a second stage on refrigeration installations will contribute significantly to this energy savings and reduce the use of fossil fuels.
The methodology used shows that from the energy consumption figures for refrigeration, residual heat of condensers can be derived. New developments can refer to these data to establish optimized designs for a sustainable refrigeration and heat pump technology in the future.

Flexible heat supply
and sustainability

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photo: www.shutterstock.com

Abstract
Because of the growing share of wind and solar energy in our electricity supply the electricity rates will be more volatile. In Germany, even negative rates occur regularly. Industrial companies can benefit from fluctuating electricity prices by using combined heat and power (CHP) during high rates and heat pumps and vapor recompression at low electricity rates. This hybrid energy supply leads to a robust energy supply system with heat pumps whereby companies are less vulnerable to fluctuations in the electricity market and are more attractive for shareholders with long term targets.

1. Sustainability and politics
Almost the whole scientific community recognizes our CO2-emissions lead to unacceptable environmental impact with huge economic and demographic consequences. In Western Europe politicians decided to price CO2-emis-

sions with the Emission Trade System1 in order to phase out fossil fuels. This emission trade system does not function yet because of the generous provision of free rights and exemptions for firms competing with non-European countries. The current tariffs are only a small part of the energy costs and do hardly effect investment decisions to replace fossil fuel by sustainable alternatives. For achieving their reduction targets countries stimulate replacement of fossil fuels by sustainable alternatives by subsidies on investments and feed-in tariffs. These last instruments are more successful as shown in the spectacular growth of sustainable electricity in European countries (fig. 1).

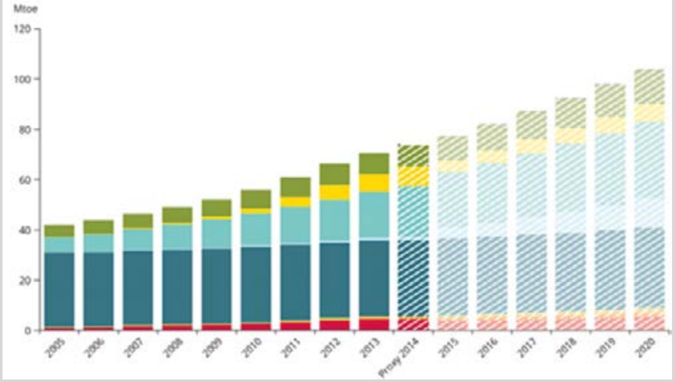

Figure 1

2. Volatility of prices

Especially the growth of wind and solar energy lead to challenges how to balance production and consumption. With a surplus of production of wind- and solar the electricity tariff drops to even negative values in a country as Germany (fig. 2). With a sudden shortage the price increases.

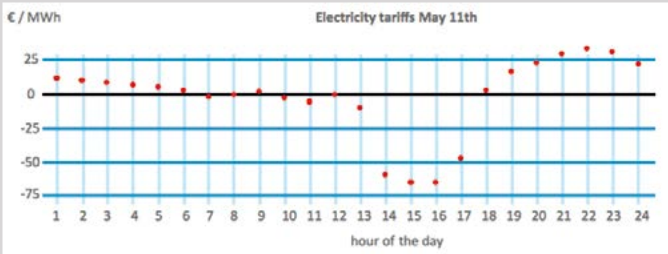

Figure 2

Not only Germany but also countries with a great share of hydro power have to handle surplus of electricity, for example by resistance heating in a steam boiler as seen in Norway (fig. 3).

Figure 3

3. Hybrid energy systems

The volatility of the tariff of electricity leads to a situation industrial firms are challenged to develop a hybrid energy supply. In situations with high power prices CHP is an attractive system. In situations with low or even negative tariffs electrical heating with resistance heating or electrical heat pumps is preferable. The switch between heat pumps and CHP must be possible within hours because the availability of wind and solar changes over daytime as does the energy consumption. Flexibility is key.

Though the share of wind and solar will increase the expectation is that the negative tariffs will vanish because market parties invest in storing cheap electricity in batteries, cold stores or heat buffers. Also, in some industries, the production of (semi-finished) products is possible if a production line has overcapacity.

4. Shareholder value

Shareholders with long term targets prefer firms with a high level of sustainability. Firms depending too much on fossil fuels are vulnerable for sudden increase in energy tariffs. Uncertainty leads to lower shareholder value. Besides, consumer markets are sensitive for environmental aspects. So most of the firms have ambitions to go sustainable (corporate social responsibility). Some of them seize opportunities in the neighborhood like heat of waste incinerators or waste steam of exothermal processes. The future worldwide availability of biomass is uncertain.

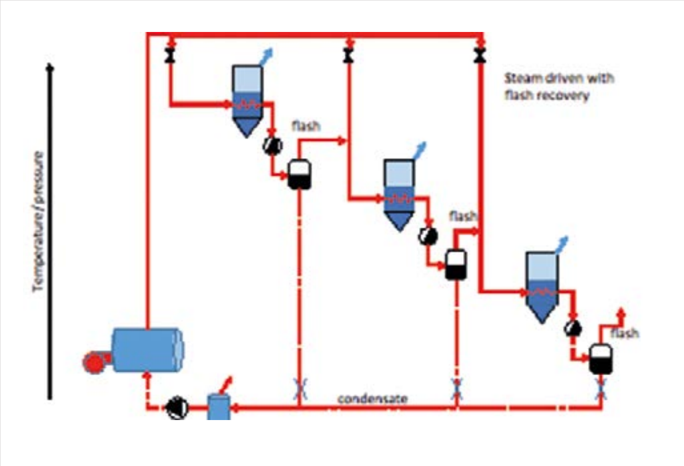

Figure 5

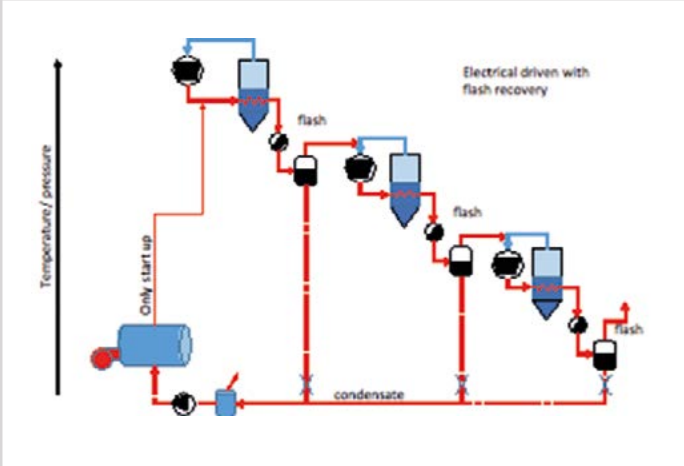

Figure 6

Besides, biomass leads to environmental problems in cultivation, transport, during storage and locally burning bio-mass causes emission of fine dust. Geothermal sources for steam production are extremely expensive if the wells are deep and seen as high risk investments. Besides, the capacity of tens of MW's does not fit in most of the industries. Some firms invest in wind-farms to allocate green electricity. Other firms buy green electricity with certificates. The purchase of green gas made of biogas has much more financial implications because of the price. Besides, the availability is restricted. Other fuels like green hydrogen and bio-liquid are not produced on an industrial scale yet. For the short term green electricity is the only serious way to go sustainable. Probably the community develops to an all-electric society!

5. Closing loops

Do we have an energy or an exergy problem? Most industrial firms produce products with ambient temperature using feedstock of ambient temperature as well. So in principle you don't have to supply heat for production. So why do we need so much fuel for production? The answer is that processes are developed with the idea high pressure steam and high temperature flames are the basis for processes. All systems are developed with a cascade of temperature levels (fig. 4). Waste heat of high temperature processes is fed to lower temperature processes. Why don't we close loops at every temperature level (fig. 5)? The only thing we have to do is to compensate exergy losses. And exergy losses can be compensated with electricity. So if an industrial firm wants to go sustainable, it has to redesign the process with the exergy approach in mind.

6. Challenge for the heat pump society

In short term all waste heat must be evaluated to reuse at a higher temperature level to minimize the losses at the end of the cascade. The developing of heat pumps and mechanical vapor compression for every temperature level is the challenge for the heat pump society. The price of these systems must be made attractive for the market by standardizing and cooperation in development.

An all-electric sustainable slaughterhouse, relying on heat pumps for its heat

Author: Fons Pennartz (KWA), photo: www.shutterstock.com

The food industry can make big steps forward by the use of residual heat from the refrigeration plant and by heat pumps. It is not only beneficial for the energy bill, but also helps to improve the production process.

Hutten slaughterhouse for beef moved in 2015 into a new factory, designed by the owners, the Hutten family. The old factory hall had become too small to accommodate the expected business growth. Hutten has chosen for a state-of-the-art and energy efficient process building, which even received the BREEAM Excellent prediction. The sustainable character of the building is largely determined by the reuse of the heat released by the ammonia refrigeration plant. The production facility has eliminated fossil fuels for its heat supply.

The heat provides a comfortable temperature in the offices, but is also used to ventilate the production halls and preheat the tap water. The new plant is now in use for two years and the installations meet the owner's expectations. "The theory looked good and convinced us, and now in practice it works fine"

Reuse of the heat of refrigeration systems seems to be an obvious solution in the food industry. But in practice people show hesitation, says Paul Ten Have, who technically assisted the Hutten project on behalf of Amersfoort KWA Bedrijfsadviseurs. "It is far from common practice, it's a difficult story to rely on a large amount of residual heat from the refrigeration plant and heat pump and leave out heat by fossil fuel. "Despite this, the potential of this heat is enormous".

Technically, there are three points where the refrigeration system produces residual heat, of varying quality. The largest part, about 85% of the residual heat, is low heat from 20 to 35 degrees C. It is generated by the condenser and has a back-up air-cooled condenser on the roof. The refrigeration installation is in fact a heat pump by itself. The remaining 15% of the heat is high grade at 60 to 75 degrees generated by the superheated discharge gas and the oil coolers of the screw compressors. The heat is exchanged by plate heat exchangers.

Machine room with refrigeration installation an d heat pumps

The low temperature heat (25 degrees) is transferred into a closed loop glycol system and is used to preheat the ventilation air, for ventilating and drying the slaughter and processing rooms at a high ventilation rate of 10. This system is activated when the outdoor temperature reaches below 18 degrees. Dry production rooms are of vital importance to the meat processing industry because wet areas can cause bacteria and fungi to grow. The conventional method of getting rid of the moisture is by turning on the cooling in order to condensate moisture on the evaporator coil. This takes a long time and costs a lot of energy. Ventilation by preheated air is more beneficial and effective.

"This ventilation process takes three hours, instead of seven hours when drying with cooling. As a result, there is less chance of bacteriological growth, "says Ten Have. That means cleaner work areas. And because the process takes less time, the rooms are also longer available for production. "This gives us the opportunity to increase production and better use the plant," says Hutten.

Another low temperature application is floor heating in the cleaning area. Heat is also used in the first stage of heating up the tap water. The following step of heating the tap water takes place with the available high temperature residual heat. This water is stored in an underground storage tank. Heat pumps, added on top of the refrigeration installation, give the temperature a further boost up to 60 degrees C. Hutten needs water of at least 82 degrees (food safety requirement), for flushing and sterilizing the knives. That temperature cannot delivered by the heat pumps and this done by a small electric boiler. Ten Have expects that in the future heat pumps can also reach 82 degrees C.

In total, all high temperature heat is reused and approximately 50% of the low-grade heat. Hutten: "Our energy bill is now lower and we have taken another step in terms of hygiene. I dare to sell the meat in Europe with the best bacteriological results and therefore the best shelf life. This is not only because of our new drying method, but it does play a role. "

Residual heat is thus both beneficial for the production process and for energy efficiency. But the technique is not applicable everywhere. For example, in freezing storage, there is little opportunity to reuse the heat, because such companies hardly have a heat demand. Ten Have: "In companies where both supply and demand of cold and heat fluctuate and are shifted over time, it is also difficult, so you need to create solutions with, for example, heat or cold buffers."

Hutten is a family business dating back to 1928. Hutten plans in terms of decades than years: "For us, this sustainable investment is beneficial. You must develop a vision of the future and willing to invest for the longer term. "

Article by Rijksdienst voor Ondernemend Nederland, revision A. Pennartz

Heat recovery making use of compression resorption heat pumps

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Abstract
This paper identifies the advantages of compression resorption heat pumps for the recovery of waste heat and its upgrading to temperatures significantly above 100 oC. Experimental work illustrates the operation of the heat rejection side of the heat pump which indicates the feasibility of operation under such conditions. The experiments are mainly executed with ammonia-water as the working fluid but additionally experiments are reported with ammonia-water-carbon dioxide as the working fluid. These last experiments indicate that the ternary mixture leads to increased performance of the heat pump. It is concluded that the relative simplicity of compression resorption heat pumps and higher performance in comparison to alternative heat pump concepts makes this heat pump type very attractive for heat recovery purposes.

1. Introduction
Lee et al. (2017) make use of the data reported by Bobelin et al. (2012) to illustrate the distribution of heat demand in the different industrial sectors in a European country. Table 1 illustrates the largest heat demands in the range 100 to 139 oC.

Table 1 makes clear that large amounts of heat are needed in this temperature range. As reported by van de Bor et al. (2015) a significant amount of low grade waste heat is available as spent water from cooling towers with tem-

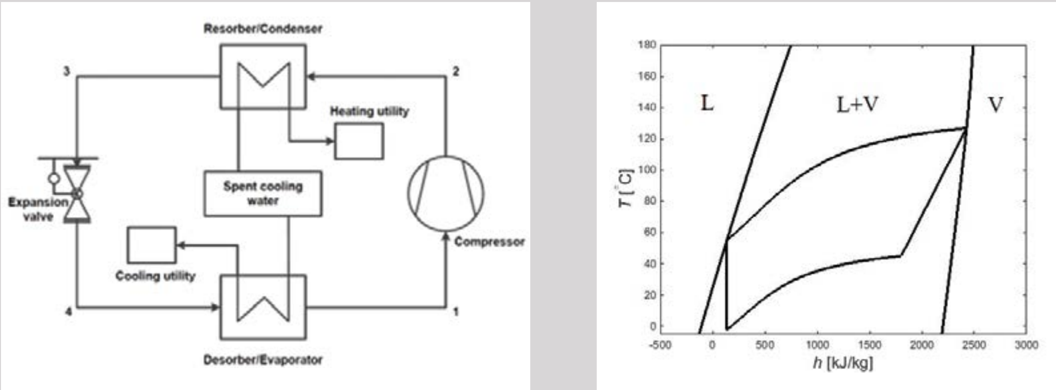

Fig. 1 – Left: schematic of the CRHP for heat recovery from spent cooling water. Right the operating conditions of the CRHP in a T-h diagram of ammonia-water with 35 wt% ammonia. Spent cooling water is, for instance, partly heated from 50 oC to 120 oC in the resorber and partly cooled down from 50 oC to 20 oC in the desorber. This operating conditions of the CRHP lead to a COP of 3.7.

peratures in the range of 45 to 60 oC. These authors indicate that about 17 PJ exergy is emitted at 60 oC as cooling water only from a specific plant in The Netherlands. Heat pumps can be used to upgrade these cooling tower flows to valuable thermal energy which can be used to operate processes as listed in Table 1. Van de Bor et al. (2015) have compared several heat pump types; dry and wet compression resorption heat pumps (CRHP), transcritical carbon dioxide heat pump (TCHP), and vapor compression heat pump (VCHP). These heat pumps can be used to upgrade cooling tower water flows to temperature levels above 100 oC, see Table 2 for an example. Assuming a compressor efficiency of 70 %, the wet CRHP is the most attractive option. Compared to the traditionally used VCHP almost 15 % improvement in performance is expected. In their study the working fluid for the CRHP is an ammonia water mixture. A recent study by Gudjonsdottir et al. (2017) indicates that for many applications even better performance can be achieved with wet CRHP by using NH_3-CO_2-H_2O as a working fluid instead of the traditionally used ammonia water.

2. Experimental performance of the heat delivery side of the CRHP

Experimental data was collected making use of a mini-channel heat exchanger which consisted of 116 tubes with internal diameter of 0.5 mm in a shell with external diameter of 25 mm. The heat exchanging length was 0.80 m. The heat exchanger was equipped with a fractal distribution system making the flows of the heat pump working fluid (ammonia-water, which circulated through the tubes from top to bottom) and water that needed to be heated (which circulated through the shell side) pure counter currently. The operating regime is similar to what could be expected in a plate heat exchanger where also pure countercurrent flow dominates. Fig. 1 illustrates the operating conditions of the CRHP considered during the experiments with left the schematic of the heat pump and right the operating conditions in a T-h diagram of ammonia-water with an ammonia concentration of 35 wt %.

2.1 Ammonia-water experiments

Fig. 2 shows how the temperature of the shell of the heat exchanger varies along the absorption process. In this case the spent water flows from bottom to top of the heat exchanger while its mass flow is maintained at 15 kg/h. The water enters the heat exchanger with 50 oC and is heated to 128, 130 and 132 oC respectively for ammo-

Sector	Temperature range 100 – 119 oC	Temperature range 120 – 139 oC
Drying	3.6 PJ	24.1 PJ
Evaporation, crystallization, concentration	11.9 PJ	8.3 PJ
Liquid & gas heating	2.9 PJ	6.1 PJ
Distillation	5.0 PJ	6.1 PJ
Thermal treatment	2.2 PJ	1.8 PJ

Table 1 – Yearly heat demand in some industrial sectors of France (Lee et al., 2017).

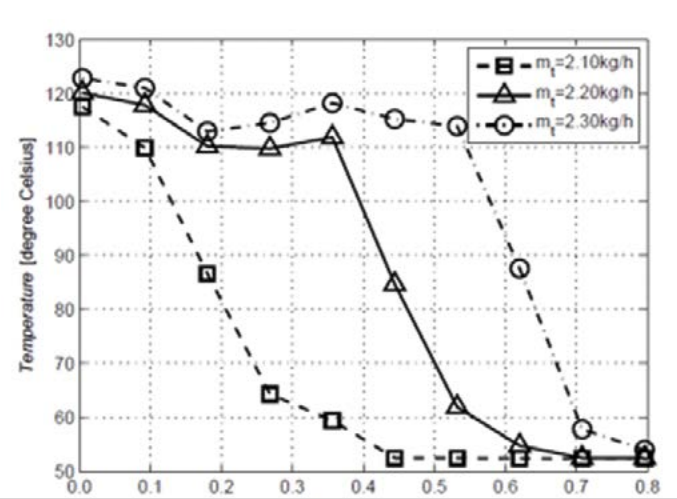

Fig. 2 – Shell outside temperature along the length of the resorber. The shell temperature is slightly lower than the water temperature. Larger ammonia-water flows displace the absorption process to positions further downstream in the heat exchanger.

nia-water flows from the top of 2.10, 2.20 and 2.30 kg/h.

Fig. 2 makes clear that, depending on the ammonia-water mass flow, a superheating zone may exist at the resorber inlet and generally also a subcooling region will develop in the outlet region of the resorber. This is similar to what Sarraf et al. (2015) have reported for the condensation process in plate heat exchangers in which the temperatures of the two fluids have a very small approach. The water outlet temperature for these experiments approaches the ammonia-water inlet temperature with less than 3 K. The ammonia-water outlet temperature approaches the water inlet temperature with 0.1 K for the lowest flows and with 3 K for the largest flow. Fig. 3 shows the temperature profiles along the heat exchanger as predicted by Shi et al. (2017) for the conditions of the experiments illustrated in Fig. 2. It makes clear that the water temperature closely follows the ammonia-water solution temperature. It is evident that the ammonia-water absorption process is ideal to raise the temperature of cooling tower water flows to elevated temperatures so that its heat can be recovered for heating purposes.

2.2 Ammonia-water-carbon dioxide experiments

Experiments have also been performed for which carbon dioxide has been added to the ammonia-water solution. Table 3 illustrates the impact of the addition of CO2 (2% by weight) for three different operating conditions.

Interesting is to see that the water side outlet temperature is significantly increased when CO2 has been added. These results indicate the performance of the system can be significantly improved when CO2 is added. Since the kinetics of the absorption of carbon dioxide are relatively slow, the addition of CO2 requires longer residence times for the absorption process to be totally absorbed. Therefore pumping instabilities were noticed during the experiments and experiments with higher mass flows than 10 kg/h failed. The geometry of the heat exchanger seems to have considerable influence. When the working fluid flowed from bottom to top instead of top to bottom the stability increased. In this case the overall heat transfer however decreased.

3. Conclusions

CRHPs are an attractive option for waste heat recovery and have the potential to have significantly improved performance compared to the traditionally used VCHP. The experimental work reported in this study illustrates how the temperature glide in the resorber of a CRHP can perfectly match the temperature glide needed for upgrading

T_{avg} = 105 oC	CRHPwet	VCHP	TCHP
60 oC → Hot	150 oC	105 oC	150 oC
60 oC → Cold	6 oC	45 oC	26 oC
COP	3.20	2.81	2.61
Max. savings (k€/yr)	274	177	127

Table 2 – Comparison of alternative heat pump systems (Van de Bor et al., 2015).

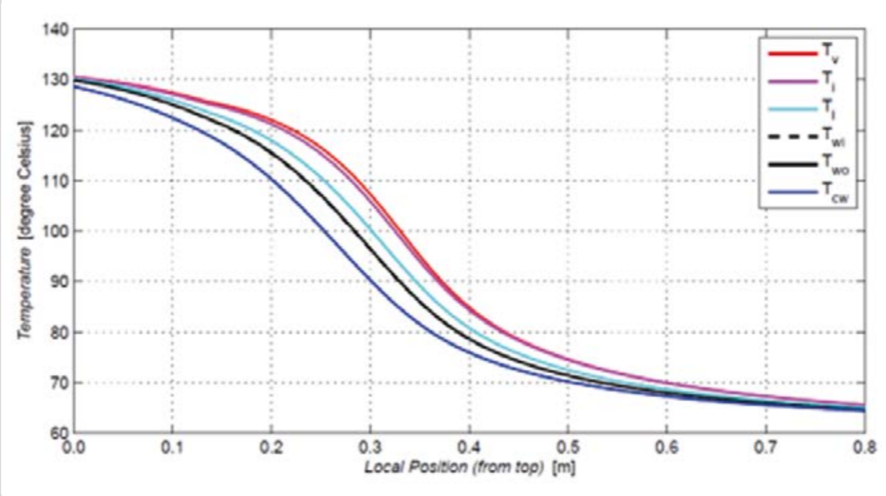

Fig. 3 – Model prediction of the internal temperatures along the length of the resorber. Tv is the vapor temperature, Ti is the vapor-liquid interface temperature and Tl is the liquid (film) temperature, all in the ammonia-water side of the heat exchanger. Tcw is the water temperature. Twi and Two are the internal and external wall temperatures.

waste streams, like spent cooling tower water, to more useful temperature levels. The experiments additionally indicate that the potential of CRHP can be even greater with NH3-CO2-H2O as a working fluid.

Acknowledgements

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Parameter	Unit	without CO2/with CO2	without CO2/with CO2	without CO2/with CO2
M_s	kg/h	5.0/5.0	7.5/7.5	10.0/10.0
$T_{t,in}$	oC	132.0/132.5	131.0/131.4	135.4/134.3
$T_{s,out}$	oC	127.1/131.3	125.9/130.1	127.2/133.4
$T_{t,out}$	oC	50.7/58.5	49.9/58.4	50.2/54.8
$T_{s,in}$	oC	50.6/50.8	49.7/50.1	50.1/50.2
LMTD1	K	1.1/3.5	1.5/3.8	1.9/2.2

1LMTD is short for logarithmic mean temperature difference

Table 3 – Comparison of the resorber performance without and with CO2 added to the ammonia-water solution. is the mass flow of the binary / ternary mixture. Tt stands for the temperatures in the resorber side. Ts stands for the water side temperatures.

Development of a thermoacoustic heat pump for distillation column

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Abstract
A thermoacoustic heat pump is a promising innovative heat pump technology which can be applied in the industry to upgrade industrial waste heat. The thermoacoustic heat pump consists of a regenerator flanked by two heat exchangers and placed in a gas (helium) filled acoustic resonator. An acoustic wave is generated and sustained by an acoustic driver in the resonator. The acoustic wave forces the working gas in the regenerator to undergo a Stirling cycle and to pump heat from the low temperature heat exchanger to the high temperature heat exchanger. The advantage of the thermoacoustic heat pump is that it can operate over a large range of (high) temperatures and can achieve large temperature lifts. One of the industrial applications where the application of the thermoacoustic heat pump can be beneficial is the distillation process. Distillation is one of the largest energy consumers processes in refining and bulk chemical industries. In a thermoacoustic heat pump assisted distillation column, latent heat from the condenser in pumped to the reboiler. An energy saving potential of about 10-20 PJ/year is estimated for the Netherlands, 100-200 PJ/year for Europe, and 300-600 PJ/year for the world. This paper presents the design, construction, and test of a bench scale electrically driven thermoacoustic heat pump. A reciprocating piston compressor is tested as acoustic driver for the heat pump.

1. Introduction
The application of heat pumps in the industry can lead to large energy savings and reduce global warming emissions. Heat pumps can be used to upgrade waste heat. This enables to reuse the huge quantities of energy that

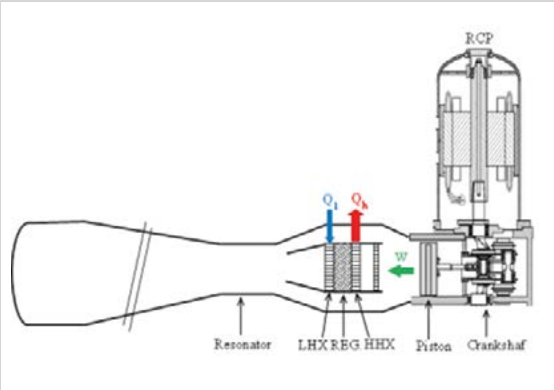

Figure 2 Thermoacoustic heat pump consist of a regenerator (REG), a hot heat exchanger (HHX), and a low temperature heat exchanger (LHX), placed in a resonator. A reciprocating compressor consisting of a piston driven by rotary electrical motor via a crankshaft is used as driver.

Figure 4 Picture of the reciprocating piston compressor

would otherwise be rejected to the environment. One of the industrial applications where the application of a heat pump can be beneficial is the distillation process [1-2]. Distillation is one of the largest energy consumers processes in refining and bulk chemical industries. It is estimated that distillation columns consume about 40% of the total energy used to operate plants in these sectors [3-4]. The distillation process is a very inefficient process as heat at high temperature is supplied to boil a mixture of liquids and most of this heat is released at a lower temperature level during condensation. In a conventional distillation column energy is supplied to the system via the reboiler to evaporate the feed for the separation process. The vapors from the top of the column are liquefied in the (water) cooled condenser. About 95 % of the energy needed for the reboiler leaves the system as waste heat. In a heat pump assisted distillation column, the condenser is linked to the reboiler via the heat pump where the temperature of the vapor from the top of the column is increased and fed to the reboiler where it is condensed. For the Netherlands, the total energy saving potential is estimated to 10-20 PJ/year. The extrapolation to Europe and the world, based on production capacities, leads to more than 100-200 PJ/year for Europe and 300-600 PJ/year for the world.

However, widespread use of large scale heat pumps is not yet common due to the low operation temperatures and limited temperature lifts of conventional heat pumps. Innovative heat pump technologies are needed which can help to overcome these difficulties. One promising innovative heat pump technology is the thermoacoustic heat pump which uses acoustic power to increase the temperature of a waste-heat stream to a higher, useful temperature. Thermoacoustic heat pumps can be electrically or thermally driven and can operate over a large scale of (high) temperatures and can achieve large temperature lifts.

An electrically driven thermoacoustic heat pump is driven by a reciprocating piston which compresses and expands the working gas. The piston can be powered by an electrical linear motor, an electrical rotary motor, an internal combustion engine, or a turbine. Small scale electrically driven thermoacoustic heat pumps usually use a linear electrical motor to drive the piston. The linear motor has the advantage that the piston is directly attached to the moving part of the linear motor. However, linear motors are not available at high power. The largest commercially available linear motor has a power of 10 kW. A reciprocating piston compressor (RPC) consisting of a rotary electrical motor which drives a piston using a crankshaft can be an alternative to drive an industrial thermoacoustic heat pump. RPC's are commercially available over a large power spectrum up to tens of megawatts. They are usually used in the industry to increase the pressure of gases but it is not known if they can be deployed to drive thermoacoustic heat pumps.
The objective of the study presented in this paper is to evaluate whether an RPC can be used as an acoustic driver for an electrically driven thermoacoustic heat pump (EDTAH). An existing bench scale EDTAH [5] which is driven by a linear motor is adapted to be driven by RPC. The specifications for the RPC are dictated by the requirements of the EDTAH.
The remaining of this paper is organized as follows: section 2 discusses the working principle of the thermoacous-

Figure 5 Picture of the compressor placed in a vessel pressure and to be coupled to the heat pump via the flange on the right of the picture.

Figure 6 Picture of the regenerator

tic heat pump. Section 3 presents the thermoacoustic heat pump assisted distillation column. In section 4, the design and construction of the heat pump is discussed. Section 4 presents the experimental results. In section 5 conclusions are drawn.

2. Thermoacoustic heat pump

The electrically driven thermoacoustic heat pump uses acoustic power W to pump heat Ql from a lower-temperature heat source and to deliver heat Qh to a high-temperature heat sink. Figure 1 shows a thermodynamic illustration of a thermoacoustic heat pump operating between a low temperature source at Tl and a high temperature sink at Th. The acoustic power W necessary to the operation of the heat pump is delivered by a piston compressor which converts electrical power into acoustic power.

The working principle of a thermoacoustic heat pump is based on the Stirling cycle. However, in contrast to a conventional Stirling heat pump where a power piston and a displacer are used to force the working gas to execute the Stirling cycle, in a thermoacoustic heat pump a sound wave takes over the task of these mechanical parts [6- 8]. The acoustic wave takes care of the compression, displacement, expansion of the working gas and for the timing necessary for the Stirling cycle. Similar to a conventional Stirling heat pump, the core of a thermoacoustic heat pump consists of a regenerator placed between two heat exchangers. The core is placed in a gas filled acoustic resonator (tube). The acoustic wave is generated by an oscillating piston driven by an electrical motor as shown in Figure 2. A compact acoustic network creates the local traveling-wave phasing necessary for the Stirling cycle. The circuit consists of the resistance of the regenerator, a compliance, and a feedback inertance which are arranged in a loop configuration. The compliance consists of a volume of gas and is indicated by "C" in 2 and the feedback inertance consists of a tube and is indicated by "L" in 2. Extended explanation of the working principle of thermoacoustic systems can be found in [6-10] and references therein.

In a thermoacoustic heat pump assisted distillation column, the low temperature heat exchanger of the thermoacoustic heat pump is coupled to the condenser of the column and the hot heat exchanger to the reboiler. In this way the heat released during condensation of vapors at the top of a column is upgraded by the thermoacoustic heat pump and delivered at high temperature to the reboiler of the column. A schematic illustration of a thermoacoustic assisted distillation column is shown in 3.

Figure 3 Schematic illustration of the application of a thermoacoustic heat pump to a distillation column.

4. Design and construction of the thermoacoustic heat pump

A schematic illustration of an EDTAH is shown in 2. The heat pump is designed to operate between the condenser temperature and the boiler temperature of an industrial distillation column. The working medium is helium gas at an average pressure of 50 bar and the operation frequency is 80 Hz. A piston compressor delivers the acoustic power needed by the heat pump. The specifications of the thermoacoustic heat pump are summarized in1.

The heat pump is designed and optimized using the thermoacoustic computer code DeltaEC [9]. A short descripti-

Figure 7 Picture of the heat exchanger

Figure 8 Picture of the thermoacoustic heat pump

on of the different components of the system will be given in the following.

Piston compressor

The objective is to evaluate whether a piston compressor can be used to drive the EDTHP. The compressor has to deliver the required acoustic power by the heat pump of about 3.5 kW. Because low power piston compressors are not commercially available, an outboard engine is adapted to function as a piston compressor. The engine is made oil free to avoid contamination of the heat exchangers and the regenerator. The engine is coupled to an electrical motor. The drive shaft of the electrical motor is attached to the crank shaft of the engine using a Bowex M28 consisting of two hubs and one M-sleeve. The electrical motor drives the piston of the engine via the crankshaft of the engine. The electrical motor and the engine are aligned and fixed in metallic frame. A picture of the compressor is shown in 4.

The specifications of the reciprocating compressor are summarized in 2 .

The drive ratio is defined as the ratio of the dynamic pressure amplitude at the piston and the mean pressure of the gas. The compressor will be placed in a pressure vessel to operate at 50 bar as shown in 5. The pressure vessel of the compressor has a flange which is used to couple it with the heat pump as shown on the right of 5.

Regenerator

The regenerator consists of a 30 mm thick stack of 140-mesh stainless-steel screen punched at a diameter of 26 cm. The diameter of the screen wire is 56 m. The stack is placed in a thin-wall tube. The regenerator is designed so that the hydraulic radius is small compared to the thermal penetration depth which is necessary for a good thermal contact of the gas with the regenerator matrix. The hydraulic radius of the screen is 44 µm and the volume porosity is about 76 %. A picture of the regenerator is shown in 6.

Heat exchangers

The heat exchangers consist of a cylindrical steel block where passes are machined. Copper fins with a density of 86 fins/in are brazed on the helium gas side to increase the heat transfer area. On the thermal oil side fins with a density of 50 fins/in are used. The diameter of the heat exchangers is 26 cm and the length is 3 cm for LHX and HHX. The volume porosity of the heat exchangers at the

Working gas	Helium
Average pressure (bar)	50
Frequency (Hz)	80
High operation temperature (°C)	100
Low operation temperature (°C)	60
Thermal power at 100 °C (kW)	10

Table 1 Specifications of the thermoacoustic heat pump.

Operation frequency (Hz)	80
Piston diameter (mm)	50
Swept volume (liter)	0.14
Acoustic power (kW)	4.5
Drive ratio (%)	5
Average pressure (bar)	50

Table 2: Specifications for the piston compressor

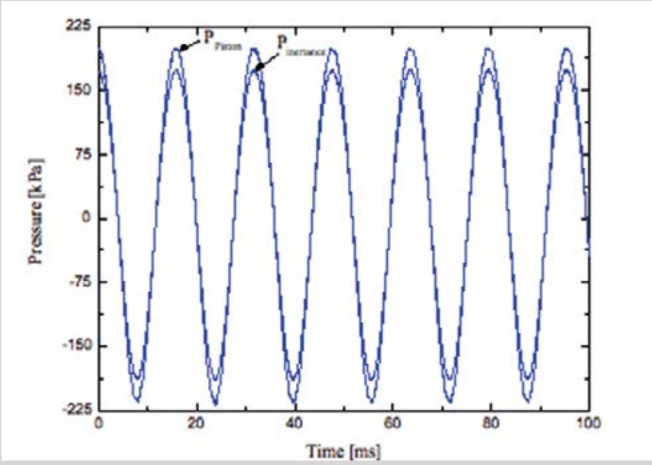

Figure 9 Acoustic pressure measured at the piston and in the inertance as function of time.

helium side is 20 %. A picture of the heat exchanger is shown in 7.

Acoustic circuit

The feedback inertance, compliance, and regenerator acoustic resistance are designed to get the traveling-wave phasing in the regenerator and minimal viscous losses. The acoustic pressure and acoustic velocity have to be in phase at the regenerator midpoint. The inertance (L) consists of a tube with a diameter of 23 cm and a length of 40 cm. The compliance (C) has a volume of 10 liter.

Resonator

The resonator consists of two straight tubes connected by a cone. The first straight tube has an inner diameter of 22.3 cm and length of 82 cm, the conical tube has a start inner diameter of 22.3 cm, a length of 462 cm, and a final inner diameter of 48.4 cm. The last tube has an inner diameter of 48.4 cm and a length of 55 cm.
8 shows a picture of the thermoacoustic heat pump.

5. Experimental results

A thermal bench using water and thermal oil simulates the low temperature heat source (20-80 ^{0}C) and the high temperature heat sink (100-200 ^{0}C). The high temperature heat exchanger (HHX) and the low temperature heat exchanger (LHX) are connected to the thermal bench using flexible tubing.

Various sensors are placed through the system to measure the operating parameters of the heat pump. The oil and water flow into the heat exchanger is measured with flow meters. The temperatures of the water and oil at the inlet and the outlet of the heat exchangers are measured with thermocouples. Several pressure sensors are placed throughout the system to measure the acoustic pressure at different locations in the system. The signals from the thermocouples are read by a data logger and sent to a computer. The pressure signals (magnitude and phase) are first measured by lock-in amplifiers then read by the data logger and send to a computer. The signals are recorded and displayed using Labview.

Acoustic measurements

The acoustic pressure generated by the compressor is measured as function of time using a Pico oscilloscope and the measurements are shown in Figure 9. The signals are measured at the piston location and in the inertance. The signal has a sinus shape which is required for the driving of the heat pump. The measurements show thus that the piston compressor can be used to generate and maintain an acoustic wave to drive the thermoacoustic heat pump. When the driving frequency of the compressor approaches the acoustic resonance of the heat pump, the electric power consumed by the electric motor of the compressor increases rapidly. This leads to an overheating of the motor which does not have a cooling system. The consumed electric power is proportional to the frequency. To avoid overheating of the compressor the operation frequency has to be decreased. The frequency can be decreased by using a gas mixture of helium and argon. A gas mixture consisting of 6 % argon and 94 % helium resulted in a decrease of the resonance frequency from 78 Hz to 63 Hz. Although the heat pump is designed for pure helium, the simulations show that the heat pump still perform well with 6 % argon in helium.

Performance measurements are conducted using a mixture of helium (94 %) and argon (6 %) and for a drive ratio (Dr) of 3.49 %. The drive ratio is defined as the ratio of the dynamic pressure amplitude at the piston and the mean pressure of the gas. The operation frequency is about 63 Hz. The measurements are summarized in Table 3. Tl and Th are the low- and high-temperature of the heat pump, respectively. Ql is the heat pumped at the low temperature and Qh is the heat delivered by the heat pump at the high temperature. W is the acoustic power used by the heat pump and COP is the coefficient of performance of the heat pump (Qh/W).

The acoustic power (W) used by the heat pump is deduced from the energy balance. The COP given in the Table 3 is internal and thus does not include the acoustic losses in the compressor and in the resonator. The measurements show that an internal COP of about 4.8 is achieved by the heat pump with a thermal power of about 3 kW at 109 ºC and a drive ratio of 3.49 %. It is expected that at the design drive ratio of 5 % the required 10 kW thermal power can be delivered by the heat pump. This is because the thermal power is a quadratic function of the drive ratio [10]. The drive ratio of 5 % can be achieved by the compressor if electric motor cooling is implemented.
In future measurements, the acoustic input power to the heat pump will be measured so that the external COP can be determined. The electrical motor will be provided with water cooling so that measurements can be done at resonance with pure helium and at drive ratio of 5 %.

6. Conclusions

It can be concluded that reciprocating piston compressor can be used to drive a thermoacoustic heat pump, if it is designed to operate oil-free and to operate at 50 bar.

The RCP driven bench scale thermoacoustic heat pump delivers about 3 kW of thermal power at 109 ºC with an internal COP of 4.8. It is expected that at the design drive ratio of 5 % the required 10 kW thermal power can be delivered by the heat pump. This is because the thermal power is a quadratic function of the drive ratio [9]. The drive ratio of 5 % can be achieved by the compressor if electric motor cooling is implemented.

In the near future, the piston compressor will be provided with water cooling so that the measurements can be done with pure helium and at a drive ratio of 5 %.

Dr (%)	3.49
Tl (ºC)	40.80
Th(ºC)	109.02
TTBT(ºC)	22.63
Qh (kW)	2880
Ql (kW)	2.519
W(kW)	0.498
COP	4.78

Table 3 Performance measurements for the thermoacoustic heat pump

Acknowledgements

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Test results R600

pilot heat pump

Author: Wemmers A.K., Haasteren A.W.M.B. van, Kremers P. Kamp J van der1, photo: www.shutterstock.

Abstract

A pilot scale compression heat pump using butane as refrigerant with a power output of 160 kW was designed, build and integrated with the paper production process on site at the SmurfitKappa Roermond mill. The project was executed to determine if it is possible to build an industrial heat pump able to produce low pressure steam from existing components and to verify the performance.

A 160 kW pilot scale heat pump able to deliver low pressure steam (1 barg)from 60°C waste heat was demonstrated at the SmurfitKappa Roermond mill. The successful demonstration confirmed that is possible to build such a heat pump based on commercially available components.

Higher steam pressures, up to 2.4 barg, were demonstrated. It is concluded that the heat pump can operate under the maximum designed steam pressures with COP's according to the design calculations.

1. Introduction

Steam is the most common energy carrier in industry for temperatures up to 200 °C. In Europe an estimated total of 2100 PJ steam per annum is used in refineries, chemical industry, food processing and paper industry. To give an idea of the value: 2100 PJ of steam costs approximately 18 billion euros to generate, based on the value of natural gas (28 €/MWh). At the same time recoverable waste heat is available in abundance at low temperatures. The use of heat pumps to recover and upgrade that low temperature waste heat to process steam would greatly enhance the energy efficiency of industrial processes.

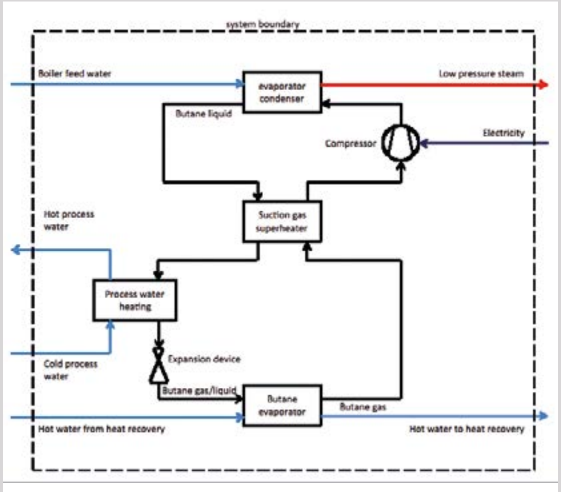

Figure 1: Evaluated heat pump system.

		Calculated performance	
measurement date	steampressure [barg]	COPsteam [-]	COPsteam + hot water[-]
mrt '15	0,5	3,7	5,1
okt '15	0,8	3,3	4,9
okt '15	1,4	2,7	4,5
okt '15	2,0	2,2	4,3
okt '15	2,4	2,0	4,2

Tevap 54°C .. 55°C

Table 1: Calculated performance of the pilot heat pump.

The reason why heat pumps are not widely applied are both technical and economical. Due to the fact that in the past electricity was far more expensive compared to natural gas, the steam boiler and cogeneration have long been the work horses for steam generation. Since there was no market demand for electrically driven industrial heat pumps no technology was developed. However, since the turn of the century the prices for electricity have been going steadily down while the price of natural gas remained more or less the same. As a result the application of electrically driven heat pumps starts to make sense from an economic point of view, sparking an interest from process owners. Commercially available heat pumps are only able to generate process heat up to approximately 90 ˚C, thus seriously limiting the application of heat pumps in industry.

For these reasons, a new heat pump was designed based on a refrigerant (Butane, R-600) able to deliver steam of at least 2 bara (120 ˚C) from waste heat of 60 ˚C. The R-600 heat pump is developed from components (compressor, heat exchangers, controls, etc.) that are available in the market to ensure quick market introduction. This report contains the measurement results of the performance tests on an experimental R-600 heat pump. The heat pump was tested on site at the Smurfit Kappa Roermond mill. Apart from size, the experimental heat pump is identical to a full scale system (over 1.5 MW process heat) which makes the results from the tests directly applicable to full scale heat pumps. The experiments were executed to answer the following questions:

- Is it possible to build an industrial heat pump from components available in the market which is capable of producing low pressure steam (~2 bara) from low temperature (~60˚C) waste heat;
- What are the technical and safety implications of integrating the heat pump with the paper production process;
- Can the heat pump operate under a variety of conditions;
- Is the measured performance consistent with the calculated performance;

The measure by which the performance of heat pumps is determined is the Coefficient of Performance (COP). The COP is the ratio between delivered process heat over supplied electricity. The investigated heat pump delivers process heat both as steam and as hot water, therefor two COP's are established: COP steam and COPsteam+hot water.

2 Industrial heat pump system

An industrial heat pump system contains the heat pump itself and the auxiliary systems necessary to integrate the heat pump with the industrial process.

The heat pump

Figure 1 below depicts the process flow diagram of the heat pump. The cycle starts in the evaporator, where butane is evaporated at low pressure with the aid of low-temperature waste heat. The low pressure butane vapor is superheated in the suction gas super-heater to avoid liquid butane during compression. The low pressure, superheated butane vapor is then compressed to produce high pressure butane vapor. The high pressure, high temperature butane vapor is condensed in the evaporator/condenser to generate low pressure steam. The sen-

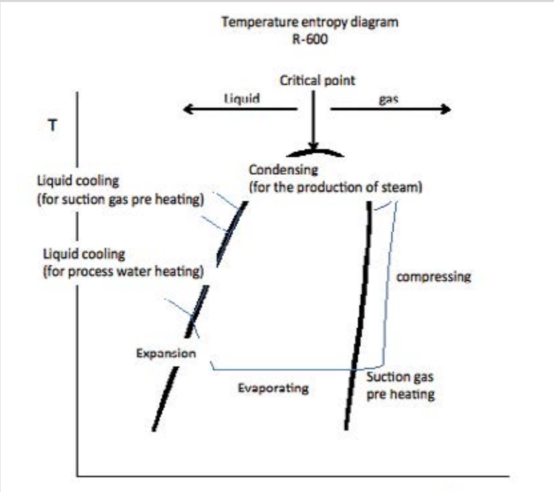

Figure 2: Temperature Entropy diagram for butane.

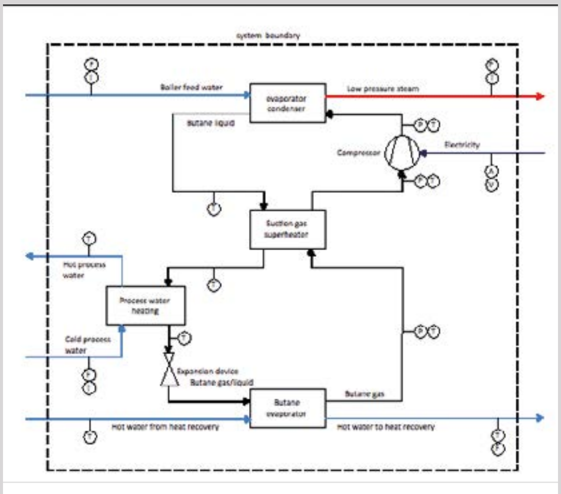

Figure 3: Evaluated heat pump system and measuring points.

sible heat in the condensate in the butane liquid leaving the evaporator/condenser is then cooled in order to heat process water. In this way, more heat is extracted from the heat pump without the need to supply more work to the compressor. After all, for the compressor does not matter whether the low pressure vapor is generated by flashing in the expansion device or by evaporation in the butane evaporator. The final step in the cycle is the pressure drop of butane in the expansion device.

Auxilliary systems

- The auxiliary systems contain at least the following components:
- The heat recovery system which recovers low temperature from the process;
- The connections with the steam, condensate and the process water systems of the process;
- The electrical connection;
- Integration with the process control system, including the control software.

The R600 pilot heat pump in the CATCH-IT project is used to produce low pressure steam from boiler feed water and hot process water from fresh but already preheated water. The source heat is recovered from moist exhaust air from the paper machine drying section.

2.1 Design calculations

To calculate a COP input parameters are needed. These input parameters are categorized in three groups: Environemtal and safety requirements, end-user requirements and engineering choices.

Environemtal and safety requirements

To acquire a licence to operate the heat pump it has to meet the requirements regarding environmental and safety requirements set by the government. This applies in particular to the requirements with respect to pressure devices (PED) and combustible substances (ATEX).

End-user requirements

- The end-user requirements are of a set of external conditions under which the heat pump must operate:
- Delivered steam pressure and capacity;
- Delivered hot process water temperature and capacity;
- The temperature(s), characteristics (sensible heat, (partly) latent heat) and availability of the source heat.

The end-user requirements for the R-600 pilot heat pump: Steam pressures to be delivered at 0.5 barg up to 2.4 barg. Design capacity is approximately 150 kW and is determined by the size of the smallest available compressor. The exact capacity is depending on the operating conditions;

- Hot process water at 70˚C (maximum possible 100˚C), the capacity is the maximum possible capacity given the heat pump configuration;
- The source heat is waste heat recovered from the exhaust of the paper machine hood. The source heat is transported in a water/glycol system, the Twater/glycol = 5 K, the water/glycol approach temperature varies from 60 ˚C up to 75 ˚C. The source heat is always available when there is a demand for process heat.

		Calculated performance		Measured performance	
measurement date	steampressure [barg]	COPsteam [-]	COPsteam + hot water[-]	COPsteam [-]	COPsteam + hot water[-]
mrt '15	0,5	3,7	5,1	3,6	5,1
okt '15	0,8	3,3	4,9	3,0	4,7
okt '15	1,4	2,7	4,5	2,6	4,6
okt '15	2,0	2,2	4,3	2,2	4,6
okt '15	2,4	2,0	4,2	1,9	4,4

Tevap 54°C .. 55°C

Table 2: Comparison between calculated and measured performance.

Engineering choices

The engineering choices concern the set of internal parameters which influence the COP, the capacities and the CAPEX of the heat pump:

- Temperature differences (T's) over heat exchangers;
- Compressor type with resulting efficiencies;
- Electric motor and controls;
- Suction gas superheating;
- Condensate sub-cooling.

Temperature difference

For ease of calculation the average temperature difference instead of the more accurate logarithmic temperature difference over the heat exchangers is used:
where: is the maximum temperature difference between the streams and is the minimum temperature difference between the streams. In the engineering calculations an average temperature difference of 5 K has been used.

Compressor efficiency

The compressor efficiency depends on the type and design of the compressor. Three categories of efficiency losses are to be considered:

- Fluid flow and mechanical friction resulting in a higher compressed gas temperature compared to isentropic compression, e.g. pressure losses over valves, leakage past the piston or lobes, etc.;
- Fluid flow and mechanical friction dissipated via the lubricating oil, cooling of the compressor casing, etcetera;
- Non adiabatic compression caused by excessive oil cooling, excessive cooling of the compressor casing, cooling of the compressed gas either via unintended heat exchange between suction gas and compressed gas or via active cooling of the compressed gas.

In the engineering calculations adiabatic compression is assumed. Furthermore the heat loss via oil is assumed to be small (<5% of the compressor work). Based on these assumptions the performance of the compressor can be calculated using an isentropic efficiency of 0.85.

Electric motor and controls

The efficiency of the motor and the controls is taken at 0.95.

Suction gas superheating

The amount of suction gas superheating influences the COP's and the capacity of the heat pump. An increase is suction gas super heating results in an increase of the COPsteam, a decrease of the COPsteam + hot water and a decrease of the capacity. The suction gas superheating varies depending on the operating conditions. The set point of the suction gas superheating is given by the desired 5 K superheating of the compressed gas temperature.

Condensate sub-cooling

The amount of condensate sub-cooling influences the amount of heat transferred to the hot process water flow. The temperature before the expansion device is taken at 75 °C.

2.2 Results of the design calculations

The performance of the pilot heat pump for five different steam pressures is given in table 2.1 below.
Given the Carnot efficiency temperature of the heat delivered divided by temperature lift) it is expected that the COP reduces when the temperature lift increases. Visible from table 1 is that COPsteam decreases faster than the COPsteam + hot water. This is best explained using the Temperature Entropy diagram shown in Fig. 2.
Increasing the condensation temperature means that the heat of evaporation becomes less. Finally, when the condensation temperature is at the temperature of the Critical point (150.8 °C, for butane) there is no heat of evaporation left, hence no steam is produced. The sensible heat of the butane liquid leaving the condenser increases with increasing condensation temperature. As a result the amount of heat available for heating process water increases hence the COP for steam and hot water remains higher.

3 Measurement program

Although a complete heat pump system is necessary to operate the heat pump, the performance evaluation focus-ses on the heat pump itself. The system boundary for the performance evaluation of the heat pump and the measuring points used to collect the necessary data are depicted in the schematic of Figure 3. To be able to identify the equipment used for data collection the TAG numbers are also given in Figure 3.1. The heat pump forms an integral part with the Paper Mill's process control system (PI system from OSIsoft) which means that all the measuring points are continuously logged on an one minute interval and stored for future reference.

Measuring method

For the heat pump depicted in Figure 3 an external and an internal energy balance are made for each operation condition. From these energy balances the COP's are calculated and compared to the theoretical COP's. The theoretical COP's are calculated from the design conditions and assumptions. By comparison of the measured and calculated COP's, conclusions are drawn on:

- The performance of the heat pump;
- The reliability of the design calculations.

External energy balance

The external energy balance consists of the heat and electricity which flow over the system boundary, depicted as the dashed line in Figure 3 The heat flows are determined by:

The mass flow () of stream i and its temperatures are measured for the boiler feed water, the hot process water and the heat recovery water. To establish the enthalpy of the produced steam, the steam pressure and temperature are also measured. The specific enthalpy (h) is determined using the NIST Standard Reference Database 23, version 9.1 (Lemmon et al., 2013) which enables the calculation of specific enthalpies from measured temperatures and, in case of vapor, from temperatures and pressures.

The electricity used by the compressor is measured at the frequency controller of the compressor motor.

Internal energy balance

The internal energy balance consists of the amounts of heat and work transported by the butane inside the reverse Rankine cycle. The internal energy balance is calculated per kilogram of butane since the butane mass flow is an unknown. The temperatures and pressures of the butane are measured at the points given in Figure 3. The specific enthalpy (h) of the butane at each point is determined using the NIST Standard Reference Database 23, version 9.1 (Lemmon et al., 2013). To be able to compare the COP's from the internal and external energy balance the amount of compressor work is divided by the efficiency of the electric motor, the efficiency is set at 95%.

3.3 Comparison calculations and measurements

In table 1below the measured performance date is compared to the calculated performance data.

Comparison of the measured COP's and the calculated COP's show (small) differences. Most of the differences can be explained by differences between the conditions during testing and the conditions for the calculations. Most noticeable:

- The temperature differences over the heat exchangers are smaller in practice than the assumed 5 K in the calculations;
- The isentropic efficiency of the compressor is lower (approximately 0.8 depending on specific operating condition) than the assumed 0.85;
- In most cases the COPsteam internal is somewhat lower than the theoretical COPsteam at the same time the COPsteam + hot water internal is somewhat higher than the theoretical COPsteam + hot water. This suggests that heat is transported from the compressor to the process water heater via the oil and cylinder head cooling circuit.

4 Conclusions

A 160 kW pilot scale heat pump able to deliver low pressure steam from 60°C waste heat was demonstrated at the SmurfitKappa Roermond mill. The successful demonstration confirmed that is possible to build such a heat pump based on commercially available components.

The heat pump was tested at different operating conditions. The critical operating conditions from a technical point of view are higher steam pressures. Higher steam pressures are associated with higher pressures and temperatures which impose additional thermal and mechanical strains to the heat pump. Higher steam pressures, up to 2.4 barg, were demonstrated. Operating conditions which impose no additional strain to the heat pump and which were solely intended to verify expected variations in capacity and COP's were not demonstrated. It is concluded that the heat pump can operate under the maximum designed steam pressures.

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Pinch based scan

method for heat pumps and vapor recompression

Author: Ir. Jan Martin Grift1, photo: www.shutterstock.com

Abstract
The possibilities for the use of heat pumps and vapor recompression can quite easily be determined by pinch technology. In a dozen companies in the Netherlands scans were carried out. From these scans, multiple possibilities have been found that are already viable. The potential of a hybrid energy supply for industrial partners in the long term is even greater. A heat pump scan using supporting software can identify the opportunities in just one day.

1. Introduction
Waste heat is getting more attention by industrial firms because of their ambition to go sustainable. Studies show that approximately 30% of the industrial primary energy

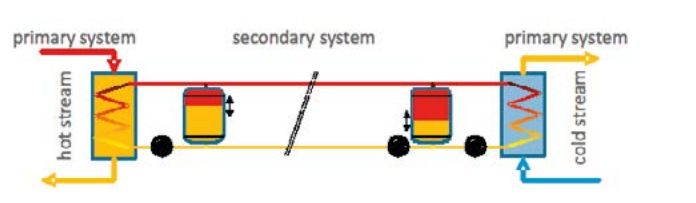


Figure 1. General approach with a heat pump integrated.

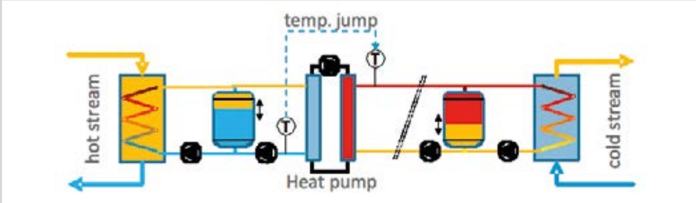


Figure 3. Pinch diagrams specific case with the calculated

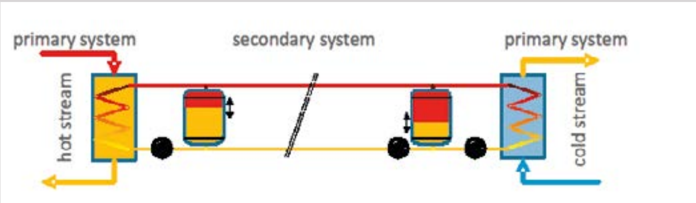


Figure 2. General approach with vapor compression integrated.

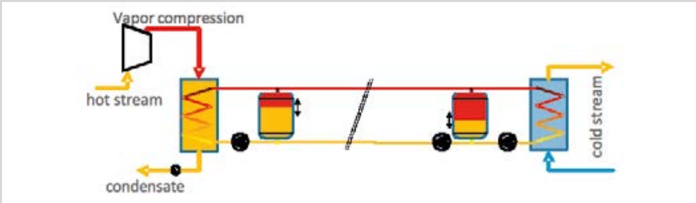


Figure 5

consumption is at a level below 200 oC [1]. This is in the temperature range where heat pumps and vapor recompression are attractive solutions (60-200 oC). Direct heat exchange between waste heat streams (hot streams) and streams to be heated (cold streams) can be determined by pinch methodology [2] as developed (further) by Bodo Linnhoff. The so called grand composite curve gives insight in the theoretical minimum heat and cold demand, but also in the utilities needed. So the potential of heat pumps and cogeneration can be determined from one graph, i.e. the grand composite curve (Fout: Bron van verwijzing niet gevonden). The grand composite curve is essential for heat pump evaluation because ad hoc measures can block future developments towards a sustainable situation.

After the potential for heat pumps and cogeneration is determined, the challenge is to define feasible solutions, mostly in an existing situation. The feasibility depends on aspects such as:

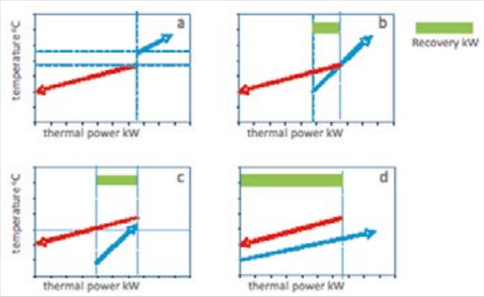
- Temperature levels
- Mean Power available and needed
- Fluctuation in power
- Load hours
- Physical condition of streams (Gas, liquid, evaporating, condensing)
- Physical properties (specific heat, enthalpy)
- Contaminations
- Distance between points (coordinates)
- Cross overs needed (piping bridge, underground, indoor)
- (Marginal) energy tariffs

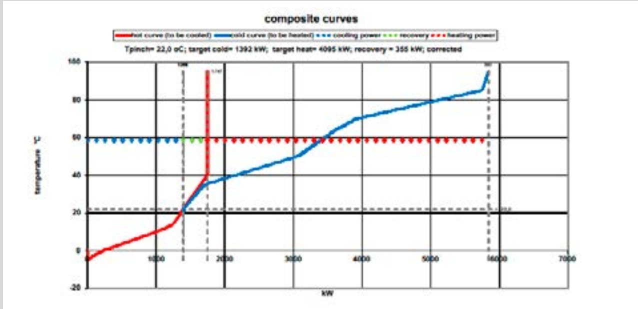
In order to determine the feasibility of connecting waste heat streams with cold streams, (directly or with heat pumps or vapor compression), Energy Matters developed a simple method which is explained in the next section.

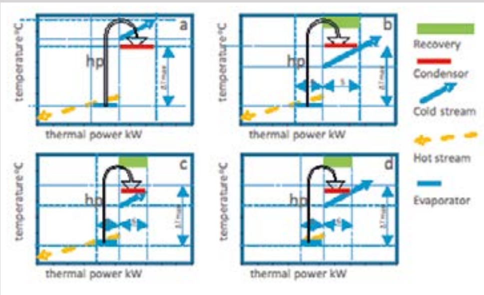
2. Approach

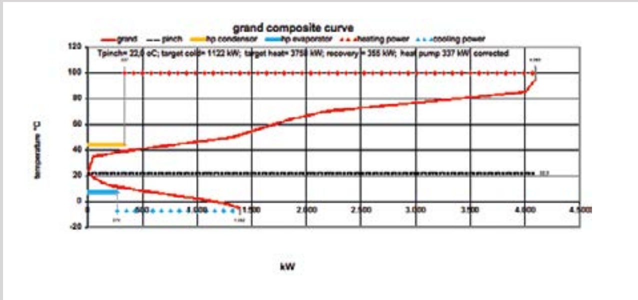
The pinch method starts with collecting data (temperatures and flow rates) of the hot and cold streams available. In most situations there is some distance between these streams. A common way for transporting heat from one process to another is by using pipes in which a liquid is circulated (mostly water). For every point the temperatures of the process streams (primary streams) are directly converted in the liquid temperatures of the transport system (secondary streams). The advantage is that the pinch can be calculated more accurate because in the pinch method, you always have to choose the minimum temperature difference in the heat exchangers, although the dominating streams are not known yet. So in this approach the temperature difference at the pinch equals zero provided that the transport liquid temperatures are brought in instead of the process temperatures. The temperature difference between process and transport liquid depends on the phase (change) of the process stream. If the process stream is directly utilizable in the transport system the difference is zero.

Another challenge is to deal with fluctuations in both the heat source and the heat demand. In order to reduce these fluctuations, hot water buffers are placed at both sides of the system. To prevent exergy losses from mixing liquids at different temperatures, these vessels are stratified. The liquid is brought in very slowly by using screens on top and at the bottom of the vessels (Fout: Bron van









verwijzing niet gevonden). If a hot or cold stream has a constant flow, vessels and or pumps can be eliminated. If the hot or cold stream can be transported easily one heat exchanger can be eliminated.

3. Integrating heat pumps

If the hot stream is not able to deliver heat to the cold stream directly, because the temperature level is too low, the temperature can be increased by using a heat pump (Figure 1). Another technique is using recompression which is possible when the hot stream is in the gaseous phase (Figure 2). From an economical perspective the maximum increase in temperature of both systems is about 50 K (Coefficient of Performance (COP) > 4)1. Also in these cases the buffers and heat exchangers are only placed if necessary.

4. Pinch diagrams single connections

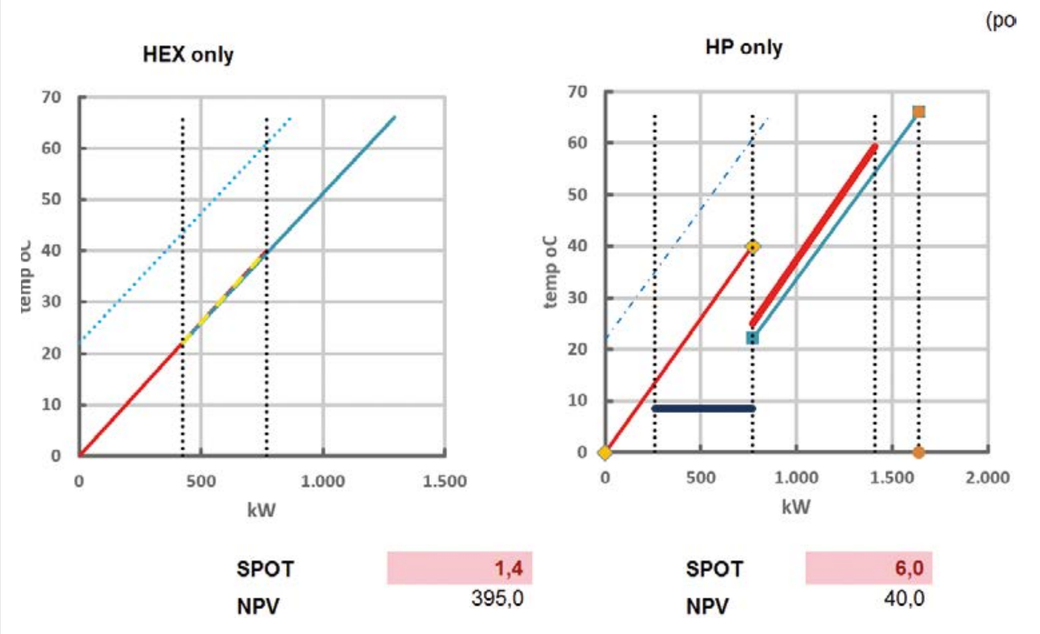
For single connections the heat exchange can be presented in pinch diagrams to understand the results of calcu-

TOP HOT STREAMS

T below oC		Warm sewer water (polluted with grease/fibers)	Air exit 4th compartm. dryer 2 with cond.	Air exit regen. silicagel dryer 1 with cond.	Air exit 4th compartm. dryer 2 (< 80%) winter + summer	Air exit regen. silicagel dryer 1(<80%)					
	ranking	1	2	3	4	5	6	7	8	9	10
	stream	3	5	2	4	1	0	0	0	0	0
95											
85											
70											
66											
65											
50											
40		96									
35		250									
22		39									
20		116									
14		19		14							
13		39	51	28							
11		19	26	14	14						
10		193	257	142	139	57					
0			26	14	14	6					
-1			26	14	14						
-2			77	43							
-5											
		770	463	270	181	63	-	-	-	-	-

TOP COLD STREAMS

T belo	Warm process water	Dryer tunnel 2	Warm water air heating dryer tunnel 1	Air in dryer 1 summer	Warm water regeneration LCI dryer 2 winter+ summer	28 Warm water regeneration LCI dryer 2 winter+summe	Warm water air heating dryer tunnel 2	Warm process water	Warm processw	
	1	2	3	4	5	6	7	8	9	10
	1	9	4	5	8	7	6	2	3	0
95								93		
85			750		300	300	350	140		
70					80	80		37		
66	20				20	20		9		
65	297			275						
50	198	500		183					33	
40	99	250		92					17	
35	257								43	
22									7	
20										
14										
13										
11										
10										
-1										
-2										
-5										
	870	750	750	550	400	400	350	280	100	-



lations. For direct heat exchange 4 possible solutions exist (Fout: Bron van verwijzing niet gevonden). Also the heat pump configuration has 4 solutions.

5. Example food industry

A food industry producing gelatine has carried out a heat pump scan. Firstly, the hot and cold streams are described (Fout: Bron van verwijzing niet gevonden). Secondly, the streams are visualized in a temperature-power table (Fout: Bron van verwijzing niet gevonden) and thirdly, in the composite curve (Fout: Bron van verwijzing niet gevonden). Finally, the grand composite curve (Fout: Bron van verwijzing niet gevonden) is constructed. The results of the feasibility calculations of single combinations as shown in the above pictures are presented in table 3.
The waste heat in the outlet of the silica gel air dryer is split into a cascade of two different streams: an economizer and a condenser (hot 1,2 and 4,5). This is because the cooling trajectory is not linear. Most of the cold streams can be used directly in the water circuit. Only the drying air must have a heat exchanger with a temperature difference of 30 K (table 1; cold 5). The total available waste heat is 1.747 kW and the heat consumption 4.450 kW.
The table with the circuits temperature ranges and the power on the specific level (Fout: Bron van verwijzing niet gevonden) gives insight in the possible connections. Technically seen a direct heat transfer in the ranges 50 – 35 oC is possible between sewer water and warm process water (96+250=346 kW).

The composite curve of this case (Fout: Bron van verwijzing niet gevonden) shows a pinch temperature of 22 oC. The potential of direct heat exchange (the overlap) is 355 kW. The minimum power needed theoretically for heating is 4095 kW. All displayed temperatures are circuit temperatures (corrected for the “temperature loss” in heat exchangers).

The grand composite curve (Fout: Bron van verwijzing niet gevonden) shows a heat pump potential of 337 kW at a temperature level of about 40 oC (fig. 8).
The feasibility calculations (Fout: Bron van verwijzing niet gevonden) show three combinations with a positive Net Present Value (NPV). Unfortunately, the presented solutions all share the same source of heat (the sewer water). Pre-heating warm process water gives a positive NPV of € 395.000,-. The same case with a heat pump results in a net present value of € 40.000,- and a payout time of six years. So in this case a heat pump is attractive, however with the fuel and electricity tariffs in the Netherlands, this case is not very convincing. All cases are documented automatically with detailed information about the piping dimensions, estimated costs, maintenance, energy savings, profits and payback period. Also combinations of direct heat exchange and heat pumps are evaluated but prove to be not attractive. Also the pinch diagrams of all cases can be shown (Figure 3).
Simple Pay Out Time (SPOT) and the Net Present Value (NPV). The heat pump case let see a CO_2 heat pump with a gliding temperature of the condenser (thick red line).

6. Other investigations

- Energy Matters executed a survey for the evaluation of high temperature heat pumps in the Netherlands.

ranking	line	nr in list	Maximum NPV's in k€		MOST ATTRACTIVE COLD STREAMS				
					Warm proces water	Dryer tunnel 2	Warm proceswater		
				ranking	1	2	3	4	5
				line	0	0	0		
most attractive hot streams				nr in list	0	0	0		
				stream nr	1	9	3		
1	0	0	Warm sewer water (polluted with grease/fibers)	3	395	80	20		

About 10 industrial firms were involved and executed a "one day heat pump scan". Some remarkable results are :

- Most firms are not aware of the technical possibilities of waste heat utilisation;
- Most firms have problems to acquire reliable data of waste heat streams;
- All industries can find waste heat utilisation with a positive NPV2;
- One of the firms could achieve an all-electric situation by rearranging and extending vapor recompression;
- The investments of industrial heat pumps (especially high temperature > 80 oC) are rather high;
- Though the savings of heat pumps are superior, direct heat exchange is preferred mostly;
- Low electricity tariffs in combination with growing fuel tariffs makes industrial heat pumps attractive;
- Pilot projects are needed to give more information about the advantages of industrial heat pumps.

7. Further development

Further development is initiated to:

- adjusting investment relations based on realized projects;
- bringing in heat transformers;

8. Conclusions

The scan method developed gives insight in the possibilities of waste heat utilization. If the process parameters are known it takes only one day to bring in the data needed and to make a report. By using the pinch method for calculations and presentations, customers are better able to comprehend the used method.

HOT STREAMS				Process				HEX	clean C	Circuit							coördinates [m]		
				Power	running	Tstart	Tend	dThex		power	Tstart	Tend	dT	C	Oper.	part			
hot	line	nr	description	peak kW	hrs/yr	oC	oC	K		kW mean	oC	oC	K	kW/K	hrs/yr		X	Y	Z
1	0	0	Air exit regen. silicagel dryer 1(<80%)	63	2.800	40	29	30		63	10	-1	11	5,73	2800	100%	58	54	7
2	0	0	Air exit regen. silicagel dryer 1 with cond.	270	2.800	29	10	15		270	14	-5	19	14,21	2800	100%	58	55	7
3	0	0	Warm sewer water (polluted with grease/fibers)	770	8.000	45	5	5		770	40	0	40	19,25	8000	100%	136	72	0
4	0	0	Air exit 4th compartm. dryer 2 (< 80%) winter + su	181	4.500	41	28	30		181	11	-2	13	13,92	4500	100%	56	64	7
5	0	0	Air exit 4th compartm. dryer 2 with cond.	463	4.500	28	10	15		463	13	-5	18	25,72	4500	100%	56	65	7
			total	1.747 kW				total		1.747 kW									

demo case 1

COLD STREAMS				Process				HEX	clean C	Circuit							coördinates [m]		
				Power	running	Tstart	Tend	dThex		power	Tstart	Tend	dT	C	Oper.	part			
cld	line	nr	description	kW	hrs/yr	oC	oC	K		kW mean	oC	oC	K	kW/K	up hrs/yr		X	Y	Z
1	0	0	Warm proces water	870	8.064	22	66	0	C	870	22	66	44	19,77	8064	100%	80	48	1
2	0	0	Warm proceswater	280	8.000	65	95	0	C	280	65	95	30	9,33	8000	100%	80	49	1
3	0	0	Warm proceswater	100	5.000	20	50	0	C	100	20	50	30	3,33	5000	100%	80	50	1
4	0	0	Warm water air heating dryer tunnel 1	750	7.500	70	85	0	C	750	70	85	15	50,00	7500	100%	66	30	1
5	0	0	Air in dryer 1 summer	550	3.000	5	35	30		550	35	65	30	18,33	3000	100%	58	56	5
6	0	0	Warm water air heating dryer tunnel 2	350	7.500	70	85	0	C	350	70	85	15	23,33	7500	100%	66	30	1
7	0	0	28 Warm water regeneration LiCl dryer 2 winter+s	400	4.500	65	85	0	C	400	65	85	20	20,00	4500	100%	66	32	1
8	0	0	Warm water regeneration LiCl dryer 2 winter+sum	400	4.500	65	85	0	C	400	65	85	20	20,00	4500	100%	66	32	1
9	0	0	Dryer tunnel 2	750	7.500	35	50	0	C	750	35	50	15	50,00	7500	100%	67	30	1
			total	4.450 kW				total		4.450 kW									

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Heat recovery in milk powder trying by using a liquid sorption process

Author: Diego Pineda Quijano, Michel van der Pal, Carlos Infante Ferreiraa, Robert de Boer, Jasper Vollenbroek

Abstract

The last step in the production of milk powder is drying, an energy intensive process that demands 30% to 40% of the total energy input of a typical plant. It takes place in Spray Dryers (SD), where concentrated milk is sprayed and placed in direct contact with hot and dry air that cools down and gains humidity as water is evaporated from milk. The warm and humid air leaving the SD contains a small portion of potentially recoverable sensible heat and a large portion of latent heat that is impractical to recover by direct condensation due to the low dew point of this stream and due to the presence of milk powder particles that become sticky at high relative humidity values. In this research, the thermodynamic feasibility of a liquid sorption system for the recovery of heat from the exhaust of SD's was investigated. The system proposed has two main advantages: the dehumidification of air in the absorber for reuse in the SD, and the production of medium pressure steam in the regenerator for integration to the steam network of the plant. A mathematical model was implemented in Matlab, and two system configurations were evaluated. The calculations showed that a SD equipped with this system can achieve energy savings between 58% and 99% when using aqueous solutions of phosphoric acid as liquid desiccant depending on the system configuration. The challenge with this liquid desiccant remains on the construction materials.

1. Introduction

Drying is an energy intensive process that accounts for 10

Nomenclature		reg	Regeneration
	Energy savings, [%]	ref	Reference
	Specific enthalpy, [Jkg-1]	SHX	Solution Heat Exchanger
	Mass flow, [kgs-1]	slte	Solute (anhydrous desiccant)
	Pressure, [kPa]	sltn	Liquid desiccant solution
	Heat transfer rate, [W]	ss	Strong solution
	Absolute temperature, [K]	ws	Weak solution
		v	Vapor
Greek letters		VHX	Steam condenser
	Mass fraction of solute, [kgsltekgsltn-1]		
	Abs. humidity of humid air, [kgwaterkgda-1]	*Superscripts*	
Subscripts			Gas phase
a	Humid air		Liquid phase
abs	Absorber	*sat*	Saturation
j	Position in a column		Vapor (refer to vapor pressure)
da	Dry air		

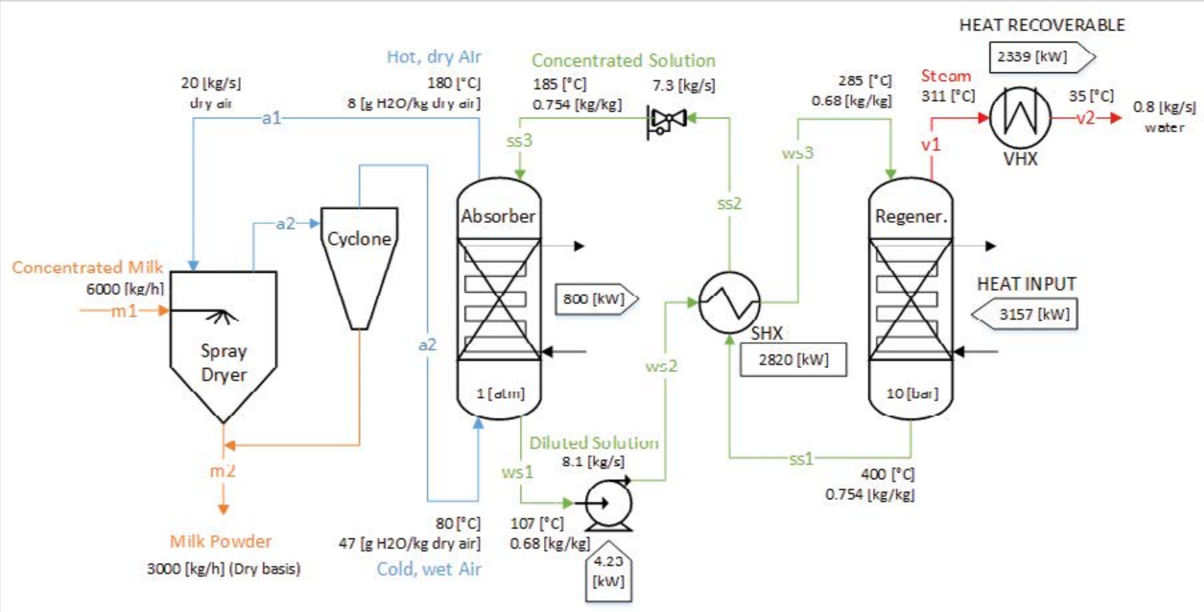

Fig. 1. Closed configuration of the EELS system integrated to a SD.

– 20 % of the total energy used in industry in developed countries [6]. It is the last step in the production of many products in several industrial sectors including food and beverage, pharmaceutical, paper, petrochemical, among others. The large energy consumption of drying processes is due to the large heat of evaporation needed to remove water or other solvents from the solid. A reduction of the energy consumption of drying processes would contribute substantially to reduce the use of fossil fuels and the greenhouse gas emissions in industry.

The drying process of milk powder is a convective-drying process that commonly takes place in Spray Dryers (SD). In a conventional milk powder SD, air enters at temperatures between 180 °C to 200 ° C and with an absolute humidity between 4 to 8 gwater/kgdry air. Concentrated milk enters the SD with 50 % solids content and at approximately the wet bulb temperature of the entering air. Special nozzles spray the concentrated milk inside the SD, and the milk droplets get into direct contact with the hot, dry air. Water is evaporated from the milk droplets and swept away in the air stream. Air leaves the SD at temperatures between 70 °C to 80 °C and humidity content between 45 to 50 gwater/kgdry air. A conventional SD for the production of 3000 kg/h of milk powder, dry basis, uses approximately 3.3 MW for heating about 20 kg/s of ambient air from 20 ºC to 180 ºC. Two thirds of this energy flow are actually used for the vaporization of water from milk, i.e., converted into latent heat, and the rest remain as sensible heat.

The simplest method for the recovery of energy in milk powder SD's is to implement an exhaust-to-inlet-air heat recovery system, which offers energy savings of the order of 15 to 20 % in comparison with the energy consumption of a SD without heat recovery [9]. The effectiveness of cross-flow, air-to-air heat exchangers, which according to Golman and Julklang (2014) [5] ranges between 50 to 70 %, limits the energy savings achievable by this method. Furthermore, even after filtering, the exhaust air contains a small fraction of milk powder particles, which may become sticky and pollute the heat exchanger if the relative humidity increases above a certain limit as the exhaust air is cooled down [1]. As a result, the temperature of exhaust air must remain above approximately 50 °C, which also limits the energy savings achievable by this method. On the other hand, the recovery of latent heat from the exhaust air by direct condensation is impractical because of the low dew point of this air stream and because of the stickiness of milk powder particles.

The sorption methods emerge as an alternative for the recovery of heat from the exhaust of milk powder SD's. Researchers of the Wageningen University and TNO, The Netherlands, have proposed the use of the zeolite-wheel technology for the recovery of heat from the exhaust of SD's [10]. Calculations indicate that this system offers energy savings between 45 to 55 % of the energy consumption of a conventional SD. However, this technology still needs to overcome some challenges related to the large heat of regeneration of the zeolite, a part of which is difficult to recover, and the risk of pollution of the zeolite material with milk powder particles present in the exhaust air.

An alternative to the zeolite-wheel system is the liquid

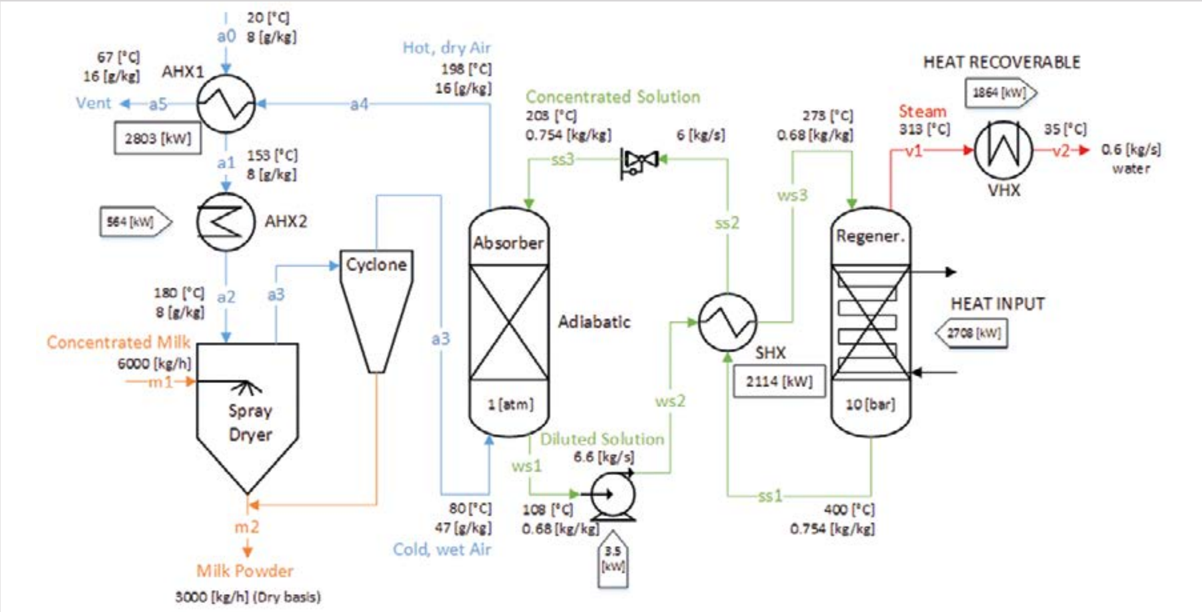

Fig. 2. Open configuration of the EELS system using an adiabatic absorber and an exhaust-to-inlet-air heat exchanger to recover sensible heat of the partially dehumidified exhaust air before it is vented.

sorption technology, which is evaluated in this study. The name given in this paper to the liquid sorption system for the recovery of heat in milk powder SD's is 'EELS', which stands for 'Energy Efficient drying by using Liquid Sorption'. The EELS system consists of an absorber, a regenerator, and a solution heat exchanger added to a conventional SD. A liquid desiccant absorbs water vapor from the exhaust air in the absorber yielding hot and dry air for reuse in the SD if a closed configuration is used or for preheating the inlet air of the SD in the case of an open configuration. The regenerator requires high-temperature heat for the regeneration of the liquid desiccant and yields medium-pressure steam. The use of this steam in other heating processes of the plant represents the energy savings of the EELS system. This study evaluates the feasibility of the EELS system from a thermodynamic standpoint. This paper will present a description of two configurations of the EELS system and a description of a mathematical model built for the evaluation of the thermodynamic limits of these configurations.

2. Description of the EELS system

The EELS system is a liquid sorption system that can be installed as an add-on to existing milk powder SD's. Two basic configurations of the EELS system will be considered in this paper: a closed configuration on the air side, and an open configuration on the air side, which will be described in what follows. A review of liquid desiccants for the EELS system is also presented at the end of this section.

2.1. Closed configuration of the EELS system

Fig. 1 shows a basic scheme of the closed configuration of the EELS system, in which air is completely recirculated to the SD after passing a non-adiabatic absorber. Ideally, the absorber yields dry, hot air at 180 °C and 8 gwater/kgdry air, the conditions required for the SD. The liquid desiccant solution enters at the top of the absorber with a temperature of about 185 °C and with a vapor pressure of approximately 1 kPa and leaves at the bottom with a temperature of approximately 107 °C and a vapor pressure of about 6 kPa. Part of the heat of absorption is transferred to the air stream, and the rest is removed from the absorber by an external cooling medium. On the other hand, the regenerator yields medium-pressure steam at the top and a concentrated solution at high temperature at the bottom. The steam produced in the regenerator is a standard commodity useful for several heating processes in the plant. The solution heat exchanger, SHX in Fig. 1, reduces the heat of regeneration by preheating the weak solution with the hot concentrated solution. The regeneration of the liquid desiccant requires a high-temperature heat flow that is comparable in quantity to the heat input of a conventional SD. The advantage of the EELS system is that the heat of regeneration is almost completely recoverable. This is because the temperature of the steam produced in the regenerator and the final temperature of the fluid used for cooling the absorber are high enough for several heating applications in the plant.

2.2. Open configuration of the EELS system

Fig. 2 shows a scheme of an open configuration of the

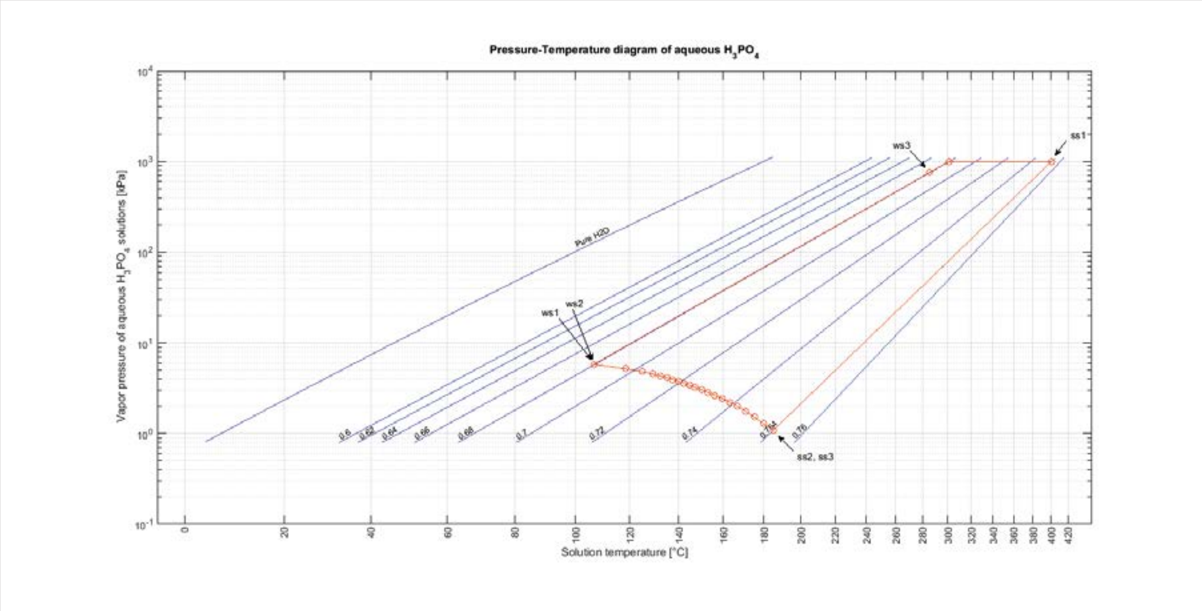

Fig. 3. P vs T diagram for aqueous solutions of phosphoric acid depicting the cycle on the liquid desiccant side of the EELS system. The concentration is expressed as the mass fraction of P2O5. The states correspond to the states of the connecting lines indicated in Fig. 1.

EELS system, which uses an adiabatic absorber for the dehumidification of the exhaust air. It also uses an exhaust-to-inlet-air heat exchanger for recovering sensible heat from the partially dehumidified exhaust air. This configuration has two main advantages. First, the simplicity of the adiabatic absorber, which also represents a lower cost. Second, air is not recirculated to the SD after passing the absorber, which eliminates the risk of contamination of milk powder with traces of liquid desiccant. The regenerator of this configuration works like the regenerator of the closed configuration. A disadvantage of this configuration is that the energy savings are lower because the exhaust air vented still contains some sensible and latent heat.

2.3. Liquid desiccants for the EELS system

The ideal liquid desiccant for the EELS system should have the following characteristics:

- It must be able to attain a temperature over 180 °C for a vapor pressure of 1 kPa. This is in order to allow dehumidifying and heating the exhaust air to the required levels for reuse in the SD.
- It must have very low or zero volatility at the conditions of the absorber and regenerator.
- It must have a high absorption capacity.

From a more practical perspective, some additional characteristics of the liquid desiccant are desirable:

- It must have low corrosiveness, or suitable construction material should be available.
- It must be non-flammable and non-toxic.

A review of the properties of several liquid desiccants commonly used in 'liquid desiccant-based air conditioning systems' led to the conclusion that these liquid desiccants are not suitable for the EELS system. Liquid desiccants like aqueous solutions of Lithium Chloride (LiCl), Calcium Chloride (CaCl2), Lithium Bromide (LiBr), Calcium Nitrate (Ca(NO3)2) were considered. It was found that these solutions are unable to reach temperatures above 180 °C for a vapor pressure of 1 kPa [3], [8], [11], condition that is necessary in order to dehumidify and heat up the exhaust air to the required levels for the SD.

On the other hand, it was found that aqueous solutions of phosphoric acid can reach temperatures over 180 °C for a vapor pressure of 1 kPa. These solutions have also negligible volatility of phosphate species for concentrations below 76 wt% P2O5 [3]. Phosphoric acid is a very corrosive fluid, but it was decided to ignore this fact in this study in order to validate the EELS concept. Therefore, it was selected as the working fluid for the EELS system. The results of the calculations presented in this paper correspond to the use of aqueous solutions of phosphoric acid as the liquid desiccant of the EELS system.

3. Mathematical model

A steady-state model of the different components of the EELS system was developed and implemented in Matlab in order to determine heat and mass flows, and the state of the liquid desiccant solution and the air stream at different locations of the system. The model yields the energy flows required for the calculation of the percentage of energy savings of both configurations of the EELS system. The subscripts of the variables in the equations present-

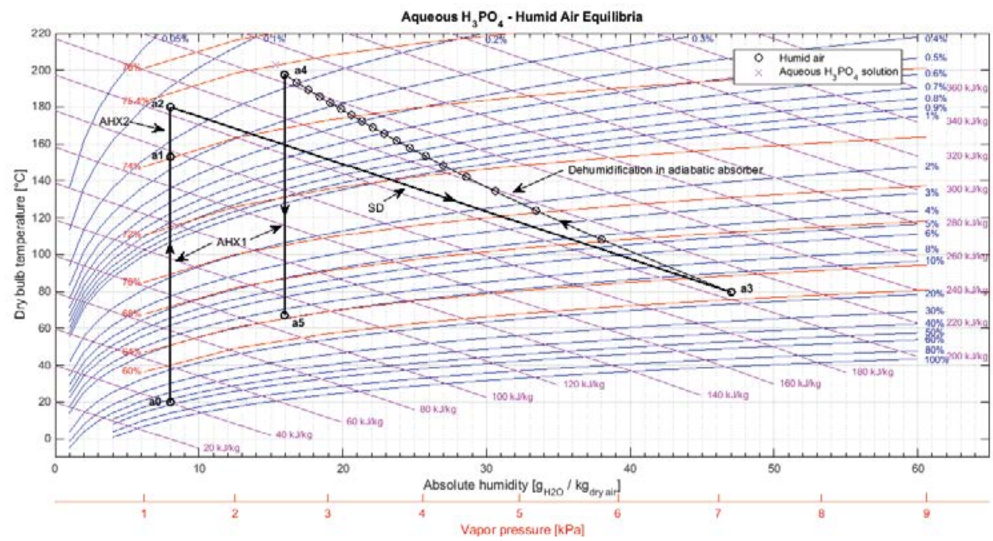

Figure 4. Psychrometric chart depicting the process of the air side of the open configuration of the EELS system

ed below coincide with the tags of the air and desiccant streams in Fig. 1 and Fig. 2.

3.1. Absorber

The main assumptions of the model of the absorber are:

- The absorber is divided into stages.
- Equilibrium is achieved between the liquid desiccant and the air stream leaving every stage. This means that humid air and the liquid desiccant have the same temperature and vapor pressure at the leaving ports of every stage.
- The absorption of other substances (i.e. CO2, O2, N2) is neglected.
- The entrainment of desiccant in the air stream is neglected.
- The desiccant is not volatile.
- Plug flow (i.e. no mixing of desiccant or air in the absorber)

The equilibrium-based model of the absorber is based on the MESH equations (Mass balance, Equilibrium, Summation, and Heat balance). In this case, due to the assumptions indicated above, the summation equations can be merged to the mass balance equation so that for every stage of the absorber only three equations are attainable. Equations (1), (2), and (3) present the mass balance, the vapor pressure equilibrium, and the heat balance for an equilibrium stage. It is relevant to mention that these equations are written in the form , which is convenient for the application of the Newton Raphson numerical method for the solution of the system of equations.

(1) $M_{H_2O,j} = \frac{\dot{m}_{slte}}{\xi_{j-1}} \cdot \left(1 - \xi_{j-1}\right) + \dot{m}_{da} \cdot \omega_{j+1} - \frac{\dot{m}_{slte}}{\xi_j} \cdot \left(1 - \xi_j\right) - \dot{m}_{da} \cdot \omega_j = 0$

(2) $E_{H_2O,j} = P^v_{sltn}(T_j, \xi_j) - P^v_{air}(\omega_j, P) = 0$

(3) $H_j = \frac{\dot{m}_{slte}}{\xi_{j-1}} \cdot h^L_{j-1} + \dot{m}_{da} \cdot h^G_{j+1} - \frac{\dot{m}_{slte}}{\xi_j} \cdot h^L_j - \dot{m}_{da} \cdot h^G_j - \dot{Q}_j = 0$

This model allows calculating the temperature and concentration of the liquid desiccant solution leaving the absorber, the rate of heat that must be removed from the column, the mass flow of desiccant needed, the number of stages, and the temperature and concentration profiles of the liquid desiccant solution and humid air along the column. The model needs as inputs the temperature and absolute humidity of the air streams entering and leaving the absorber, the mass flow of dry air throughout the column, and the temperature and concentration of the solution entering the absorber. The enthalpy and vapor pressure of the desiccant solution must be provided as functions of temperature and solute mass fraction. Similarly, the enthalpy and vapor pressure of humid air must be provided as functions of temperature and absolute humidity.

3.2. Solution Heat Exchanger

Equation (4) gives the energy balance of the solution heat exchanger. The heat duty of the solution heat exchanger can be obtained by the application of Equation (4).

(4) $\dot{Q}_{SHX} = \dot{m}_{ws} \cdot (h_{ws3} - h_{ws2}) = \dot{m}_{ss} \cdot (h_{ss1} - h_{ss2})$

3.3. Regenerator

Equations (5) and (6) correspond to the mass and energy balances of the regenerator respectively. The temperature of steam leaving the regenerator is assumed as five degrees above the saturation temperature of the weak solution at the pressure of the regenerator, as indicated by Equation (7).

$$(5)\quad \dot{m}_v = \left(\frac{1}{\xi_{ws}} - \frac{1}{\xi_{ss}}\right) \cdot \dot{m}_{slte} = (\omega_{a2} - \omega_{a3}) \cdot \dot{m}_{da}$$

$$(6)\quad \dot{Q}_{reg} = \left[\left(\frac{1}{\xi_{ss}}\right) \cdot h_{ss1} - \left(\frac{1}{\xi_{ws}}\right) \cdot h_{ws3} + \left(\frac{1}{\xi_{ws}} - \frac{1}{\xi_{ss}}\right) \cdot h_{v1}\right] \cdot \dot{m}_{slte}$$

$$(7)\quad T_{v1} = T^{sat}_{sltn}(P_{reg}, \xi_{ws}) + 5$$

3.4. Air-to-air heat exchanger and air heater

Equation (8) presents the energy balance of the air-to-air heat exchanger of the open configuration of the EELS system. On the other hand, Equation (9) defines the effectiveness of this heat exchanger. This definition takes into account that the stream with the lowest heat capacity is the ambient air stream entering the system. Equation (9) is important because it allows calculating by assuming the effectiveness of this heat exchanger. Finally, Equation

(10) gives the heat duty of the air heater.

$$(8)\quad \dot{Q}_{AHX1} = \dot{m}_{da} \cdot (h_{a1} - h_{a0}) = \dot{m}_{da} \cdot (h_{a4} - h_{a5})$$

$$(9)\quad \varepsilon_{AHX1} = \frac{T_{a1} - T_{a0}}{T_{a4} - T_{a0}}$$

$$(10)\quad \dot{Q}_{AHX2} = \dot{m}_{da} \cdot (h_{a2} - h_{a1})$$

3.5. Energy savings

Equation (11) is useful for the calculation of the energy savings offered by both configurations of the EELS system considered in this paper. Equation (11) compares the overall energy consumption of the EELS system with the energy consumption of a reference system. The reference system is a conventional milk power SD without heat recovery for the production of 3000 kg/h of milk powder, dry basis. Equation (12) gives the heat input of the reference system, where the ambient temperature is assumed equal to 20 °C and the temperature of the drying air is assumed equal to 180 ºC. The mass flow of dry air for this SD is 20. kg/s.

The definition of the energy savings assumes that the steam produced in the regenerator of the EELS system is used elsewhere in the plant. Therefore, the heat released by condensing this steam represents energy savings for the system. The definition of the energy savings also assumes that the heat removed from the absorber by an external cooling medium is used elsewhere in the plant, and, therefore, this heat flow also represents energy savings for the system. On the other hand, it is important to highlight that is zero for the closed configuration and that is zero for the open configuration as it can be inferred from Fig. 1 and Fig. 2 respectively. Furthermore, this definition leaves out the power input of the solution pump because it is negligible in comparison with the heat flows.

$$(11)\quad ES = \frac{\dot{Q}_{ref} - (\dot{Q}_{reg} + \dot{Q}_{AHX2} - \dot{Q}_{abs} - \dot{Q}_{VHX})}{\dot{Q}_{ref}} \times 100$$

$$(12)\quad \dot{Q}_{ref} = \dot{m}_{da} \cdot (h_{a@180^\circ C} - h_{a@20^\circ C})$$

4. Thermodynamic Properties

The mathematical model of the EELS system requires correlations for the calculation of the vapor pressure and enthalpy of aqueous phosphoric acid solutions. A literature review led to reported data of vapor pressure and enthalpy of phosphoric acid solutions, which allowed deriving the required correlations as functions of the temperature and concentration of the solution. On the other hand, the model also requires a correlation for the calculation of the enthalpy of humid air. This correlation was derived by using the ideal gas mixture approach and enthalpy data of water vapor and dry air obtained from Refprop.

4.1. Vapor pressure of aqueous phosphoric acid solutions

A correlation was derived for the calculation of the vapor pressure of phosphoric acid solutions. This correlation relates the vapor pressure of the solution with its temperature according to Antoine Equation, as shown in Equation (13). The coefficients A and B of Equation (13) were correlated from experimental data as third degree polynomials of the concentration of the solution.

$$(13)\quad \log(P^{v}_{sltn}) = A - \frac{B}{T}$$

The accuracy of this correlation was checked with reported experimental data [2], [4], and good agreement was found in the working range of the absorber.

Property	SD	Abs.	Property	Abs.	SHX-Cold	Reg.	SHX-Hot
Air side			Liquid desiccant side				
[°C]	180	80	[°C]	185	107	285	400
[°C]	80	180	[°C]	107	285	400	185
[gkg-1]	8	47	[kgkg-1]	0.754	0.68	0.68	0.754
[gkg-1]	47	8	[kgkg-1]	0.68	0.68	0.754	0.754

Table 1. Summary of temperatures and concentrations of the closed configuration

Property	AHX1 cold	SD	Abs.	AHX1 hot	Property	Abs.	SHX cold	Reg.	SHX hot
Air side					Liquid desiccant side				
[°C]	20	180	80	198	[°C]	203	108	273	400
[°C]	153	80	198	67	[°C]	108	273	400	203
[gkg^{-1}]	8	8	47	16	[$kgkg^{-1}$]	0.754	0.68	0.68	0.754
[gkg^{-1}]	8	47	16	16	[$kgkg^{-1}$]	0.68	0.68	0.754	0.754

Table 3. Summary of temperatures and concentrations of the open configuration

Property	Value	Property	Value
[kgs^{-1}]	20	[kW]	800
[kgs^{-1}]	5.5	[kW]	3157
[kgs^{-1}]	0.8	[kW]	3367
		[kW]	2339

Table 2. Summary of mass and heat flows of the closed configuration

4.2. Specific enthalpy of aqueous phosphoric acid solutions

The enthalpy of aqueous phosphoric acid solutions per unit mass of solution was calculated by using a correlation derived from the enthalpy data reported by Luff (1981) [7]. Equation (14) gives the derived correlation. Parameters 'a' and 'b' of Equation (14) were correlated as second degree polynomials of the concentration of the solution.

(14) $h_{sltn} = 1000 \cdot (a \cdot T - b)$

5. Results

The mathematical model was useful for determining the thermodynamic limits of the closed and open configurations of the EELS system considered in this paper. The results presented in this section correspond to ideal systems working under ideal conditions.

5.1. Closed configuration

Table 1 summarizes the states of the main air and liquid desiccant streams obtained from the calculations concerning the closed configuration, and Table 2 shows the relevant mass and heat flows. Fig. 1 presents these results in a process flow diagram, and Fig. 3 shows the cycle of the liquid desiccant side on a pressure, temperature, concentration diagram (PTX diagram) of aqueous phosphoric acid solutions, which was obtained by using Equation (13). In the absorber, the dehumidification of the exhaust air is approximately isenthalpic because an external cooling medium removes 800 kW of heat that corresponds approximately to the difference between the heat of absorption and the latent heat of condensation of water vapor. The desiccant solution remains at subcooled conditions along the absorber as it is possible to see in the PTX diagram of Fig. 3 if taking into account that the absorber works at atmospheric pressure. It is also possible to see in the PTX diagram that the maximum temperature of the liquid desiccant is 400 °C, corresponding to the solution leaving the regenerator.

This configuration offers a percentage of energy savings of 99 % with respect to a SD without heat recovery. The calculation of energy savings by using Equation (11) considers that the heat rejected from the absorber is used for heating other processes in the plant and that the steam produced in the regenerator is condensed and subcooled down to 35 °C.

5.2. Open configuration

Fig. 4 shows a psychrometric chart depicting the process of the air side of the open configuration of the EELS system. As it is possible to see in Fig. 4, the absorber yields air at 198 °C and 16 gwater/kgdry air, a higher temperature and a higher absolute humidity in comparison with the previous case. This results from the use of an adiabatic absorber and from the fact that the liquid desiccant enters the absorber at a higher temperature, 203 °C. Heating the exhaust air up to such a high temperature in the absorber is convenient in order to heat the ambient air entering the system up to a higher temperature in the exhaust-to-inlet-air heat exchanger. The effectiveness of this air-to-air heat exchanger was assumed equal to 75 %. Table 3 summarizes the states of the main air and liquid desiccant streams obtained from the calculations of the open configuration. These results are also indicated in Fig. 2. On the other hand, Table 4 summarizes relevant mass and heat flows. From Table 4, it is relevant to highlight that the mass flow of steam is smaller in the open configuration because less water is transferred from air to the desiccant solution in the absorber. This also leads to lower energy savings by condensation of steam in other processes of the plant.

Property	Value	Property	Value
[kgs-1]	20	[kW]	564
[kgs-1]	4.5	[kW]	2708
[kgs-1]	0.6	[kW]	3367
		[kW]	1864

Table 4. Summary of mass and heat flows of the open configuration

This open configuration of the EELS system offers a percentage of energy savings of 58 %. Variables such as the number of stages of the absorber, the mass flow of desiccant, and mainly the effectiveness of the exhaust-to-inlet-air heat exchanger influence the energy savings achievable with this configuration.

6. Conclusions

This paper described the EELS system proposed for the recovery of heat from the exhaust of milk powder spray dryers. It was found that aqueous phosphoric acid solutions have the required thermodynamic properties for the dehumidification of the exhaust air in the absorber in such a way that its latent heat is converted into sensible heat. Two configurations of the EELS system were evaluated from a thermodynamic point of view.

The closed configuration offers energy savings of about 99 % in comparison with a SD without heat recovery. In this configuration, heat enters the system only in the regenerator and leaves the system with the cooling medium of the absorber and with the steam flow from the regenerator. The temperature of these two streams is high enough for several heating applications in the plant. Therefore, under ideal conditions, 99 % of the energy input of this configuration is potentially recoverable.

On the other hand, the open configuration of the EELS system offers energy savings of 58 % with respect to the energy input of a conventional SD without heat recovery. In this case, heat enters the system in the regenerator and in an air heater and leaves with the exhaust air, vented after preheating the incoming ambient air, and with the steam flow from the regenerator. The absorber of this configuration is adiabatic. Energy is only recoverable from steam because the final temperature of the exhaust air is too low, 67 ºC. As a result, the energy savings are much lower in this case.

The simplicity and thus reduced cost of the adiabatic absorber makes the open configuration attractive from a practical point of view. Furthermore, this configuration eliminates the risk of contamination of milk powder with liquid desiccant.

For the conditions of the absorber, construction materials such as Hastelloy C and graphite composites can handle phosphoric acid solutions properly. For the conditions of the regenerator, there is a lack of suitable construction materials. Further research is needed in this direction.

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Waste Heat recovery in industrial batch processes: analysis of combined heat storage and heat pump application.

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Abstract

Heat is an important energy source for the chemical industry to drive their processes. Waste heat recovery and re-use therefor provides attractive energy saving and cost saving opportunities. However, in batch processes the availability of waste heat and the demand for heat are shifted in time and storage of thermal energy is needed to overcome this mismatch.

This study looks into the feasibility of a collective waste heat recovery system for a series of batch reactors. An exothermal reaction is carried out in these reactors and the surplus heat is in the current configuration removed by cooling water. In the studied configuration for waste heat recovery, the heat from the batch reactors is collected in a thermal storage system, which is used as heat source for a heat pump system. This heat pump can lift the temperature of the waste heat to the level that it can be fed to the on-site steam supply system.

A dynamic model is developed that incorporates the waste heat supply from the reactors, the thermal storage system and the heat pump. The model is used to study the impacts of thermal storage capacity and heat pump capacity on the waste heat recovery potential and amount of steam produced by the heat pump. The thermal storage system levels out the fluctuations in heat supply, enabling a more constant operation of the heat pump and a reduction in need for any backup steam supply. The analysis indicates that the heat storage system can be economically feasible, achieving pay back times of less than 5 years. The price of electricity is an important factor on the economic analysis.

1. Introduction

Improving the security of energy supply and reducing greenhouse gas (GHG) emissions are priority concerns for the Energy Union. Given that it helps to moderate energy

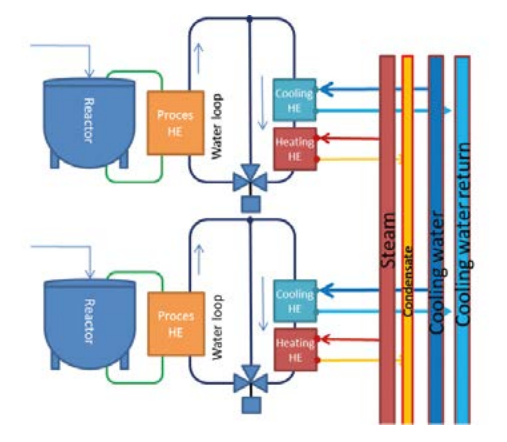

Figure 1

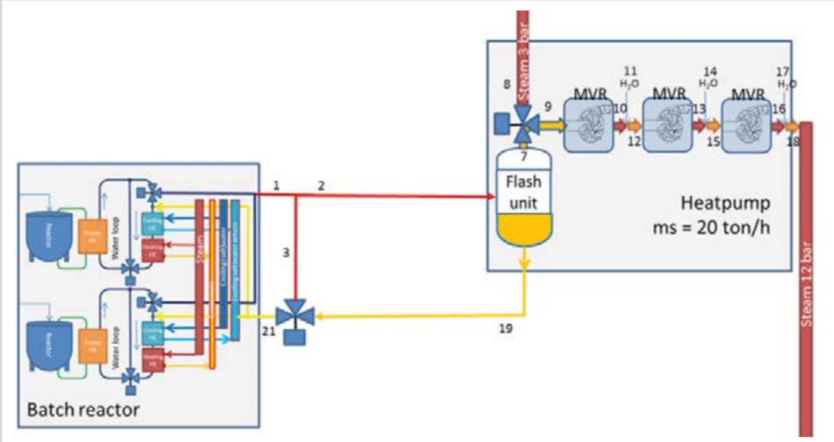

Figure 2

demand and mitigate climate change, energy efficiency is one of the five pillars in the EU quest for greater energy security, sustainability and competitiveness [1] .

In the process industry a large share of the energy is needed for heating purposes. About 2/3 of the Dutch industrial energy demand is needed for heating purposes. This can be direct gas fired heating of reactants, production of steam and hot water. At the same time, waste heat at lower temperatures is released to the ambient through cooling towers and cooling water circuits. Re-using this waste heat within the processes can be a good solution to increase the industrial energy efficiency, and reduce the CO2 emissions. The application of heat pump technology to upgrade the temperature level of industrial waste heat to allow re-use is considered as a strong building block for industrial energy efficiency [2].
In the chemical and food industry a significant share of the processes is operated in batch mode. These processes have a discontinuous character, which limits the possibilities for heat integration and direct re-use of waste heat [3]. Thermal energy storage uncouples in time the supply and demand of thermal energy and thus allows to enlarge the potential for re-use of waste heat.

The current study is looking into the contribution of a thermal storage system in improving the recovery and re-use of waste heat in industrial batch processes. A case study is done to analyse the feasibility of waste heat recovery system consisting of a mechanical steam recompression system combined with a thermal storage unit to smooth the variations in waste heat supply from the batch process.

2. System description

The industrial process considered in the current study, consists of a series of batch reactors that are operated independent from each other. The batch reactors are used for a polymerization process, that requires initial heat input to initiate the reaction, followed by heat removal

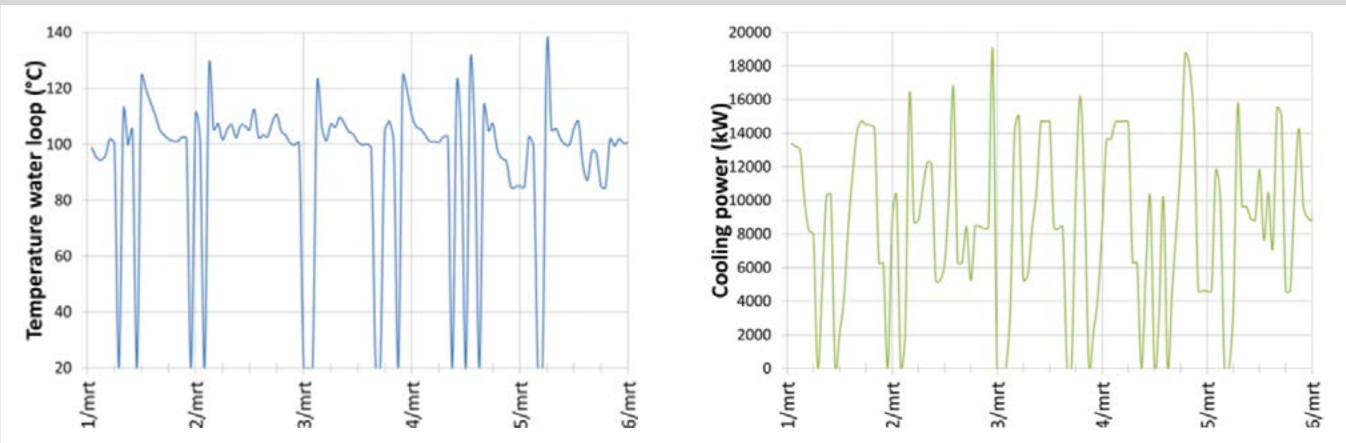

Figure 4

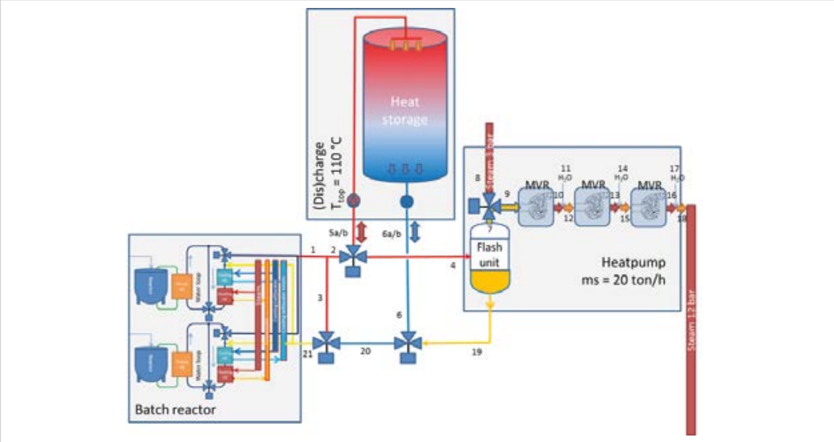

Figure 3

to keep the exothermic polymerization process within its temperature boundaries When the batch is ready the product is cooled down before emptying the batch. The heating and cooling operation is applied to a pressurized intermediate water loop that transfers the heat to and from the reactor product, as is schematically drawn in figure 1. The intermediate water loop operates in the range of 20 to 140°C. The heat supply to the reactors is done by the steam from a centralized steam utility, the heat removal from the reactors is done by the cooling water circuit.

To recover the waste heat from the batch reactors a new scheme is proposed that allows to use the exothermal heat of the batch reactors as a source for a steam compression heat pump system. (see figure 2) The heat pump, a mechanical vapour recompression (MVR) system, is connected with a flash vessel that provides low pressure steam at the suction side of the MVR. The flash unit receives the pressurized water from the intermediate water loop and the excess heat is removed by flash evaporation. The existing cooling and heating facilities remain operational to assure the required temperature control of the batch reactors.

The proposed MVR operates in the range from atmospheric pressure at the suction side to produce steam at 12barg at the outlet, which can be integrated with the existing steam lines on-site. In order to reach the desired temperature and pressure increase, a 3 stage compression process is introduced with desuperheating by water injection between the stages. The design output of the heat pump is set at 20 tons of steam per hour.
However, the waste heat supply from the batch reactors strongly varies over time, and sometimes equals zero. This leads to large variation in the heat pump operational conditions. For that reason an additional steam supply at 3 bar from the existing steam lines is connected at the inlet of the MVR. This allows the MVR to remain operational in case the waste heat supply would drop below the threshold inlet value of the MVR.

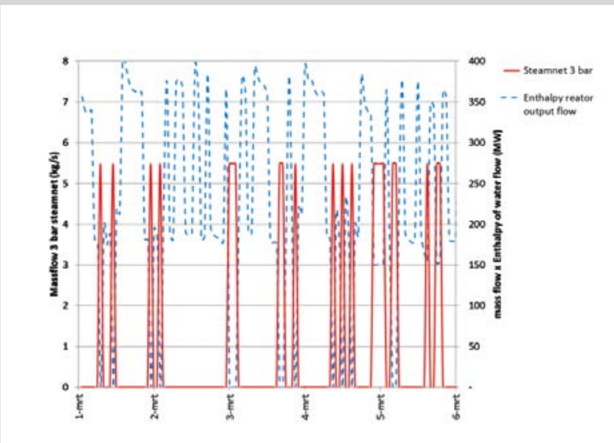

Figure 5

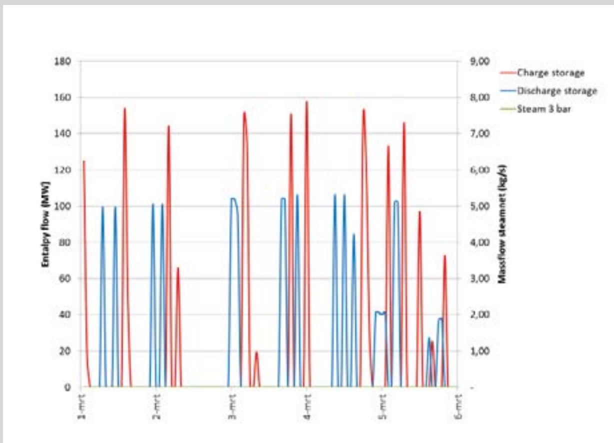

Figure 6

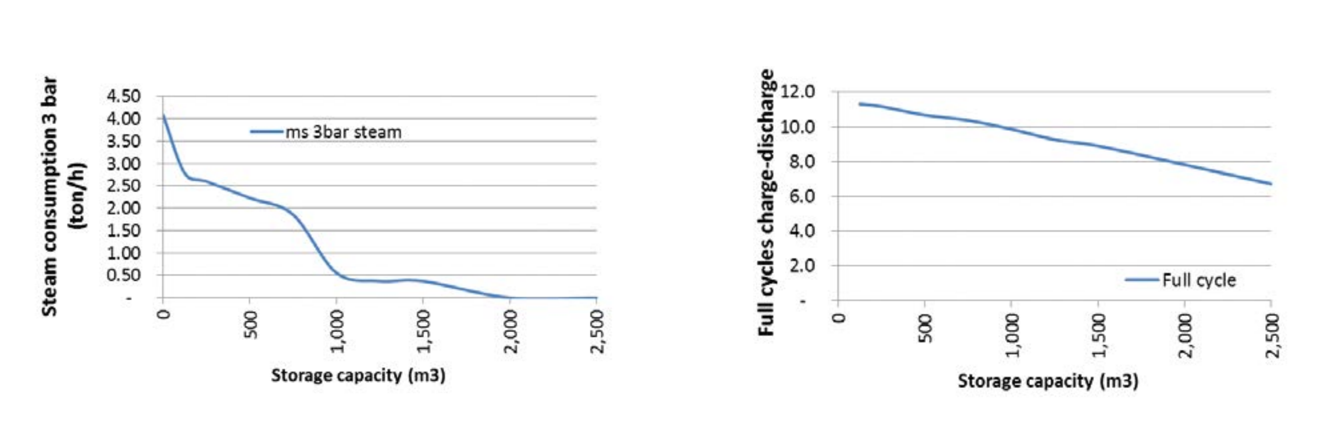

Figure 7

Whenever the waste heat supply is higher than the maximum intake of the MVR, the existing cooling system of the batch reactors will remove the surplus heat.

To improve the effective heat recovery of the proposed concept, a thermal storage system is added to it. The thermal storage system allows to take up any surplus of waste heat from the batches, and release this surplus in times of insufficient waste heat supply form the batches. The system lay out is shown in figure 3. The storage system is based on a pressurized water tank, placed between the batch reactors and the MVR. The thermal storage system is assumed as a stratified water tank that can maintain a temperature gradient inside the tank and that no mixing of hot and cold water is taking place inside the tank.

3. System model

A system model was built to simulate the process conditions and to allow analysis of the dynamics of the heat pump operation both with and without a thermal storage systems. MS-Excel is used as modelling tool combined with Refprop that provides the thermodynamic data of the steam to calculate the MVR conditions as well as the flash vessel conditions.

The MVR system was modeled as a three stage compressor having independent drive-units for the individual stages, in order to have each stage operating at its optimum efficiency. The sizing of the MVR is derived from the average value of waste heat supply, being around 10 MW. The outlet condition of the compressor was fixed at 12 bara pressure. At an inlet pressure below 0,75bara, the MVR can no longer achieve the required pressure increase. At this point the heat supply from the storage or from the 3 bar backup steam supply will be provided, in order to keep the MVR to produce 12 bar steam.

The thermal storage unit was modeled as a pressurized water tank, capable of storing hot water in the range between 140 to 80°C. The water storage volume is taken as a parameter to vary between 500 and 2500m3. Heat losses were neglected in the simulation.

The supply of waste heat from two of the batch reactors is taken as input to the model with time steps of 1 hour. Five days in a row of the waste heat supply were taken as a representative period for the operational conditions of the batch reactors. The temperature and the thermal power of the waste heat source during the 5 day period are shown below in figure 4.

4. Model results

The results of the calculations for the five day period of production for the system without thermal storage are plotted in figure 5. The plot shows in red the timing of the steam needs during the five days for those periods where waste heat from the batches is too low. In this configuration the model calculates a back-up steam need during 22 hours with an averaged steam consumption of 87 ton/day When including the thermal storage system the surplus of waste heat can be stored and during times of too low waste heat supply the stored heat is released from the storage. For the situation of 1000m3 and 2500m3 of hot water storage these charge and discharge cycles are analysed. The results of these analyses are shown below in figure 6.

The 1000m3 storage system is already very effective in reducing the backup steam needs by 85%. Only for 3 hours over a period of 5 days, the backup steam is resulting in an average steam need of 12 ton per day. On further increasing the storage size to 2500m3 the need for any backup steam has gone to zero.

The impact of the thermal storage size on the residual steam use was analysed for a range of storage sizes from 0 to 2500m3. The amount of steam needed decreases sharply with storage sizes increasing to 1000m3, but on fu-

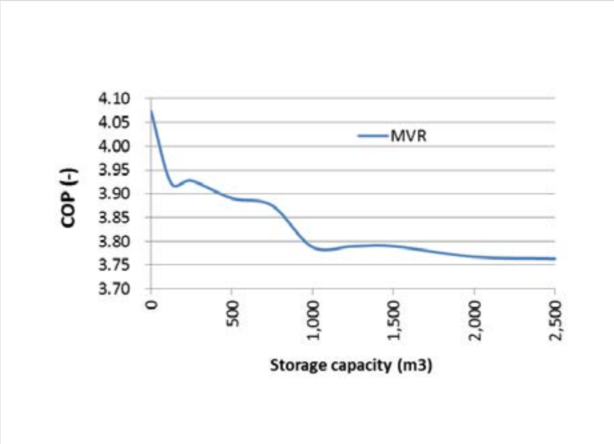

Figure 8

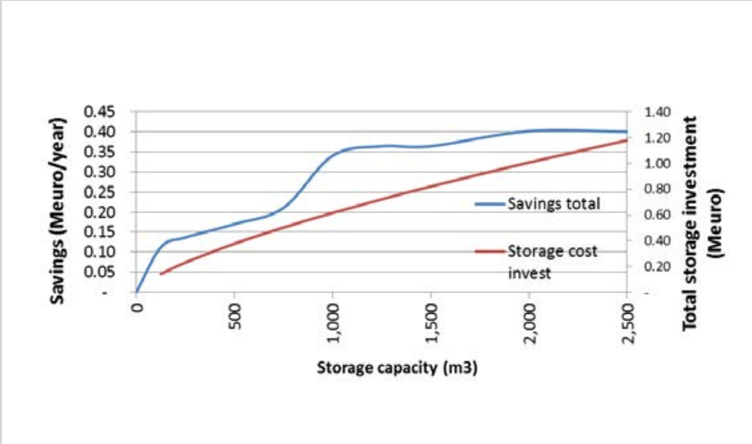

Figure 9

rther increasing the reduction in steam consumption is limited. Also the effective use of the thermal storages was calculated based on the amount of heat charged to the storage divided by the storage capacity. and expressed as the number of full cycles. The results of both analyses are shown below in figure 7.

The efficiency of the MVR system in terms of COP_heating varies with the capacity of the storage system. Without a thermal storage system the frequent backup steam supply at 3 bar, reduces the needed compression ratio in comparison to using the waste heat directly from the batch reactors or from the thermal storage system. With the 3 bar steam as a source the heat pump can thus run with higher average COP.

Techno-economic analysis

In the economic analysis of the thermal storage system the cost savings are obtained by the reduction of the amount of backup steam for the compressor. The results of the analysis for the 5 day period are extrapolated to 365 days obtain the annual figures. The calculated savings are based on steam price of 12.50 €/ton for 3 bar steam. The capital cost for the storage system is based on literature values for large industrial size systems, that are installed as thermal storage tanks in district heating systems. For a 1000m3 storage tank the investment cost are assumed to be € 600.000. A plot of the cost savings for the range of storage capacities considered is shown in figure 9, together with investment cost for the various storage sizes. The calculated steam cost saving for the 1000m3 storage system is around € 350.000.

The integration of the storage unit reduces the COP of the heat pump system, which results in additional consumption of electricity. This additional cost needs to be subtracted from the earnings to calculate the simple payback times. The cost of electricity are calculated for a range of prices from 20 to 50 €/MWh.

The calculated simple payback times for the thermal storage system for a range of capacities and various electricity prices is shown in figure 10 below.

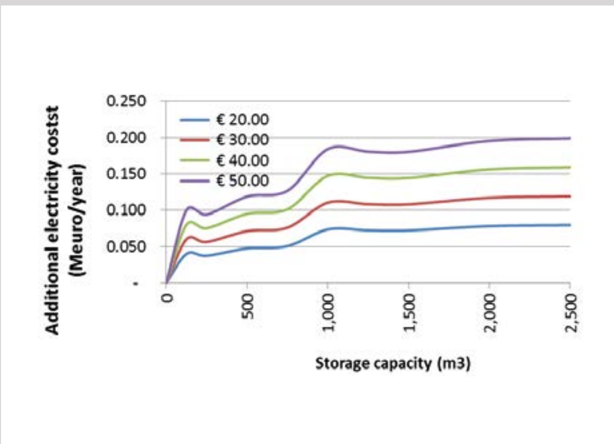

Figure 10

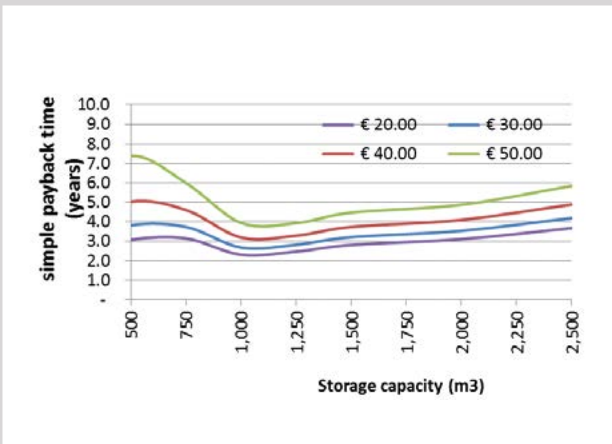

Figure 11


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5. Discussion and conclusions

In this study the recovery and re-use of waste heat from batch processes is considered, applying a mechanical vapour recompression combined with a heat storage system. The addition of the heat storage system allows to recover a larger amount of the waste heat from the batch procoss, and reduces the amount of low pressure backup steam for the MVR. A system model was developed to simulate on hourly basis the dynamics of the heat recovery system, using the dynamic waste heat supply from a real world batch reactor process.

The developed model can quantify the impact of the heat storage system on the annual need for steam backup to the MVR as well as on the COP obtained. These factors, need for backup steam and electricity demand, determine the payback time of the thermal storage. The electricity price is an important factor in the calculation of the payback time. The model simulation allows to identify an optimal range for the heat storage capacity to be installed, based on the fluctuating pattern of waste supply and selected MVR capacity. The analysis indicates that the heat storage system can be economically feasible, achieving pay back times of less than 5 years.

It is recommended in the further analysis of the business case for waste heat recovery of batch processes, to do calculations using smaller time steps, for example in steps of 5 minutes, to obtain a more detailed view on the compressor dynamics. It is further recommended to include any site specific requirements that will have impact on the cost of installation.

Acknowledgements

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Heat pumps in the process industry

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Abstract

This paper discusses a case study in which the waste heat from a refinery is used to heat up district heating water for a local district heating (DH) network. The potential of using heat pumps in order to recover heat from low temperature process water streams for DH is to be analyzed. The heat pump will have to extract heat from a source at 37°C and add it to the DH water to increase its temperature as much as possible with reasonable values of the coefficient of performance (COP). The process water is presently cooled down using cooling towers. Extracting heat from the process water serves two purposes – waste heat recovery for DH and reduction in the load sent to the cooling towers.

1. Introduction

The modern society faces a variety of challenges to meet the energy requirements of a growing population. Considering that process industry is among the most energy demandingsectors, chemical engineers have embarked on a quest for shaping a sustainable future. Due to the limitation of fossil fuels, the need for energy independence, as well as the environmental problem of the greenhouse gas effect, there is a large increasing interest in the research and development of chemical processes that require less capital investment, reduced operating costs, and lead to high eco-efficiency. The use of heat pumps is a hot topic due to many advantages, such as low energy requirements as well as an increasing number of industrial applications. Although the research and development carried out

Parameters		Units
Maximum network load	100	MW
Supply temperature (maximum)	130	°C
Return water temperature	75	°C

Table 10.1 Operating conditions of district heating system

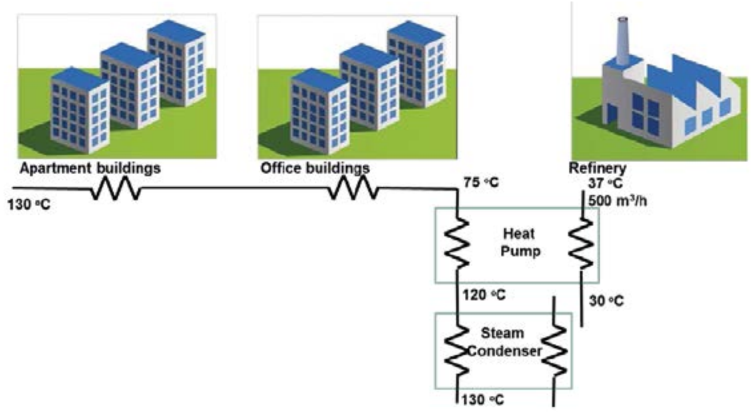

Figure 10.1 Waste heat stream of refinery to deliver heat to district heating system

in academia and industry in this field are expanding quickly, there is still no book currently available focusing on the use of heat pumps in the chemical industry.

The authors of this paper have recently published a book (Kiss and Infante Ferreira, 2017) which provides an overview of heat pumps technology applied in the process industry, covering both theoretical and practical aspects: working principle, applied thermodynamics, theoretical background, numerical examples and case studies, as well as practical applications in the chemical process industry. The worked out examples instruct the students, engineers and process designers about how to design various heat pumps used in the process industry. The reader will benefit from learning more about heat pumps, what they are, when and how to use them properly; getting more information about the theoretical and practical background of HP; understanding how to identify the need, select, design and apply heat pumps; discovering the existing and potential applications of heat pumps in a process; and finding the specifics of heat pump applications in the process industry.

The book starts with an introduction to heat pumps that provides an overview of heat pumps, and the possible sources of heat and cold usable by HPs in the chemical process industry (CPI). Then, it continues with some theoretical background related to the thermodynamics of heat pump cycles (e.g. fundamentals, enthalpy, entropy, equations of state, chemical / phase equilibrium), entropy production minimization and exergy analysis, as well as Pinch analysis and process integration (e.g. minimize energy use, maximize heat recovery). After that, the selection of heat pumps is explained, including the required steps during development, demonstration and deployment of heat pumps in industry. The next chapters cover more in-depth several types of heat pumps such as: mechanically driven heat pumps (e.g. vapor recompression, vapor compression, compression resorption, trans-critical, and Stirling heat pumps), thermally driven heat pumps (e.g. liquid-vapor absorption, solid-vapor adsorption, ejector based heat pumps), and solid state heat pumps (e.g. magnetic refrigeration, thermoelectric, and thermos-acoustic heat pumps). Heat pump applications and several case studies (e.g. application to distillation, evaporation and refrigeration) are included in the final chapters.

In this paper one of the case studies discussed: “a vapor compression heat pump for heat recovery” is reproduced as an example of the contents of the book.

2. Vapor compression heat pump for heat recovery

Waste heat from a refinery is to be used to heat up district heating water for a local district heating network. This case has been investigated by Ravi (2010). The details of the existing district heating (DH) network may be found in Table 10.2.

Purpose is to investigate the possibility of extracting waste heat from the refinery and constructing a district heating substation. The potential of using heat pumps in order to recover heat from low temperature process water streams for DH is to be analyzed. The heat pump will have to extract heat from a source at 37°C and add it to the DH water to increase its temperature as much as possible with reasonable values of the coefficient of performance

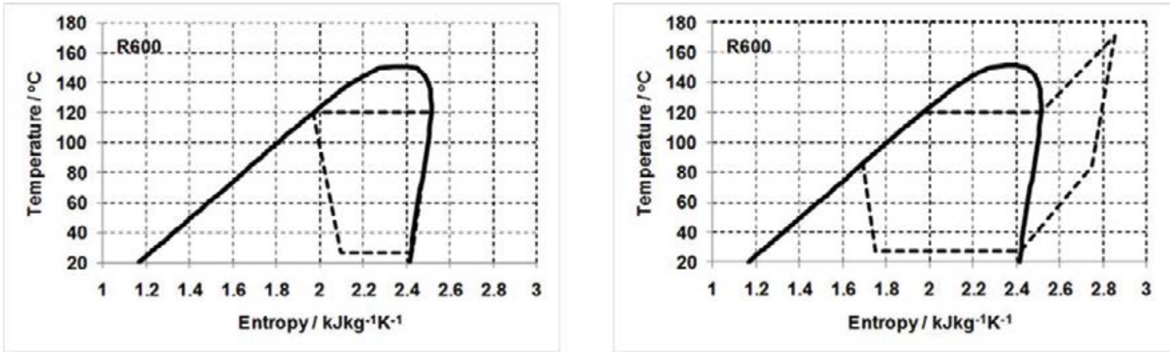

Figure 10.2 Single stage heat pump cycle of butane (R600). Left without internal heat exchanger and right with internal heat exchanger

(COP). The process water is presently cooled down using cooling towers. Extracting heat from the process water serves two purposes – waste heat recovery for DH and reduction in the load sent to the cooling towers. The available volumetric flow rate of process water is 500m3/h.

Vapor compression heat pumps are the most widely used type of heat pumps. Current maximum temperatures are limited to 120°C and, since this is the technology closest to market applications, it is the most interesting technology for this refinery case.

Current industrial heat pumps refer to 90°C as high temperature applications. However, in the case of the considered heating network, the temperatures of interest are higher than 100°C. In order to study the potential of using high temperature heat pumps, first a market study should be conducted to investigate if machines operating within these temperature ranges are available. Thereafter, a study is conducted to analyze the use of different refrigerants for the heat pump.

Commercially available high temperature heat pumps usually provide water at 90 °C. Very few manufacturers have the experience to supply large scale units (greater than 1 MW thermal capacity) in the 90 °C range. Friotherm (a Swiss firm) has extensive experience in waste heat recovery and has installed many heat pumps for district heating. Also Thermea and Combitherm (both from Germany) have experience with heat pumps that can be used to produce hot water at 90 °C for district heating. At the moment, the preferred working fluid is generally R134a (1,1,1,2-tetrafluoroethane). High temperature heat pumps are custom made and all three manufacturers had no previous experience in operation of a heat pump with sink temperature at 120 °C. For achieving the high temperature of 120°C on the sink side, Friotherm and Combitherm have proposed to use R245fa (pentafluoropropane) as the refrigerant whereas Thermea has proposed the use of CO2 refrigerant. A market survey has shown that there are no commercially available absorption heat pumps that can deliver water at 120 °C. Similarly, mechanical vapor recompression (MVR) heat pumps cannot be used in this case as the source is a liquid. The district heating water that has to be heated from 75°C is considered to be part of the refinery heat exchanger network. The pinch point for the refinery has then been calculated at 75 °C for the cold stream and 85 °C for the hot stream. Thus, using a heat pump for transfer ring heat from the stream below the pinch point (cooling water at 37 °C) to one above the pinch point (DH water at 75 °C) would be an efficient solution. DH water is to be heated from 75°C to 130°C. Some possible temperature levels for the implementation of heat pumps need to be considered. A solution that is proposed is to heat DH water from 75°C to at least 120°C using a heat pump and then to do the additional heating with medium pressure (MP) steam or waste heat recovered from flue gas. This way, the requirement for MP steam would go down and the heat pump will work with reasonable efficiency limits. The aim of the process is two-fold – cooling down process water and heating up water for district heating.

The process water can be cooled down to 30 °C. The operating conditions of the heat pump are schematically represented in Figure 10.5.

Working fluids	GWP [-]	$p_{suction}$ [bar]	p_{ratio}* [-]	$\dot{V}_{compressor}$* nohex /with hex [m^3/h]	$T_{discharge}$* no hex [°C]	$T_{discharge}$* with hex [°C]
Ammonia	0	10.7	9.0	322/308	270	349
Butane	3	2.6	9.0	1595/1418	124	165
Isobutane	4	3.7	8.0	1248/1079	126	163
R245fa	1030	1.6	12.8	2183/1949	124	167
Pentane	4	0.74	13.1	4530/4054	123	158

Table 10.2 Comparison of the working fluids in terms of performance indicators

The method introduced in Chapter 6 can be used in order to determine the performance of vapor compression heat pumps operating with diverse working fluids. The input variables used are:

- Source (process water) temperature at inlet and outlet (Tsource,in=37 °C, Tsource,out=30 °C),
- Source (process water) flow rate
- Sink (district heating water) temperature at inlet and outlet (Tsink,in=75 °C, Tsink,out=120 °C),
- Isentropic efficiency of compressor (i 0.70),
- Pinch temperature in condenser (Tmin 3 K)

The method proposed in Chapter 6, making use of REFPROP (Lemmon et al., 2013), gives thefollowing output variables:

- Pressure, temperature, enthalpy and entropy at all relevant states of the heat pump cycle,
- COP,
- Heating Capacity, Q
- Compressor Power, W

The calculation procedure aims to determine the best solution for each working fluid among feasible solutions. In order to eliminate incorrect solutions, during the course of calculations a few aspects should be verified. If the evaluated conditions do not fulfill one of the requirements, the values of the particular cycle should be discarded.

The following requirements should be met by the evaluated cycle:

- The compressor outlet temperature must be higher than the sum of the sink outlet temperature and the minimum pinch temperature
- The quality of vapor at the outlet of the compressor must be greater than 1
- The quality of the two phase vapor entering the evaporator must not exceed 0.9

The operating conditions of the condenser should prevent a temperature cross. A Q -T diagram for the condenser is used to verify this.

Three options have been considered for the cycle:

- Single stage heat pump cycle
- Single stage heat pump cycle with internal heat recovery
- Two-stage heat pump cycle

Figure 10.6 illustrates the cycles in T-s diagrams for butane. Without internal heat exchanger the discharge temperature of the compressor is close to saturation. This may lead to unacceptable operating conditions and damage of the compressor. The internal heat exchanger will generally improve the efficiency of the heat pump cycle. It always leads to an increase of the discharge temperature of the compressor. For butane the temperature reaches 165.3 °C when a hot water temperature of 120 °C is required. This is close to the limit of operation for the current compressor /lubricant designs. Notice that this only happens when an approach temperature of 3 K can be realized in the internal heat exchanger. With a less efficient heat exchanger the temperature can be limited but this will reduce the COP of the cycle.

Figure 10.7 shows the heat pump COPs that can be attained with the most suitable refrigerants. Ammonia shows a lower performance when high hot water temperature is required. It also shows a decrease in performance when an internal heat exchanger is added to the ammonia system. What is not visible from Figure 10.7 is that the compressor discharge temperature becomes extremely high when ammonia would be selected. The performance of pentane, R245ca, R245fa and butane is very similar. Since pentane and butane are fluids which most probably are already available in the plant, these fluids are much cheaper than

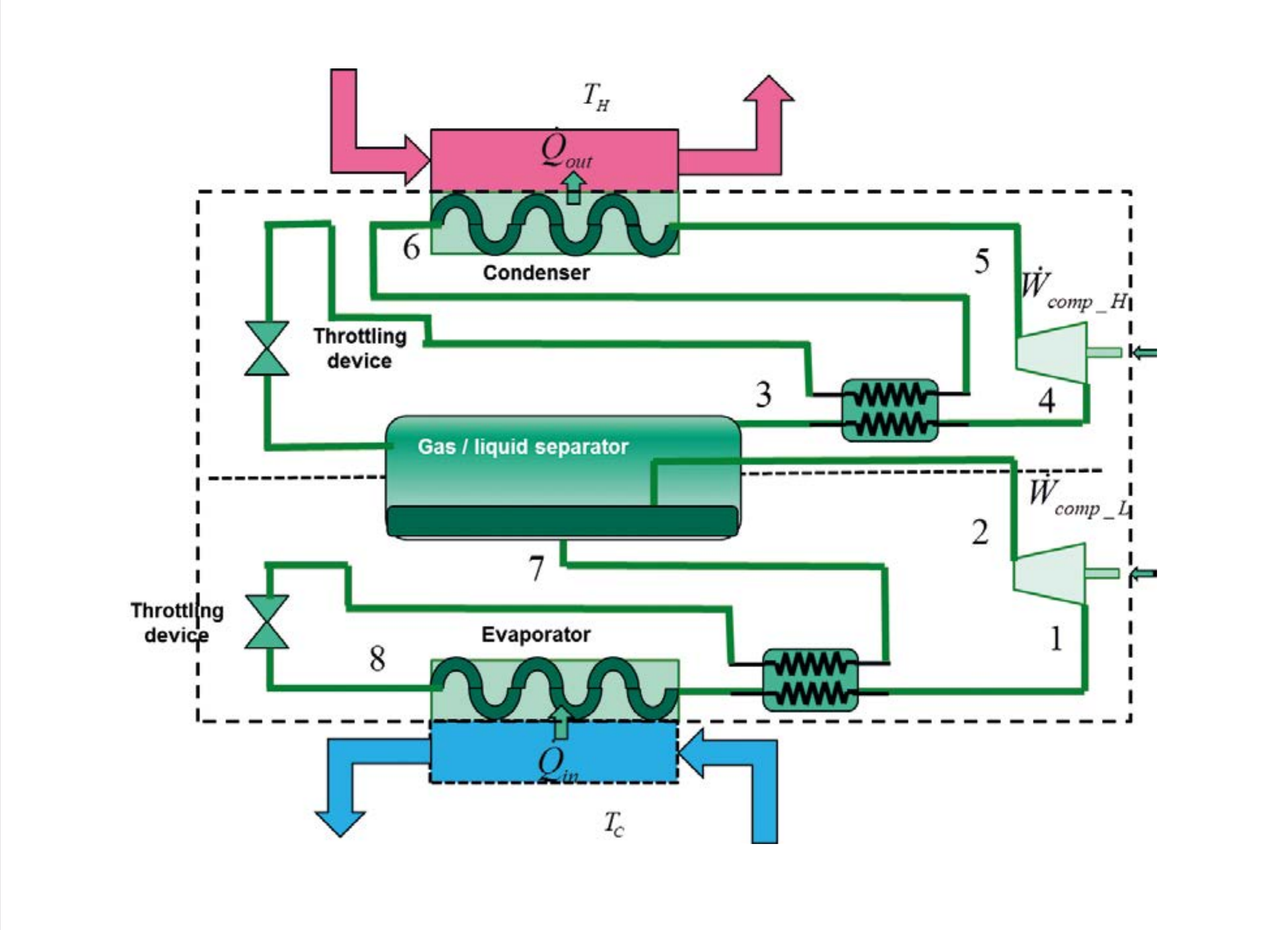

Figure 10.5 Schematic of the two stage heat pump with two internal heat exchangers.

the HFCs and the HFCs have high GWP factors, pentane and butane should be preferred as working fluids.

Figure 10.3 COP of single stage heat pump cycle of diverse refrigerants: without internal (left) and with internal heat exchanger (right)
Table 10.3 provides more details about the performance indicators of the different working fluids. The table makes clear that the operating conditions of butane are more favorable than those of pentane (operation always above atmospheric pressure and significantly lower pressure ratio). From the table is also clear that from the selected working fluids only ammonia shows extremely high discharge temperatures

* These values apply when the hot water outlet is 120 °C; the volume flow applies per MW heating capacity.
The volume flow at the compressor inlet gives an indication of the compressor size required. The smaller the volume flow, the smaller the compressor and its initial costs.
As indicated in Table 10.2 the water needs to be heated to 130 °C so that it can be delivered to the district heating network. It is always possible that part of the heating is done with the heat pump and that a fuel is further used to reach the required temperature. In this case it has been assumed that the thermal efficiency of the electrical grid is 0.42 and of a boiler 0.86. It has further been assumed that electricity costs 65 €/MWh and gas costs 32 €/MWh. Heating the water flow from 75 °C to 130 °C with a boiler would cost 10.1 M€/year. Depending on the share of the heat pump and on its COP, the heat pump will give the yearly savings shown in Figure 10.8. Obviously, using the heat pump as much as possible is always advantageous even if the COP decreases to values around 3.0 (see also Figure 10.7). The working fluid with the highest COP gives the largest yearly energy costs savings.

Figure 10.4 Yearly energy costs savings when the heat pump is applied to heat the waste heat stream from 75 °C to the indicated temperature. The remaining temperature increase up to 130 °C is obtained making use of fuel fired boiler. Left without internal heat exchanger; right with internal heat exchanger. The large pressure ratios in Table 10.3 suggest that two stage compression will possibly lead to more suitable operating conditions. For this reason the performance of two stage systems with open flash

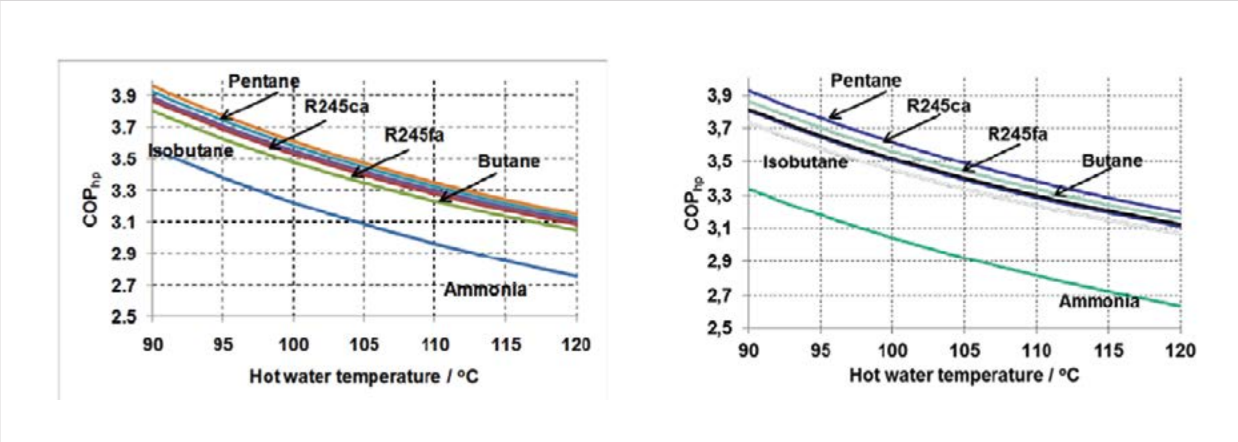

Figure 10.6 COP of two stage heat pump cycles of diverse refrigerants (left) and of single stage cycle with internal heat exchanger (right).

tanks has also been investigated. This design considers two internal heat exchangers between the liquid flow and the vapor flow before entering each of the compressors. The schematic of the process is illustrated in Figure 10.5.

An energy balance for the separator, considering it externally adiabatic, gives the ratio of the mass flows through both compressors so that the COP of the cycle can be calculated:

$$COP_{2-stage} = \frac{h_5 - h_6}{(h_2 - h_1) \times \dfrac{h_3 - h_7}{h_2 - h_9} + (h_5 - h_4)}$$

The indexes of the enthalpy correspond to the states indicated in Figure 10.9.
The ratio [(h3-h7) / (h2 -h9)] is based on the energy balance around the separator considering it externally adiabatic and gives the ratio between the mass flow through the low pressure compressor and the mass flow through the high pressure compressor. Figure 10.10 shows on the left side the predicted COP for the two stage heat pump system when diverse refrigerants are applied. To facilitate a comparison with the single stage performance (when an internal heat exchanger is applied), the results for the single stage are shown in the right hand side.

From the figure it is evident that the performance of the ammonia system significantly improves while the improvement of the other fluids is small (lower temperatures) to negligible (higher temperatures). The calculations have assumed the isentropic efficiency of the compressor to be maintained with the pressure ratio. In reality, at least for screw compressors, the efficiency will decrease with the pressure ratio so that the two-stage solution offers a better solution.

3. Conclusions

The book by Kiss & Infante Ferreira (2017) serves as a reference for scientists, researchers, engineering procurement and contracting (EPC) organizations, operators of chemical and biorefineries production facilities, R&D engineering departments, and other industry practitioners involved in heat pumps and energy efficient technologies. The example case study reproduced in this paper gives an indication of the contents of the book and illustrates the possible advantages of the implementation of heat pumps in the process industry.

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Kiss, A.A., Infante Ferreira C.A. (2017) Heat Pumps in Chemical Process Industry. CRC Press. Taylor & Francis Group, Boca Raton, USA.

Towards a sustainable
industry by electrification

Author: Ir. Jan Martin Grift rea, photo: www.shutterstock.com

Summary
Due to the cost reduction of solar and wind energy and possibilities for storage as well as other technical solutions profit from these renewables, the electrification of industry is foreseen by us as the most logic solution towards sustainability. In short term the industry should make masterplans for electrification to avoid capital destruction. The exergy losses in industry can and should be compensated by heat pumps and vapor recompression instead of fossil fuels. Pinch analyses are necessary to determine internal waste heat circuits as well as heat grids. Technically speaking, industry can become sustainable in the short time by electrification of all processes. Depending on the situation, it can be an attractive route towards sustainability.

Introduction
Industry in Europe has the ambition of achieving full CO2-neutrality before 2050. Because production lines can have lifespans over 30 years, redesign of processes and utilities at short time is necessary. Finding alternatives for fossil fuels is difficult because of the scale and the high investments needed. Geothermal sources are very capital intensive because of the deep wells needed and the uncertainty about the geological structure. The sourcing of biomass for the long term is difficult because of the rise in demand in new and existing applications. The costs of solar and wind energy are decreasing and will reach grid parity with fossil fuels within decades, even for industry. The development of affordable batteries is going very fast, increasing the possibilities for a high penetration of renewables. The all-electric solution therefore seems to be an attractive route for industry to achieve sustainability. Especially in industrial sectors where the temperature levels required are relatively low, for example the food industry. This article presents a practical approach for electrification and gives an overview of the technical solutions.

Practical approach Energy Matters
The first step is to determine the temperature levels required and the power of the so called hot and cold streams in a factory. For new processes, the “primary” pinch based on the product streams can be made. For existing proces-

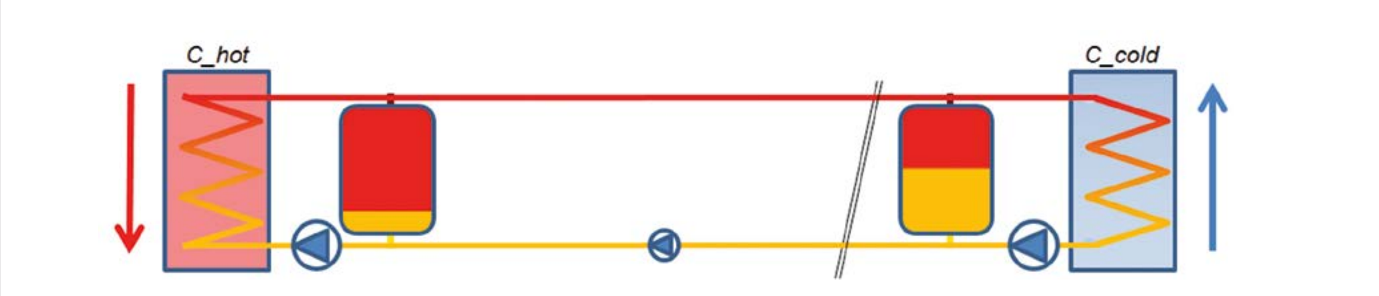

Figure 1 heat transport with continuous power by using stratified buffers.

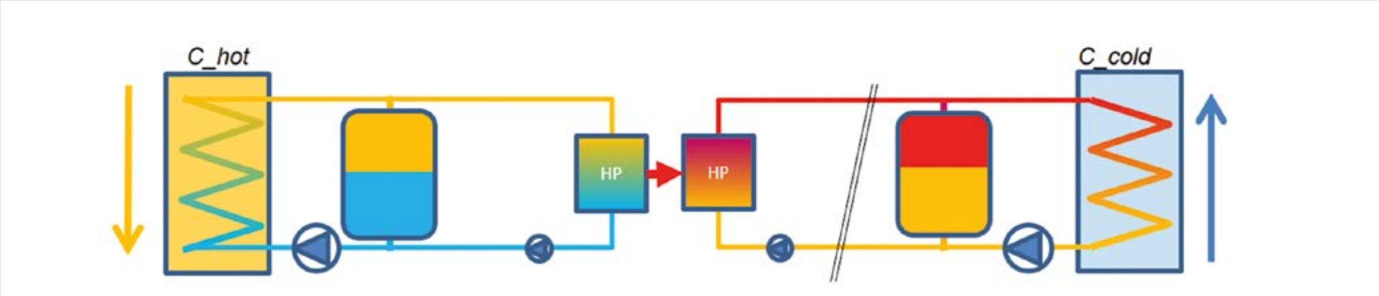

Figure 2 heat transport with continuous power and temperature lift with a heat pump (HP)

ses a secondary pinch based on the utilities is sometimes more attractive. In all cases, not only the momentary power must be determined but also the fluctuations. The power needed for batch processes can be smoothed by storage of heat in stratified fluid buffers, phase changed materials, chemical buffers or in concrete or steel. With storage systems, the heat recovery is more robust and leads to lower investments in piping and equipment.
In most cases, the heat sources and heat sinks are at some distance so a transport medium must be chosen. This can be water, steam or thermal oil. To transport heat with gases is possible but usually not attractive.
Every heat exchanger introduces exergy losses, so the design and the chosen surface area are key in closing loops in a factory. For heat recovery from hot gases a temperature difference of 30-40 K is necessary. For condensing vapours 10-20 K is achievable. For fluid-fluid heat exchangers sometimes 1 K is possible. Clean fluids can be used as transport medium without losses in temperature.
In the approach from Energy Matters, the hot and cold process streams are directly transferred to transport circuit conditions with a continuous power and temperature by using storage systems as seen in Figure 1.
If the temperature levels do not fit, a heat pump, heat transformer or vapour recompression unit is necessary as shown in Figure 2.
If sources are of the same temperature levels they can be combined in waste heat grids. The heat grid at the condenser side also can be combined as a utility to heat up product streams. With the “one day heat pump scan” software tool as developed in the Netherlands by Energy Matters a first impression of the technical and economical potential can be determined.

Technical solutions for electrification

Heat pumps and heat transformers can lift waste heat to temperature levels above 100 oC, up to 140 °C. The temperature lift of a single system is about 50 K. A single compression step of vapour compression is practically 2-4 times the absolute feed pressure. By cascading electrical heat pumps and/or vapour compression (with intercooling) the temperature lift is limited to the critical point of steam (374 °C). The development of thermal acoustic heat pumps can lead to temperatures above 400 °C. For processes with even higher temperatures, electrical heating can be realised by using techniques as Ohmic heat, infrared, induction or microwave, all with a COP of 1. These systems mostly have higher efficiencies than convective heating based on final energy.

Conclusions

Technically, industry can become sustainable in the short time by electrification of all processes . The limitation is the investments needed for the heat exchangers, the internal grids and the heat pumps. Investments are also needed for the (reinforced) connection with the electricity grid. The operational costs can be attractive depending of the tariff structure and the COP of the equipment. Electrification with heat pumps can therefore be an attractive route to go sustainable.

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Electrification in the process industry has potential that requires development

The Dutch industry can make a significant contribution to the climate goals of the Paris Agreement through electrification of its processes. This requires system and process innovation. In this respect, it is useful to make a distinction between different electrification strategies. This is shown in a report carried out by Berenschot, CE Delft, Industrial Energy Experts and Energy Matters, requested by RVO in cooperation with ISPT Industry & Energy. Our energy system is in the middle of a rapid transition towards more sustainable solutions.

Recently, 192 countries agreed on limiting global warming to well below two degrees Celsius during the COP21 in Paris. This means that measures need to be put in place to reduce carbon emissions with 80-95% in 2050 as compared to 1990. Although this transition still comes with many uncertainties, some promising transition pathways towards a more sustainable energy supply are slowly taking shape. One of these pathways is the electrification of industrial processes, which involves opportunities and challenges for both suppliers and end consumers.

If the Dutch industry is to be committed to the goals agreed on during the COP21 Climate Conference in Paris, it means that CO2 emissions have to be reduced higher than the current trajectory as described by the Energy Agenda, as well as in most well-known scenario studies.

Electrification is one of the possible transition pathways for the Dutch process industry to contribute to an environmentally sustainable economy. As the Dutch process industry accounts for approximately one third of total energy use in the Netherlands, the use of (sustainable) electricity in the industry can have a significant impact on CO2-reduction in the Netherlands.

By 2050, the energy landscape will be significantly different compared to how it looks today. To achieve a 80-95% decrease in CO2-emissions and meet the energy demand at the same time, a complete redesign of the energy system is required. With this development, equipment lifetimes of 30 years and more in the industry needs to be taken into account.

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